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Master's degree final project

Experimental investigation of multi-pulse injection strategies for pollutant emissions reduction in a PCCI 3.0 L diesel engine

Academic supervisor Prof. Stefano D'Ambrosio Candidate Davide Spatola

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For the first time in my life I tasted what it really feels like to be an engineer, on the field, everyday: dealing with problems and trying to solve them pouring into the job the best part of you while your colleagues/friends are by your side, ready to help.

But, above all, I wouldn't be here without my family.

And my patience too.

Abstract

This work is the result of a 9 months experience in the Internal Combustion Engines Advanced Laboratory (ICEAL) of Politecnico di Torino.

Although the recent debates on road transportation about the possibility to dismiss a technology like the employment of diesel fuels for propulsion, in the light of a more sustainable environmental impact, the discussions still can't affect much the Heavy-Duty (HD) field where the everyday covering of big distances necessarily calls for the unmatched efficiency and fuel economy offered by diesel engines: in this sector, an improvement of 1% in fuel consumption is a remarkable achievement companies are continuously striving for.

Actually, the fuel saving advantage together with the lower price of diesel fuels have been the main driver for the widespread diffusion of this technology in the private transportation too, at least in Europe and above all in Italy, where 50% of the passenger cars market share was held by diesel models.

In recent times, however, scandals along with the traditionally higher prices of diesel variants, the nowadays' almost equivalent costs of gasoline and diesel fuels and, most importantly, severe circulation restrictions in certain areas applied up to pre-Euro 5 models, have irreversibly ruined the image of Diesel.

But if the mission of a passenger car is to move people mainly around urban areas with sporadic trips outside, the one of HD transportation is to run at full payload day and night over long distances, which explains why the research on diesel combustion has never slowed down.

Low Temperature Combustion is a promising alternative concept whose aim is to make the already fuel efficient diesel combustion smoke-free and NO_X -free too, as the two represent pollutant species whose emission reduction is costly, cumbersome, subject to performance deterioration over time, requiring regular maintenance and in general preferable not to produce at all, specially inside populated areas, being their dispersion in the atmosphere slow and tending to remain in the proximities of the place where they are produced: cities, with major impacts on their inhabitants.

The combustion is then let happen around stoichiometry and temperature levels low enough to prevent the formation of both although few drawbacks like excessive concentration of products of incomplete combustion and considerable volumes of unpleasant combustion noise arise: the former are easy to manage but noise is probably the main reason for the reduced diffusion of LTC strategies.

Freight traffic is much less stochastic than the private one and this predictability can open the way to comprehensive real-time powertrain control strategies able to perform instantaneous calibrations on the basis of data exchanged in a net of interconnected trucks about the driving conditions to be expected in the next stint: manufacturers estimate to reduce in this way substantially the consumption of fuel and urea and the noxious emissions.

All in all, Diesel remains the undisputed technology in the long haulage HD transportation but the research in this field could however prove to be the ideal test bench for the optimization of new concepts eligible to be extended on passenger cars too: the future of cars says *electric!* whatever the point of view but, as a matter of fact,

the music has remained the same for the last 20 years and not only cars don't fly but also they still run on fossil fuels.

Science moves a step further every day but *that* day is still yet to come regardless of how close it looks: it's not a mirage either, it's just reality taking its time to prepare for what will really be the greatest revolution ever in the world of automotive.

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1. Introduction

Premixed Charge Compression Ignition (PCCI) is a type of Low Temperature Combustion (LTC) mode aiming at the simultaneous reduction of soot and NO_X .

It features two variants: early and late; in the former the fuel is injected a lot before the TDC while in the latter just after it, implying that in the first case ambient conditions extend the ID with benefits for the charge preparation quality but also risks of wall impingement issues whereas in the second case the mixture preparation is enhanced by the strong turbulences forming at the beginning of the expansion stroke although more EGR is required to have an ID long enough.

The variant investigated in this work is the early one allowing an extension of the ID for a more uniform mixture preparation with less rich pockets responsible for soot formation and less NO_x, thanks to the recirculation of huge quantities of exhaust gas (EGR) back into the intake manifold to dilute the charge and thus lower the O₂ concentration and combustion temperatures responsible for NO_x production.

This can offer the possibility to review the After Treatment Systems (ATS) usually affected by high costs and durability and packaging issues although the penalties in terms of noise, HC and CO emissions might limit the applicability of PCCI considering also that the reduction of the already low temperatures of a Diesel exhaust can prevent the oxidation catalysts (DOC) from working at high conversion efficiencies, especially at low loads/speeds and even more during the warmup phase.

This alternative combustion mode allows then to get rid of the usual NO_X -soot tradeoff ruling the optimization of a conventional combustion and to attain the simultaneous cut of the two pollutants without any compromise – other than those listed before – as can be seen following the in-cylinder stoichiometry/temperature evolution curve on the Kamimoto-Bae diagram reported in the following Figure 1.



Figure 1 – Combustion modes impact on the production of soot and NO_X

The shape and location of the soot and NO islands is a function of the in-cylinder conditions particular to the operating engine point, but the general behavior remains unchanged: as can be seen, PCCI combustions skim the two islands drastically cutting the production of the relative pollutants.

As said, steep in-cylinder pressure gradients leading to high CN and difficulties in managing very low oxygen concentrations due to the high EGR rates (which risk to make the combustion unstable and poorly cylinder-to-cylinder and cycle-to-cycle

repeatable) generally limit the maximum achievable load to a value around 5 bar of BMEP, which is quite low compared to the 20+ bar attained by a conventional diesel combustion, especially in the field of HD applications.

As it will be explained later, the hardware modifications operated on the series production engine utilized as a starting point for the development of the prototype on which all the following discussion is based, have extended the PCCI range to the medium/high speeds portion of the engine map and up to 8.5 bar of load, with a gain of more than 6 bar of BMEP with respect to the maximum values attainable by running under PCCI conditions on the conventional production engine: 2 bar.

The aim of this work is trying to mitigate the above mentioned side effects of singlepulse early PCCI combustion mode – mainly ascribable to massive injections of fuel into environments particularly favorable to spray penetration and wall impingement – by the usage of multi-pulse strategies, starting from double up to quadruple injection schemes, and trying in each case to find the best calibration by tuning the parameters under control in order to optimize the set of engine points listed below:

- 1400 rpm X 27 Nm (BMEP = 1.13 bar)
- 1000 rpm X 94 Nm (BMEP = 3.97 bar)
- 2000 rpm X 120 Nm (BMEP = 5.02 bar)
- 2000 rpm X 161 Nm (BMEP = 6.75 bar)
- 2000 rpm X 200 Nm (BMEP = 8.39 bar)

As said, the combustion control is exerted by acting on a given set of calibration parameters including:

- Electric start of injection (SOI) of each pulse
- Fuel loading of each pulse
- Intake flap position
- Exhaust flap position
- Variable Geometry Turbine (VGT) position
- Rail pressure
- Main injection quantity: imposed by the bench to sustain the required BMEP

For now, the above list is just meant to give a first idea to the reader as the required explanation is postponed to a dedicated chapter, later in the discussion.

The procedure consists in analyzing the impact produced by each of the calibration parameters on a set of quantities of interest like: BSFC, CO, soot, HC, combustion noise (CN) and NO_X, representing the quality level of the combustion, with the aim to further improve it besides the benefits already coming from the use of smaller jets with reduced inertia: less penetration and risks of wall impingement even under favorable ambient conditions, shorter charge preparation time, higher combustion efficiency, lower pressure rise rates, lower peak-temperatures, extended duration of the heat release for post-oxidations and consequent higher exhaust temperatures and catalyst efficiency.

2. Diesel conventional combustion

The famous study [1] by John Dec focuses on the evolution of a reacting diesel fuel jet from the start until the end of injection explaining the sequence of events that occurs as the fuel moves downstream from the injector into the steady-state jet, including the main pollutants formation processes.

Initial attempts to describe the combustion of a developed diesel jet have been adapted from studies of steady spray combustion in furnaces and gas turbines due to the lack of consistent information and direct measurements: a nearly pure fuel core with diffusion flame combustion occurring either around individual droplets or as a sheath around the jet periphery (depending on the ratio of the heat exchange between the liquid and vapor phases to the fuel heat of vaporization) after an initial premixed burn phase taking place at equivalence ratios close to the stoichiometric value.

The concept was that the fuel concentration dropped off with a Gaussian-like profile (centered along the injector axis) toward the jet periphery and that the combustion zone occurred at some distance from the center where the mixture was appropriate.

According to this model, since soot formation requires rich mixtures and temperatures higher than 1300 K thus not reachable simply by mixing the fuel with the entrained incylinder air (1000 K), combustion heating is required for this purpose: it was therefore assumed that soot would form on the rich side of the diffusion flame, where temperatures were high enough, neglecting any possible contribution from the premixed burn that indeed was thought to happen in regions that were nearly stoichiometric.



Figure 1 – Old conceptual model of the conventional diesel combustion

Once new techniques offered for the first time direct detailed measurements of the diesel combustion process, the old description, summarized in the schematic representation of Figure 1, started to be questioned as soot was found in the central regions of the jet rather than at its periphery.



Figure 2 – LII signal intensity of the soot relative concentration in volume

In Figure 2, the LII images show the soot relative concentration in volume (false color scale shown at the right) and the intensity of the signal is proportional to the particle diameter to the third power: LII captures the spatial distribution of smaller particles.

The ES images show the same thing as far as soot is concerned but they also make liquid fuel visible, if present; the intensity of the signal this time is proportional to the particle diameter to the sixth power: ES captures the spatial distribution of bigger particles.

The blacked out portions in the ES images are due to the horizontal laser sheet hitting the top of the central cone on the "Mexican hat" piston used in this experiment and creating an intense background scatter.

It is very easy to see how the two image types are quite different upstream of the head-vortex: the LII images show soot throughout the jet while the ES signal is very weak.

This result indicates first of all that there is no liquid fuel in this region, given the fact that otherwise the signal would have been very strong, having the liquid droplets a much bigger size than the soot particles' one, and secondly that soot in the upstream region is much smaller compared to that in the head-vortex.

These findings, combined with the increase in soot concentration along the jet axis, suggest that the soot formation starts in the upstream regions of the jet and proceeds along with particles growth as the soot moves down toward the head-vortex.

Typically, the AHRR curve – as depicted in Figure 3 – first goes negative right after the SOI as the in-cylinder air loses heat to the liquid fuel in order to vaporize it and then rises quickly as the vapor fuel accumulated during this ignition delay burns almost all at once when the proper ambient conditions are met: this phase is known as *premixed combustion* and precedes the *mixing-controlled combustion* in which the diesel spray structure, subject of this study, has evolved to its final form before the EOI.



Figure 4 shows a temporal sequence of liquid fuel images obtained by ES and referring to the heat release recorded in Figure 3.



Figure 4 – ES images of the fuel's liquid column penetration in time

Initially, all eight fuel jets are well illuminated and each of them appears to be very similar to the others; this has been proved to be true even after 2.0° ASI, when the tips of most of the jets have traveled outside of the laser sheet and are no longer well illuminated: only the jet at 3 o'clock remains clearly visible throughout the sequence being aligned with the laser sheet in order to determine the liquid penetration length.

Beyond 3.0° ASI the liquid length stabilizes: a look to the AHRR curve to which the above picture refers can confirm that it goes positive only around 4.0° ASI, namely a full crank angle degree after the liquid length peaks, proving that the reason of it is not to be found in combustion heating but in evaporation heating coming from the hot incylinder air entrained by the jet during its penetration inside the combustion chamber. Various studies have also shown that the maximum liquid length: decreases with increasing temperatures and pressures, is independent from injection pressure and increases with increasing diameter of the injector holes.

An important final remark might be the following: the sequence of images above say nothing about the simultaneous vaporization that is taking place inside the liquid column; the liquid length represents only the point beyond which the last droplets of fuel vaporize: this undoubtedly implies that vaporization shall start much earlier, close to the injector.



Figure 5 – Mixture strength' time evolution past the maximum liquid length

Figure 5 displays how the cloud of vaporized fuel and air continues its travel across the combustion chamber extending beyond the liquid length – not shown but tangent to the left side of each image; it is obtained employing the PLRS technique and converting the raw scatter images to equivalence ratio fields mapped to the false color scale at the bottom of the figure.

The reacting sequence was obtained using a reference fuel with cetane number 42.5 while the non-reacting sequence using a reference fuel with cetane number 15: the two columns exhibit comparable images up to 4.5° ASI, featuring equivalence ratios ranging from 2 to 4 and a sharp neat jet boundary, but after 4.5° ASI the sequences diverge significantly because the reacting jet undergoes the premixed burn phase.

The onset of combustion is clearly visible in the picture of the reacting jet at 5.0° ASI: the signal intensity drops significantly throughout the cross section of the leading portion of the jet to the point of disappearing in large regions by 6.0° ASI; it should be noted however that during this phase such images are no longer quantitative and serve only for comparison with those on the right.

As far as the non-reacting jet is concerned, during the very same time interval between 4.5° ASI and 6.0° ASI, the pictures tell that the mixing with the surrounding air induced by the jet penetration cause the equivalence ratios to progressively decrease although at a slow rate.

The loss of intensity in the reacting picture at 5.0° ASI is mainly caused by fuel breakdown: since this happens in regions that are rich but still ignitable at the same exact moment of rapid rise in the AHRR curve, it is possible to conclude that in contrast to the old description, the premixed burn phase cannot be stoichiometric.

To conclude, at the beginning it was stated that the above pictures are taken immediately downstream the liquid penetration length but it has been proven that there is also strong evidence of vapor fuel along the sides of the liquid column, linearly increasing in thickness with increasing distance from the injector so as to form a nearly constant cone angle.

A weak flame emission due to chemiluminescence can be noted even during the fuel vaporization when the AHRR curve descends toward negative values: it is the result of energetic autoignition reactions and can be assumed as the true starting point of combustion; this early, the vapor has not yet penetrated beyond the liquid and actually the above mentioned chemiluminescence mostly comes from the vapor at the sides of the liquid column.



Figure 6 – Flame chemiluminescence and soot-precursors fluorescence at the SOPC

The number in the upper right of each of the images in Figure 6 above gives the relative intensifier gain in arbitrary units with respect to the raw signal acquired for that specific image itself and the field of view is the same as the one described for the previous figure.

The gain confirms that at 5.0° ASI the natural flame emission due to chemiluminescence is about twice as bright as that at 4.5° ASI and still uniform over the entire downstream portion of the jet.

Half a crank angle degree later however, an even brighter region appears over the lower half of the jet: its size, shape and location are completely random being subject to cycle-to-cycle variability but it is very likely to be ascribed to early soot luminosity.

By 6.0° ASI the brightness extends throughout the leading portion of the jet and is 60 times stronger than that in the previous frame suggesting that it can only come from soot luminosity: a picture taken half a crank angle degree later, not reported here, would show a brightness 60 times stronger that the one at 6.0° ASI itself.

All in all, the figure shows that chemiluminescence arises gradually over the entire downstream region of the jet implying that autoignition occurs at multiple points nearly simultaneously but it does not say whether reactions are confined to the surface of the jet or take place inside the whole volume enclosed by it.

To answer this last question, PAH PLIF images at the right of the same figure can provide some help: as said earlier, chemiluminescence starts to be visible as soon as 3.5° ASI but the PAH signal is barely visible prior to 4.5° ASI and the abrupt leaning of the equivalence ratio seen in the previous figure is not expected to happen before 5.0° ASI, when the PAH signal becomes substantial too and the AHRR curve finally ramps up.

In the light of all these findings, one shall conclude that the very early light emission is not related to fuel breakdown.

Fuel breakdown and PAH signal reinforcement thus continue until at 6.0° ASI some very bright regions appear: their shape and location is again randomly variable cycleby-cycle but a rapid comparison to the corresponding natural flame emission image should make immediately clear the fact that also this sudden increase in PAH signal intensity is due to soot luminosity.

The answer to the initial question is then that the majority of the premixed burn occurs volumetrically throughout the cross section of the leading portion of the jet.

Figure 7 below shows images of early soot formation starting from 6.0° ASI, when the soot is first detected as said in the previous paragraphs, until near the end of the premixed burn.

Again, the field of view is the same seen in the two former figures and the intensifier gain still has the same meaning.

After the first two image pairs it was decided to switch to a low-sooting fuel because significant optical attenuation occurs with regular fuel by 7.0° ASI: they both behave the same way as far as ignition and early soot formation is concerned up to the point when soot obscures the images relative to the reference fuel.



Figure 7 – Early soot formation at the EOPC

As already discussed, LII images trace the spatial distribution of smaller particles while ES images do the same with bigger ones.

The very first soot at 6.0° ASI was detected by means of the LII technique only that indeed shows formation of particles occurring throughout the leading portion of the jet; this is further confirmed by the simultaneous ES image: the area of strongest signal does not coincide with the sooting region evident in the LII image suggesting that the ES signal mainly arises from the fuel vapor and that this early soot is given by very small particles.

During the following half crank angle degree, the soot concentration increases by a factor of 2.5 and the ES image reveals a strong-signal region around the periphery of the jet although in the central portion it is still weak; in those zones the signal is much stronger than that typical of vapor fuel and its boundaries exactly match the outline of the soot in the corresponding LII image thus implying the presence of particles that have grown quite large at a very fast rate: a different mechanism must be responsible for this phenomenon also taking into account that its timing coincides with the formation of the diffusion flame around the jet periphery, right before the midpoint of the premixed burn phase.

At 7.0° ASI the soot concentration is a lot higher than that visible in the previous frame although this fact is masked by the switch to the low-sooting fuel, and the shell of larger particles at the jet periphery has become thicker probably because of turbulent eddies transporting some of them inwards, being the diffusion flame confined the whole time on the jet surface as OH images prove.

Following the sequence up to the end, the intensifier gain numbers show how robust is the soot formation; concentrations also increase moving from the injector tip to the jet leading edge and the peripheral shell of soot keeps getting thicker.



Figure 8 – Soot temporal/spatial evolution throughout the mixing-controlled burn

The images in Figure 8 above describe what happens through the mixing-controlled burn up to the end of combustion; since a low-sooting fuel was demanded to mitigate optical attenuation, the intake conditions were adjusted to provide a lower TDC temperature in order to get an AHRR curve similar to the one recorded with a commercial blend. The field of view is still the same investigated in the previous figures.

The sequence starts recalling some images of the previous figure describing the latter part of the premixed burn; at 7.0° ASI, LII shows that soot is present all around the luminous part of the jet cross section with a slightly higher concentration close to the leading edge; this tendency consolidates at 9.0° ASI, coinciding in timing with the start of the mixing-controlled burn phase, and evolves through the remainder of the sequence toward the formation of a head-vortex containing large amounts of soot.

The LII signal still detects soot formation in the upstream region of the jet both at 9.0° ASI and 11.0° ASI although at this point the premixed burn should already be concluded and drops drastically, again in the very same region, at 13.0° ASI; the simultaneous ES images show a strong rapidly intensifying signal, indicating larger particles, only at the periphery of the head-vortex with a weaker signal upstream barely distinguishable from the background scatter caused by the laser light impinging on the combustion chamber walls.

All in all, the first set of pictures shown above tells that throughout the first part of the mixing-controlled burn: soot is present all over the jet cross section with increasing concentrations and dimensions moving down along the spray axis according to a pattern that persists mostly unchanged until the EOI, halfway through the mixing-controlled phase.

The second set of pictures refers instead to what happens after the EOI at about 18.5° ASI. The LII and ES signals at 19.0° ASI, immediately after the EOI, show a high concentration of large particles developing in the upstream portion of the jet; by 21.0° ASI, the jet features two separated high concentration zones of large particles: one in the upstream region and the other in the head-vortex.

Eventually, signals decay as the soot burn by the diffusive flame is no longer replaced by the newly produced one until, at the time of the final image, the HRR is complete and they fade away. This change in the soot distribution with the EOI might be due to the injector needle throttling the last portions of injected fuel as it closes, causing it to flow at a reduced speed with no possibility to catch up with the leading portion of the jet, that thus detaches; bad atomization combined with the reduced probability to reach the remaining fresh air in the cylinder for the reasons just explained lead then to the high degree of soot formation and particles growth evidenced experimentally.



Figure 9 – OH radicals PLIF signal intensity revealing the diffusion flame time development

Figure 9 is a series of OH radical PLIF images showing the diffusion flame development; the strong scattering from the liquid remains however clearly visible. At 5.0° ASI there is no detectable OH and the ignitable mixture extends well out beyond the liquid, as indicated by the dashed line, where it undergoes the premixed burn that being fuel rich explains why there is indeed no evidence of OH.

At 5.5° ASI the first OH is detected at the sides of the premixed-combustion area as indicated by the bright spots "a" and "b", the first referring to the jet under investigation at 3 o'clock thus directly hit by the laser, the second corresponding to the same location in the adjacent jet. By 6.0° ASI, OH radicals have extended all around the periphery of the leading portion of the jet and during the next half crank angle degree they finally establish the *lift-off length* by protracting toward the injector up to a point just upstream of the liquid penetration length.

Through the rest of the sequence, ending at the start of the mixing-controlled burn, OH continues to be detected where the diffusive flame is supposed to be.

All these findings, combined with the fact that OH equilibrium conditions are so strict to drop drastically just outside of the proper environment: stoichiometric and very hot like the one on the surface of the diffusive flame, suggest that the OH PLIF signal closely marks the diffusion flame zone.

As a final statement, it should be noted that when at 6.5° ASI the flame sheath encloses the jet, larger soot particles begin to form nearby: this is not a coincidence.

Attenuation effects on the OH signal relative to the jet under examination begin at 7.0° ASI; the intensity is very low in the upstream region labelled "c" but corresponding spots "d" on the adjacent jets discard the hypothesis that OH radicals are truly missing: as the laser traverses the fuel jet, its light is scattered or absorbed and the situation gets even worse when the optical attenuation ascribable to soot becomes relevant, but reality is that the diffusive flame is always there, keeping its peculiar characteristics virtually unchanged until the EOI, halfway through the mixing-controlled burn.



Figure 10 – New conceptual model of the conventional diesel combustion

The scheme in Figure 10 above shows the development of a diesel jet from the start of injection through the premixed burn up to the first half of the mixing-controlled phase, prior to the end of fuel injection: it is a tentative to summarize all the previous findings into a single picture able to grasp the concept of the conventional diesel combustion.

In a real jet the cycle-to-cycle variability always affects lengths and symmetries and its boundaries are always ragged due to small-scale turbulence: irregularities and gradients are smoothed out if pictures are cycle-averaged but this is not the case for the above representation which just wants to be as simple and direct as possible.

The sequence starts with the liquid fuel being injected (1.0° ASI); during its travel across the combustion chamber, the spray entrains air and vaporizes progressively developing a gaseous mixture that runs along the sides of the liquid column (2.0° ASI) up to when the latter reaches its maximum penetration (3.0° ASI): at this point the vapor overcomes the liquid and proceeds alone (4.0° ASI).

Double arrows show that chemiluminescence actually starts to be visible as soon as 3.0° ASI in the vapor region along the sides of the liquid suggesting that the exact point of ignition is not well defined either in space and time.

By 4.5° ASI however, the leading portion of the jet contains a relatively uniform, very rich but still ignitable mixture emitting a strong light: here, volumetric fuel breakdown takes place leading to PAH (5.0° ASI) and soot formation (6.0° ASI) as indicated by the very sharply rising AHRR.

By 6.5° ASI, just prior to the midpoint of the premixed burn spike, small soot particles are found throughout the downstream portion of the jet around which a thin diffusion flame has extended as a sheath; as depicted in the schematic, the liquid length shortens a bit due to local heating by the flame and the soot nearby the latter becomes larger forming a shell of bigger particles, not shown in the scheme, again due to some effect by the immediately adjacent flame: it is important to note anyway that this last finding is not accompanied by a simultaneous increase in soot concentration.

Through the remainder of the premixed burn, the soot concentration increases along the spray axis reaching the highest levels in the proximity of the head-vortex (8.0° ASI) and the shell of larger particles at the boundaries of the plume thickens, probably due to turbulent mixing, even though the center continues to be filled with fine particles only.

As the combustion transitions to being purely mixing-controlled, the overall appearance of the jet doesn't change much: this is probably because it was already almost in a mixing-controlled burn mode even during the latter premixed-burn phase; by 10.0° ASI, the head-vortex features the highest particles' concentrations and dimensions and what is shown below is the typical sketch of the jet overall appearance evolved to its final steady-state form, kept substantially unchanged until the end of fuel injection.

Eventually some of the soot reaches the diffusion flame where it can be oxidized by OH radical attack. The figure also shows the presence of a thin soot layer upstream of the location where soot forms across the entire central part of the jet namely, at the sides of the rich mixture about to undertake the premixed-burn, in the immediate proximity of the diffusion flame, forming a blunt conical shell: this zone, pointed by the blue arrow in Figure 11, can be detected only when using the reference high-sooting fuel

and is expected to be caused again by the local heating induced by the very close diffusion flame, promoting fuel pyrolysis and soot formation upstream of the premixed-burn zone itself.



Figure 11 – Zones of the steady-state jet requiring further investigations

The premixed-burn zone, pointed by the red arrow in Figure 11, needs some clarifications too; the sudden, robust soot production downstream of it for a great extent of the combustion event cannot be the result of hot air entrainment only (1000 K) since soot formation typically requires much higher temperatures (1300 K): as shown schematically in the following Figure 12, a standing fuel-rich premixed flame (light blue) just upstream of where the soot first appears, seems to be the most logical explanation.

Although the diffusive flame has built up and the liquid column has shortened, the entrainment through the lift-off length would still provide enough air to form a uniform even richer but again ignitable mixture compared to what happens prior to the establishing of the diffusive flame: equivalence ratios now ranging between 3 and 5 instead of 2 and 4; this mixture will then serve to feed the above said standing premixed flame that is thought to light up during the later stages of the premixed burn, from about 6.0°ASI to 8.0° ASI.



Figure 12 – Salient features of the new conceptual model for the conventional diesel combustion

The presence of this flame throughout the mixing-controlled burn would mean that all of the fuel first undergoes fuel-rich premixed combustion and then diffusion-flame combustion, the latter occurring between the products of the fuel-rich premixed combustion themselves and the remaining, almost pure, in-cylinder air.

As shown by the grey zone in Figure 12, the initial soot formation occurs just downstream of the standing premixed flame, in the products of its fuel-rich combustion; soot formation and particles growth then continue moving down to the head vortex up to reaching the diffusion flame, where the OH radicals attack completes its oxidization, at least ideally.

As far as NO production is concerned, both the initial premixed burn just after autoignition and the one occurring inside the standing premixed flame during the mixing-controlled phase cannot lead to NO formation either by the *thermal* or the *prompt* mechanisms: scarce oxygen and too low temperatures prevent the former from playing an important role while too rich mixtures, well above the equivalence ratio threshold of 1.8 observed experimentally, are instead the reasons why the latter mechanism is not significant too; however, some HCN production might still occur and the nitrogen thus *fixed* would later be released as NO at the diffusion flame.

At the diffusion flame temperatures are high, combustion is nearly stoichiometric and there is a great abundance of free oxygen: these conditions are ideal for NO formation that accordingly takes place on the lean side of the flame, as indicated by the green line in Figure 12.

Nevertheless, the one just seen might not be the location where most of the NO is produced: the thermal mechanism is a relatively slow process and this could delay the onset of significant NO production until the latter parts of the mixing-controlled burn or even in the hot-gas regions remaining in the chamber after the end of combustion; although peak temperatures might be lower, there is considerably more time at disposal and the volume of gas involved is much larger.

As a concluding remark, the differences with respect to the old description of the conventional diesel combustion are mainly due to a few facts that may have not been fully appreciated by those who developed it: in modern diesel engines, injection velocities are so high to cause the diffusion flame to stand off from the injector at a distance known as the *lift-off length*; the air entrainment still allowed by this lifting is enough to sustain a standing fuel-rich premixed flame throughout the mixing-controlled burn up to the end of injection; for turbulent jets, the instantaneous picture can be very different from a statistically averaged picture, the type upon which a vast majority of the early diesel spray theory was based, in contrast with the new model.

The new phenomenological description is though applicable in case of no swirl or wall interactions only and requires to be extended to cover the burnout phase after the end of fuel injection.

3. Diesel PCCI low-temperature combustion

The famous study [3] by Mark Musculus proposes conceptual models for a low temperature, low-load, single injection, partially premixed compression ignition condition where the dilution by EGR is high enough to bring the oxygen concentration at the intake in the range of 10 - 15%.

With respect to the conventional combustion, a low-temperature type features: longer liquid penetration, an extended ignition delay that allows more premixing of the fuel, a more distinct and temporally extended two-stage ignition, reduced and altered soot formation regions and an increased overmixing leading to incomplete combustion.

Historically, the first emission targets were loose enough to be achieved simply with a good tuning of the combustion system; the later targets, however, required also the installation of after-treatment systems with increased costs, durability issues, fuel consumption penalties and packaging constraints: improvements to in-cylinder strategies to further reduce the engine-out emissions and thereby rely less on the ATSs, with all the subsequent advantages, are then of great interest.

In diesel engines, the primary contribution to NOx is given by the Zeldovich's thermal mechanism whose production rates have an exponential dependency on the temperature: accordingly, recent research is focusing on reducing combustion temperatures by dilution either with excess charge gas to create mixtures that are more fuel-lean than stoichiometric or with moderate to high levels of EGR.

Low temperatures also slow down the formation kinetics of soot but at the same time they hinder its oxidation even more with the final result being an increased engine-out emission, as EGR increases for example; the statement holds until very high levels of EGR are used: at this point, soot formation is so low that engine-out emissions decrease even with reduced oxidation, unfortunately at the cost of excessive fuel consumption, UHC and CO emissions due to the very poor combustion efficiency.

Hence, rather than using such extreme EGR rates to deal with soot emissions, the idea is to pursue an extensive pre-combustion mixing with the aim to limit the fuel-rich regions from which soot arises.

As said before, the subject of the study cited at the beginning of this chapter, is a type of LTC combustion known as "Premixed Charge Compression Ignition", PCCI in short, which uses direct injection to create a charge distribution still heterogeneous at ignition but premixed and diluted enough to abate NOx and soot engine-out emissions: this strategy allows to use the conventional hardware to achieve LTC conditions keeping a good control of the heat release, thanks to the ignition event remaining closely coupled to the fuel injection event, in contrast with the ideal, hard to practice HCCI concept.

For conventional conditions, fuel is injected slightly before TDC and thus ignites immediately afterwards with the injection still ongoing: a negative *ignition dwell*, which is the time from the end of injection to ignition, is therefore marked; as a result, only a small portion of the fuel is premixed prior to initiation of combustion.

For PCCI conditions, the ignition dwell is positive – although short enough to ensure that the jet structure remains intact up to the moment of ignition – meaning that to some extent all the fuel undergoes a premixing phase before burning; it is achieved by

means of high EGR rates and shorter injection durations, which make this strategy suitable to low-load operations only, considering also the dangerously steep pressure rises and noise that would be obtained at high fueling rates.

PCCI strategies further divide into *early* and *late* injection timing (with respect to TDC, the reference for conventional combustion): in any case, the gases into which the fuel is injected are either initially cooler and less dense because of less compression (early) or become cooler during and after the injection because of expansion (late); combining these environmental conditions with the delaying effects of EGR on the ignition chemistry, a relevant premixing can be obtained.

So far, the advantages in terms of NOx and soot emissions provided by LTC strategies have been discussed but, at the end, it is also necessary to remember that these always come at the price of increased CO, UHC, noise and fuel consumption, often exceeding regulated limits.



Figure 1 is a series of images showing the initial penetration of liquid and vapor fuel for an early PCCI condition at a load of 4 bar (IMEP) in a heavy-duty engine; diesel fuel absorbs ultraviolet light very strongly: as a result, for the jet at 3 o'clock under investigation, the laser sheet entering from the right is appreciably attenuated by the fuel at the tip of the spray and by 3.0° ASI the upstream regions of the jet appear dark, gradually fading into a complete lack of signal before the EOI. Therefore, the intensity of the fluorescence emission should not be interpreted as an indication of the local fuel concentration but rather as a simple preliminary view of the temporal and spatial history of the leading edge of the jet.

In addition to the fluorescence imaging, a simultaneous visualization of liquid fuel by Mie-scattering is provided in the same picture but the laser sheet used for this task was at a much lower frequency: this implies the possibility to have reliable signals throughout the sequence being the attenuation phenomenon not as severe as in the previous paragraph; the liquid images however are not shown directly but have been reduced to a contour line indicating the outer boundary of the liquid column.

The overall image thus provides a first idea of the relative spatial distribution of liquid and vapor fuel: the fluorescence signal arising from within the contours may be due to both liquid and vapor fuel while the one outside them shall come mainly from the vapor phase.

The initial spray penetration displayed in the figure above is very similar to that of a conventional diesel combustion with the exception that the liquid length is longer and the jet velocity higher because of the early timing and the higher injection pressures (1600 bar vs. 600 bar), respectively.

By 3.0° ASI the liquid length is established and the vapor continues penetrating on its own; because of the signal attenuation due to the fuel absorbing the laser's ultraviolet light, fluorescence from the upstream regions of the horizontal jet under examination is weak but the corresponding areas on the adjacent jets, still able to receive some direct light, indicate that fuel is obviously present there for the whole sequence duration.

From 4.0° ASI to 6.0° ASI, the liquid length remains approximately constant at the maximum value permitted by these conditions while at 5.0° ASI the injection rate peaks to end near 7.0° ASI after a ramp-down of about 2.0°, which is approximately 300 μ s at the speed of 1200 RPM used for the experiment, during which the liquid penetration shortens up to disappear at 8.0° ASI.



Figure 2 – Liquid fuel penetration at reduced temperature/pressure intake in-cylinder conditions

Figure 2 above shows liquid fuel Mie-scatter images for an operating condition with cooler and less dense ambient gases; the blue contour outlines the boundary of the liquid column when the signal becomes too weak to be easily recognized.

For these conditions, the injection duration is so short that the liquid column doesn't last long enough to establish the quasi-steady maximum penetration length and the EOI ramp-down is also quite fast: these two peculiarities cause the liquid not to retreat back to the injector as seen in the previous figure; when the injection ends 360 µs ASI, the liquid fuel detaches from the injector and remains downstream until about 1000 µs ASI.

Operating conditions intermediate between those of the present and the ones of the previous figure would produce liquid fuel distributions that split after the EOI, with some fuel remaining close to the injector and some other far downstream, with a liquid-free gap in the middle, as shown in the following Figure 3.



Figure 3 – Liquid fuel distribution for intake in-cylinder conditions intermediate between those of Figures 1 and 2

Although the most intense elastic scattering is observed nearby the injector, distinct regions of liquid fuel can be found both in the bowl and in the clearance volume.

The curious vaporization behavior of liquid fuel after the EOI suggests that something in the mixing process must change during the injection ramp-down: the fuel-specific entrainment, namely the mixing rate per unit of fuel injected, shall increase; it is not immediately clear however if this happens because the air entrainment increases or because the air entrainment decreases but only less rapidly than the fuel delivery.

To answer this question, a simplified one-dimensional numerical jet model was developed: when the fuel flow from the injector stops an axial mass flux deficit arises near the injector, assuming a constant jet spreading-angle which is consistent with multiple experimental observations, and because of continuity constraints, the ambient air entrainment has to increase to compensate for the reduced mass flux at the nozzle.

The momentum exchange between the jet fluid and the almost standing surrounding air further slows down the former, reinforcing in turn the mass flux deficit, thus driving even more entrainment: the result of this loop is the building of a wave of *enhanced entrainment* that grows and propagates along the spray axis, locally inducing up to three times the amount of air dragged-in at a quasi-steady condition, before the EOI.

At a certain point however, after the passage of the entrainment wave, the axial velocity of the jet becomes so low to make it stagnant, with entrainment falling below that of a steady jet and eventually approaching zero; despite this fact though, local

equivalence ratios keep leaning-out as a result of the simultaneous decrease of fuel flux, resulting in a fuel-specific entrainment still significantly higher compared to the steady jet.

As a summary, with a short liquid length, the entrainment wave reaches the tip of the liquid column quickly, increasing the amount of air drawn-in and completely vaporizing the fuel downstream (closer to the boiling point) first, causing the liquid length to retreat back to the injector; with a long liquid length, the entrainment wave doesn't make it in time to reach the head of the liquid column before any vaporization in the upstream regions take place, causing the downstream portions of the liquid spray to detach from the injector while the others retreat back towards it in the same manner seen before.

Finally, splitting of liquid after the EOI like that shown in the previous figure is an intermediate situation caused by an entrainment wave not strong enough to complete the vaporization of the liquid spray portion remaining nearby the injector, thus leaving a gap in between.

The final outcome of this long discussion may be the following: at the EOI, fuel near the injector can experience greater mixing than fuel farther downstream causing the axial equivalence ratio distribution to rapidly invert from the quasi-steady jet behavior, according to which it varies inversely with the axial distance from the injector.

Quantitative experimental measurements confirm the model predictions, as reported in Figure 4, showing ensemble-averaged equivalence ratio contours calculated from PLIF of toluene, which was introduced as a tracer for the fuel.



Figure 4 – Mixture strength' time evolution at the EOI

At 0° AEI, the equivalence ratio distribution along the spray axis is still the one characteristic of a quasi-steady jet, with the richest mixtures nearby the injector; only 1° later, however, the peak equivalence ratio near the injector falls from 9 to 3, becoming at most stoichiometric by 3.0° AEI while downstream mixtures enrich to the point of inverting the quasi-steady equivalence ratio distribution.

By 5.0° AEI, mixtures near the injector are so lean that their equivalence ratio is as low as 0.5. Since the LTC strategy requires a positive ignition dwell, this type of mixing field is to be expected at the start of combustion.

This rapid leaning behavior after the EOI is apparent in instantaneous images as well, as those grouped in Figure 5 below.

Here, the mixing to stoichiometric and fuel-lean near the injector is even faster due to a shorter ramp-down period (100 μ s) with respect to the previous figure (300 μ s) giving rise to a stronger entrainment wave.



Figure 5 – Mixture strength' time evolution at the EOI after a faster fuel delivery rate ramp-down

In addition to increasing mixing rates after the EOI, the entrainment wave also limits excessive penetration of liquid fuel, which can be problematic under some LTC conditions, especially for small light-duty engines: as described earlier, the momentum exchange with an augmented quantity of stagnant air induced by the intensifying entrainment wave, slows the jet down progressively as it travels across the combustion chamber drawing more and more air in during its penetration; combining to this effect the simultaneous faster vaporization of the fuel, the final result is a reduced liquid length and a reduced risk of wall impingement.

Under LTC conditions, the initial ignition reactions proceed at a slower rate than for conventional combustion, putting in evidence the two individual stages of autoignition typical of diesel-type fuels.



Figure 6 – AHRR typical of a conventional diesel combustion

The one in Figure 6 is the typical AHRR of a conventional diesel combustion, widely discussed in the previous chapter: during the premixed burn, the AHRR is primarily controlled by the chemical kinetics in the distribution of mixtures formed prior to its rapid rise and by the total mass of fuel accumulated, now reacting all together; during the mixing-controlled phase instead, combustion occurs both in the standing fuel-rich premixed flame and in the diffusion flame at the interface between the products of the latter and the surrounding in-cylinder air: the overall burning rate is then controlled not by kinetics but rather by mixing, namely by the rate at which the two flames just mentioned are fed; at the end of injection, combustion occurs solely within the diffusion flame.



The one above is the typical AHRR under early PCCI conditions for a load similar to the one of the previous figure; the plot evidences a relatively small and short-lived first stage of positive heat release falling back to near zero before rising rapidly again in a second stage of ignition whose peak magnitude is about three times that obtained in the conventional combustion.

After that, the signal falls to nearly zero showing a marginal heat release throughout the mixing-controlled burn.



This last picture – Figure 8 – refers instead to late PCCI conditions under comparable levels of load with respect to the previous cases; with respect to the early PCCI, the premixed burn spike is less prominent but the heat released during the mixing-controlled phase is again not negligible.

Consequently, for both early and late PCCI conditions, the AHRR preceding the mixingcontrolled portion may be roughly divided into four different parts, each corresponding to a distinct phase of the chemical kinetics of ignition:

- First-stage pre-ignition
- First-stage ignition
- Second-stage pre-ignition
- Second-stage ignition

Commercial diesel blends contain thousands of chemical species among which a large fraction is represented by long straight-chain alkanes that give the ignition chemistry its typical two-stage character; a considerable insight into the process may then be gained through the analysis of a single straight-chain alkane (C_7H_{16} : n-heptane) serving as a surrogate for common diesel fuel.

A chemical kinetics simulation revealed that two different peroxide species rule the two separate stages of autoignition.



Figure 9 - Main radical/intermediate/pollutant species time evolution throughout the combustion event

Figure 9 above shows that during the first-stage pre-ignition period a pool of KHPs (ketohydroperoxides) is build up: this is the central player in the first-stage ignition; various oxidation reactions involving radical species and hydrocarbon fragments occur raising the temperature up to 850 K and hindering the formation of KHPs that commence to decompose instead.

The pool then rapidly depletes forming the radical species necessary for oxidation and heat release, that indeed peaks at this exact time; when it is finally exhausted, the AHRR also decreases significantly signaling the end of the first stage of ignition and the establishment of the so called *negative-coefficient regime* wherein the heat release rate slows despite the ever growing temperatures.

At the end of the first stage of ignition, three important species are produced: H_2CO (formaldehyde) whose subsequent evolution closely remarks that of UHCs offering the possibility to exploit it as a UHC tracer for diagnostic purposes, being easily accessed by PLIF techniques; H_2O_2 (hydrogen peroxides) namely the central player in the second-stage ignition; CO although most of it is produced later, in the early part of the second stage of ignition.

After the first stage, formaldehyde is consumed to build a relatively stable pool of H_2O_2 ; as the already available OH and other species continue to oxidize unburned hydrocarbon species, the temperature gradually rises until 1000 K are reached and the pool of hydrogen peroxides becomes unstable, suddenly decomposing into a pool of OH that yield runaway oxidation of the remaining hydrocarbons leading to the rapid temperature rise of the second stage of ignition, where most of the fuel chemical energy is released.

Late in the second stage of ignition, temperature approaches its maximum and UHCs and CO eventually disappear causing the OH population to spike up again to many times its level since the H_2O_2 pool decomposition: this event ultimately marks the end of the second-stage ignition.

As a final remark, it should be noted that such a sequence of reactions is the same for conventional combustions with the only exception that higher temperatures, richer mixtures and lower dilution levels accelerate the transition between phases so that the individual stages cannot be distinguished simply by a direct reading of the AHRR curve. For mixtures that are richer or leaner than stoichiometric, the general description is the same but the timing and the fate of UHCs and CO after the second-stage ignition, are different: richer mixtures, for example, achieve second-stage ignition earlier due to the higher temperatures at the end of the first stage, already approaching 1000 K, threshold at which the pool of hydrogen peroxides quickly decomposes into a pool of OH radicals.

Figure 9 also shows that formaldehyde forms at the first-stage ignition when the pool of KHPs is depleted and persists until it is destroyed at the second-stage ignition: being a key species for laser diagnostics, by monitoring its duration throughout the dwell between the two stages of ignition whose extension is affected by the temperatures in the way just explained, it should be possible to estimate the mixture stoichiometry.

Also, NO formation is rapid in zones of high OH concentration and the two trends indeed mirror each other: again, OH can provide an indirect and qualitative indication of likely NO distributions and regions of formation which is useful, considering the diagnostic difficulties associated with direct optical detection of NO.

An important final observation might be the following: the CO peaks in the tiny time slot between the destruction of formaldehyde and appearance of OH.



For what has been said, optical diagnostics of formaldehyde can be used to probe the first-stage ignition under LTC conditions.

Above, on the left portion of Figure 10, a set of two instantaneous PLIF images of formaldehyde for a late PCCI in a heavy-duty engine is shown.

The colorbar at the bottom is spatially discretized and shall be read in the following manner: red corresponds to formaldehyde and black to PAHs, with obvious meaning of the intermediate shades, while the false-colored red signal indicates the magnitude of the concentration of the relevant species.

On the right of the same figure are instead ensemble-averaged contours of equivalence ratio measured under non-combusting conditions applying PLIF techniques to toluene fuel tracer; the laser sheets used for the task were aligned parallel to the firedeck and therefore images don't show a cross-section along the jet axis, causing the apparent lack of signal in the regions nearby the injector.

Formaldehyde suddenly appears at 7.0° ASI near the peak of the first stage AHRR, as highlighted by the circles in the plot at right, throughout most of the jet; this distribution remains intact until the onset of the second-stage ignition around 11.0° ASI and is simultaneously accompanied by chemiluminescence emission.

The contour map gives evidence of the fact that formaldehyde is detected across zones characterized by a wide range of mixtures that are quite rich near the core and lean towards the boundary of the spray, while the fluorescence intensity gets stronger downstream proving an increase in concentrations along the jet axis: this should not be a surprise since, as said before, during the dwell time between the two stages of ignition formaldehyde has the same trend of UHCs – and more broadly of fuel molecules – which is affected by the leaning-out action of the entraining wave in the proximity of the injector.

The single shots at the left also give an idea of the wavy structure of the jet outer contour while the averaged map on the right tends to smooth-out the effects of turbulence showing a rather clean profile; an instantaneous take of the same kind of image (but this time aligning the laser sheet along the penetration axis) returns however a different outcome, as visible in Figure 11.



Figure 11 – Formaldehyde fluorescence during the first-stage ignition for a late PCCI HD application

The instantaneous formaldehyde PLIF image is also taken with the laser oriented along the jet axis, as confirmed by the fact that now there is significant fluorescence signal nearby the injector too.

Both pictures illustrate significant structure along the boundaries after the EOI and, in the absence of combustion, it evolves even more distorting to the point that the typical conical shape is lost and the individual jets merge together.

It could be convenient to mention at this point that in light-duty engines operating under LTC conditions the situation is substantially the same, except that fuel can strike the piston bowl rim even before the first-stage ignition causing the splitting of the jet and high levels of swirl further complicate the flow dynamics.

On the basis of previous discussions, during the second-stage ignition, formaldehyde is consumed and significant concentrations of OH radicals appear, both species being easily detectable with optical diagnostics too: a major insight into this stage can then be gained experimentally.



Figure 12 – Simultaneous formaldehyde, PAH and OH fluorescence images for a late PCCI HD application

Figure 12 is a series of simultaneous single-shot images of formaldehyde, PAH and OH fluorescence for a late PCCI heavy-duty application, taken in horizontal planes at different heights below the firedeck; the timing spans from the first appearance of OH near the peak in the AHRR up to the late phases of heat release.

OH is false-colored green while both formaldehyde and PAH fluorescence is falsecolored red and the colorbar at the bottom still has to be read in the same way explained before, obviously referring to the latter species only.

As seen, OH first accumulates when the temperature is near halfway through the rapid rise during the second-stage ignition; the spatial overlap with formaldehyde is minimal consistently with the founding that one species is being replaced by the other; OH appears first downstream in the jet across mixtures of intermediate stoichiometry, as shown by the equivalence ratio contour map, accompanied by an intense chemiluminescence, much stronger than that captured during the first-stage ignition.
In contrast to the conventional combustion, the spatial distribution of OH is also broader, protruding well inside the plume rather than being confined to a thin shell around the jet periphery, to the point of quickly filling much of the cross-section in a few crank angle degrees.

Nevertheless, OH does not appear everywhere in the jet where formaldehyde is absent: these initial voids labeled "a" show indeed a PAH fluorescence shortly afterwards, as indicated by labels "b"; the contour map shows that PAHs appear in regions that are typically fuel-rich: these are going to grow much larger in the next few crank angle degrees when such precursors will eventually give rise to soot; furthermore, a shorter or negative ignition dwell would result in soot forming farther upstream in the jet while a longer positive dwell would push its formation even more downstream, at the interface with the piston bowl.

Interactions between neighboring jets are also a source of soot: these contribute to the creation of fuel-rich pockets where mixing is drastically inhibited because of the closed environment created by the vicinity of the combustion chamber walls.

As far as NO are concerned, direct probing with laser diagnostic is much more difficult especially considering that LTC combustions produce very little amounts of it; however, chemical kinetics considerations show that the rapid increase in temperature that occurs during the second-stage ignition is associated with high OH concentrations to which in turn the NO formation is to be ascribed: the OH distribution in the previous figure should therefore approximate the upcoming NO distribution too and analogous (to those proposed before for OH) observations shall then be made for a comparison with the conventional combustion.

Once again, despite the differences in the spatial structures of the fuel jets due to interactions with the ambient flow and the combustion chamber walls, the available data for light-duty engine agree with the description given for the heavy-duty ones.



Figure 13 – Simultaneous formaldehyde, PAH and OH fluorescence images in the latter part of a late PCCI cycle

Figure 13 above explores the late-cycle phases of combustion and offers a continued insight into the concepts already expressed with reference to the previous figure: pockets of PAHs and soot at first grow larger but then are largely oxidized.

The combustion luminosity, mostly due to soot incandescence, peaks near 19.0° ASI and gradually fades-out until being barely detectable at 40.0° ASI: the bright red signal due to combined formaldehyde/PAH PLIF and soot LII shows an increase in size in the leading portion of the jet (18 mm) accompanied by a consistent presence in the upstream regions too (12 mm and 7 mm) that progressively blackens giving way to the surrounding OH fluorescence only.

These facts give thus evidence of soot oxidation, whose rate depends on the temperature reduction and charge dilution induced by the EGR as well as injection timing and equivalence ratio. Although downstream mixtures exhibiting broad regions of OH indicate to have attained the second-stage ignition, the same is not true for the upstream ones: here (7 mm) significant formaldehyde persists throughout the sequence and, as the piston moves down during the expansion stroke, a weak signal starts to be detected in the downstream planes too whereas the initial conical shape of the jet loses its tightness, spreading in multiple directions.

This long persistence of formaldehyde is indicative of incomplete combustion and associated UHCs and CO: the most likely cause is overmixing to conditions that are too fuel-lean to achieve second-stage ignition before the volume expansion induced by the power stroke causes a fall of the ambient conditions to levels incompatible with combustion.

More generally, at a fixed time after EOI, the chemical species present at a given location provide an indication of the local equivalence ratio: formaldehyde marks the presence of fuel-lean mixtures, OH does the same with regions of intermediate stoichiometry while PAH with those whose equivalence ratio is in excess of two; furthermore, CO spikes up the exact moment in which formaldehyde and UHCs are being completely oxidized and the pool of OH is starting to build up: significant quantities of CO are then to be expected at the border between formaldehyde and OH, spatially denoting the temporal passage from one stage to the other of the heat release process. A similar behavior is also observed in light-duty engines.



Figure 14 – Equivalence ratio influence on the oxidation kinetics of the main pollutants for a late PCCI combustion

Figure 14 shows the influence of the equivalence ratio on the oxidation kinetics for a late PCCI: the quantity on the vertical axis is indeed measured at 90.0° ATDC when chemical reactions have essentially ceased.

Formaldehyde (not displayed) UHCs and CO all show an abrupt increase below a certain threshold: mixtures are so lean that temperatures do not reach sufficient levels to promote the breakdown of the hydrogen peroxides pool into the pool of OH necessary to start the second-stage ignition causing the products of the first-stage to remain unburned; the same effect is achieved if the injection timing is further retarded and as a consequence the limit before high emissions moves to the right, as indicated by the arrow. Superimposed on the plot is also a Gaussian distribution representative of the actual strength at which each elemental portion of the ignitable mixture is: if the timing is retarded, ignition delay and premixing increase, shifting the Gaussian distribution to the left and enlarging the quantity of mixture that will therefore exceed the threshold, resulting in incomplete combustion and high emissions. Excessively retarded late PCCI produces then two effects that reinforce each other: the mixture leans-out while the minimum threshold for complete oxidation simultaneously shifts towards richer values.



Figure 15 – Equivalence ratio influence on the oxidation kinetics of main pollutants for an early PCCI combustion

The above plot is analogous to the previous one but refers to an early PCCI.

In contrast to the late case, in which the cylinder contents cool down because of the expansion stroke hindering the onset of the second stage of ignition, in the early case the upward piston motion at the end of the compression stroke helps the temperature rise enough to propel the lean mixtures into the second-stage too, limiting the incidence of incomplete combustion and allowing a greater premixing before UHCs and CO become problematic. Higher temperatures mean unfortunately higher NOx but, as a consequence of previous discussions, higher OH too: the former and the latter are main responsible for CO oxidation.

This time, the transition to high UHCs and CO as the mixture becomes leaner is much more gradual and occurs at a leaner equivalence ratio for UHCs than for CO: approaching ϕ = 3, the mixtures are still able to achieve the second-stage ignition and UHCs are consequently low thanks to the extra heating provided by the favorable piston motion but reactions proceed so slowly that when the expansion stroke finally begins, most of the CO is still unburned.

Getting even leaner, reactions require so much time that compression heating can't help to prevent the whole second-stage from being delayed until the power stroke is well underway: significant formaldehyde and UHCs are then expected although the CO production is reduced since combustion practically freezes at the end of the first-stage, when formaldehyde has just start to be consumed but the rapid CO production, prior to the high OH levels necessary to ignite the second-stage, has not yet begun.

Another factor influencing the oxidation process and emissions of both early and late PCCI, is dilution:

- The minimum threshold equivalence ratio for complete oxidation of lean mixtures increases as dilution is increased because of the combustion temperatures reduction induced by the diluent gases: the mixture strength shall then be adjusted accordingly to avoid higher UHCs and CO emissions;
- Increasing dilution will cause the Gaussian distribution described before to flatten and shift to the right, as shown by the dashed line in the previous figure, reducing the quantity of mixture exceeding the threshold and the emissions stemming from its incomplete combustion;
- Increased dilution will extend the ignition delay, providing additional time for premixing and so moving the Gaussian distribution just mentioned to the left, towards higher UHCs and CO emissions.

It is important to notice that these three mechanisms counteract each other: the overall effect will depend on the details of the operating condition and the engine geometry; as a final remark, however, the previous figure suggests that CO emissions are more problematic than UHCs emissions under early PCCI conditions: the yield of CO is indeed considerably higher than that of UHCs for all but the leanest mixtures and the range of intermediate equivalence ratios providing a low CO yield sensibly narrower, with steep rises at its edges.



Figure 16 – Late-cycle ensemble-averaged LIF measurements of UHC and CO for an early PCCI at increasing load

Figure 16 shows late-cycle ensemble-averaged LIF measurements of UHCs and CO in a vertical plane through the center of the combustion chamber for early PCCI, at three different loads. Like NO, CO is a difficult species to probe with laser diagnostics and direct data about in-cylinder spatial distributions are only available for light-duty engines (like the example reported here) although no major conceptual differences are to be expected for heavy-duty applications.

In light-duty engines, some of the fuel injected might end up in the squish volume after a spray impingement on the piston bowl lip; the head portions of such mixtures are richer than the average because of the broadly discussed inversion of the equivalence ratio distribution along the jet axis at the EOI and so, when the piston approaches the TDC at the end of the compression stroke, mainly leaner mixtures are forced out of the squish volume into the bowl since the formers have penetrated much enough to approximately reach the cylinder walls; likewise, lean mixtures from the upstream regions of the jets that have instead made their way into the bowl after the split are to be found in the central upper region of the combustion chamber.

During the second-stage ignition, the rapid gas expansion within the bowl forces much of the lean near-rim mixtures back again into the squish volume thanks also to the reverse-squish itself created by the piston at the beginning of the power stroke, where the higher heat losses prevent them from attaining the second-stage ignition: the incomplete combustion thus resulting is responsible for the CO observed in such zones.

The yield however decreases with increasing load for three main reasons: the formation of over-lean mixtures is lessened because the longer injection durations and shorter ignition delays will limit the magnitude of the equivalence ratio distribution inversion after the EOI, granting in the upstream regions mixtures rich enough to move to the second-stage rapidly; the amount of fuel injected directly into the squish volume is greater implying that dangerously lean mixtures will be harder to obtain and thus less prevalent; higher pressures and higher temperatures will speed the reaction rates up.

The behavior observed throughout the cylinder for UHCs is similar to the one for CO within the squish volume just described; however, in the region below the injector, the trend is not monotonic and CO slightly increases at medium loads for the reasons exposed in the following.

At the lowest load, this zone is filled with UHCs due to the very lean mixtures produced by enhanced mixing at the EOI and the very rich ones resulting from the injector closure transient: in either case there is no ignition to the second-stage and UHCs are then high while CO remain low thanks to the excess oxygen in the area; at medium load, higher pressure, temperatures and equivalence ratios are able to ensure the achievement of the second-stage almost everywhere with reduced UHCs but increased CO: the time required for the burnout of the latter is still too long to avoid the ambient conditions collapse as the piston moves down; at the highest load, combustion proceeds so fast that most of the fuel attains complete oxidation with marginal levels of residual UHCs and CO stemming from the central upper region of the cylinder.

It has been mentioned before that very rich mixtures may arise from the injector closure transient: bad atomization/vaporization and droplets dribbling from the sac

through the nozzle holes as pressures drop during the expansion stroke, are factors playing a major influence on the UHC engine-out yield.

As a final remark, it is convenient to remember two additional sources of UHCs emissions: the fuel injected directly into the squish volume under early PCCI operation can penetrate sufficiently far to reach the top ring-land crevice and be pushed inside by the rising piston, that will release it not before combustion is already over; the fuel deposited on piston or liner surfaces suffers bad vaporization and causes oil-dilution as well.

To conclude this chapter, an extension of Dec's conceptual model for conventional diesel combustion is offered in Figure 17 once for both early and late PCCI heavy-duty applications: the spatial extent of some features and the timing of some events may change slightly with minor impact on the main phenomena; the discussion about light-duty engines is omitted for sake of simplicity since, as seen before multiple times, there are no substantial differences in the processes except for the complication induced by interactions with in-cylinder surfaces and macroscopic turbulent flows.



Figure 17 – Conceptual model for the early/late PCCI diesel combustion in HD applications

With reference to Figure 17 above, liquid and vapor fuel penetrate together until about 3.0° ASI, when the maximum liquid penetration length is established.

Shortly after the injection rate peaks (5.0° ASI) and then ramps-down: the deceleration of the liquid fuel exiting the nozzle increases local entrainment of ambient air creating a wave that propagates downstream at twice the jet penetration rate, as indicated by the vertical dotted line.

The fate of the liquid column depends on the duration of the EOI transient: after the head of the entrainment wave passes the quasi-steady liquid length near 7.0° ASI, the latter starts retracting back towards the injector as a result of the increased mixing and within 1.0° disappears completely.

Chemiluminescence emission from first-stage pre-ignition reactions (indicated by double-ended arrows) is detected by 6.0° ASI and the AHRR shows a distinct first-stage ignition event near 7.0° ASI when formaldehyde appears throughout much of the jet, from fuel-lean to fuel-rich mixtures.

After the EOI, the boundaries of the jet gradually widen from the more strictly conical shape established during injection and become wavy until, very late in the cycle, adjacent jets blend into each other forming a vague unstructured reaction zone.

For LTC conditions, fuel injection ends before the second-stage ignition so that a quasisteady lift-off length cannot build up; OH appears instead exactly at the end of the rapid rise in the AHRR and is found throughout much of the downstream portions of the jet where mixtures are of intermediate stoichiometry: it doesn't overlap spatially with the formaldehyde farther upstream and the few dark spots embedded into its distribution are credited to be fuel-rich pockets responsible for later PAHs and soot formation.

After initially growing in size, these soot zones eventually oxidize although some residuals could persist until the exhaust gas discharge depending on the fate of OH radicals.

Downstream fuel-rich and intermediate stoichiometry mixtures achieve second-stage ignition but the fuel-lean ones upstream may not before the piston starts its descending power motion: as a result, formaldehyde, UHCs and CO cannot attain complete oxidation and engine-out emissions increase.

Finally, around 20°/25° ATDC when cylinder pressures have returned to levels comparable to those present during the fuel injection event, liquid droplets can dribble from the injector holes creating locally fuel-rich regions that contribute modestly to UHCs and CO yields.

As a conclusion to this chapter it is important to remember that the conceptual model here provided applies only to the operating conditions specified at the very beginning of it: increasing the load (injection duration) or moving the SOI to more conventional timings would shorten the ignition dwell and produce a jet more similar to the conventional one, even while achieving a LTC regime thanks to high EGR rates.

4. Diesel engine-out pollutant species

4.1. NO_X

The acronym NOx includes both the species NO and NO₂.

The fundamental NO formation mechanism, also known as the *thermal* mechanism, takes place following the sequence of reactions elaborated by Zeldovich:

 $O + N_2 \leftrightarrow N + NO$ $O_2 + N \leftrightarrow O + NO$ $N + OH \rightarrow H + NO$

where the free oxygen atom in the first reaction comes from the preliminary dissociation (omitted) of an oxygen molecule. The first reaction of the set requires a very high activation energy and is therefore very sensitive to the temperature, which shall be comprised in a range between 2000 K and 5000 K: such values can only be reached on the diffusive flame, in case of conventional combustion, and the ambient conditions collapse at the start of the expansion stroke is strong enough to freeze the reactions causing lots of engine-out NO, that is contemporarily decomposed according to the inverse reactions but at a rate up to 5 times slower than that of the direct ones, thus accumulating as the combustion event proceeds to the end.



Figure 1 – NO and NO_2 concentrations in SI and CI engine-out exhaust

Figure 1 shows how, for SI engines, NO scale is two orders of magnitude larger than that of NO_2 which means that, for such engines, NOx is almost entirely composed of NO leaving to NO_2 a share of less than 1%; instead, for CI engines, the latter can reach up to 30% implying that a relevant part of the total NOx is made up of NO_2 .

As far as NO_2 is concerned, the main formation mechanisms are two: *thermal* and *prompt* and both take place in a very similar manner both in SI and in CI engines. The following reactions illustrate the situation for CI engines: L'NO che si forma nella zona della fiamma può venire infatti rapidamente convertito in NO₂:

 $NO + HO_2 \leftrightarrow NO_2 + OH$

In seguito dovrebbe avere luogo la riconversione da NO₂ a NO:

 $NO_2 + O \leftrightarrow NO + O_2$

The second reaction comes after the first one, while in SI engines they proceed together resulting in very little net NO₂ at engine out as shown by the previous graphs, right when the ambient conditions start to become unfavorable for its advancement that eventually stops causing a consistent NO₂ yield at engine out; emissions increase at low loads because of the flame quenching caused by the large amount of fresh air in the chamber and at low speeds due to the longer time available to bring the formation reaction to completion before the expansion and the consequent temperature fall. Obviously, if it is not of interest knowing the individual shares of NO and NO₂ inside the NOx, it is sufficient not to include the two equations above in the model for a predictive simulation since, being NO₂ formed starting from NO, computing NO is equivalent to computing all the NOx already; however, some ATS for CI engines require this information to maximize their conversion efficiency.



Figure 2 – NO time evolution throughout a conventional diesel combustion event

From Figure 2 above, it is possible to see the freezing phenomenon causing the NO concentration to stay high towards the later phases of the combustion event but also an initial value different from 0, to be ascribed mostly to the presence of recirculated exhaust gases (EGR).

The factors influencing the NOx formation are essentially two: oxygen concentration and combustion temperatures but, in the case of CI engines, these reduce to one only i.e. temperatures, being mixture strengths usually characterized by air in great excess; therefore, compression and heat release become crucial since even if NO is formed almost exclusively on the diffusive flame, the conditions at that time are largely determined by what happens immediately before: the longer the ignition delay, the more intense the premixed-burn, the higher the temperatures and the greater the NOx production.



Delaying the SOI, flame temperatures go down monotonically but, as shown in Figure 3, NOx exhibits some "bumps" contrasting with the strong temperature dependence seen before: on the rich side of the diffusion flame, incandescent soot particles behave like *black bodies* and thus radiate heat outside the plume; on the lean side of the flame, the space is almost entirely filled with the extra air that hasn't participated to combustion which, being transparent to such energy flux, allows it to proceed towards the chamber walls where it is displaced by the cooling system and wasted in the external environment.

Experimental evidences confirm that during this process the soot undergoes minor temperature changes implying that some heat from the diffusive flame shall move inside the plume to compensate for the phenomenon just described by restoring the radiating losses of the soot particles.

If the SOI is very postponed, temperatures start being so low that soot isn't even produced which means that the heat from the diffusive flame has now no other choice than radiating towards the free air surrounding the jet: oxygen abundance and rising temperatures boost then NOx formation leading to the little bumps displayed in the chart.

Generally, they occur at timings unreachable under conventional conditions but very typical of late-PCCI strategies.

Other techniques to reduce NOx always aim at making the transition from premixed to diffusive burn as fast as possible: injection rate shaping and pilot injections represent the most common solutions to reduce accumulated fuel mass and ignition delay.

Nevertheless, the most effective way still involves lowering combustion temperatures by admitting EGR in the cylinder, acting according to three different mechanisms:

 Dilution effect: some of the oxygen on the diffusive flame is replaced by EGR that, being made of a mix of multiple species, causes a thickening of the flame front and a reduction of the temperature having the heat of combustion to be distributed now to a larger number of molecules;

- Thermal effect: the mixture average specific heat increases because EGR mostly comprises CO₂ and H₂O whose specific heats are bigger than that of the air;
- Chemical effect: during combustion, temperature are so high to cause the dissociation of some of the CO_2 and H_2O inducted with the use of EGR but, being the phenomenon endothermic, this in turn translates into a net temperature reduction.

The prevalent mechanism is anyway the first one: the thermal effect is negligible because even at recirculation rates higher than 50% the mass share of CO_2 and H_2O inside the charge is lower than 6% resulting in the average specific heat increasing only by 0.5%.

Such high rates are however applicable at very low loads only when the excess air is so large that exhaust gases still contain a relevant amount of oxygen that adds up with the one in the fresh air to allow a regular combustion; conversely, at high loads, charge strength is enriched to increase the power output and a further rise of the equivalent ratio might lead beyond the *smoke limit* causing massive soot emissions.

Also, deploying EGR doesn't produce any effect on the AHRR and more broadly on the in-cylinder pressure track and is able to reduce the negative impact on the volumetric efficiency, coming from subtracting space to the fresh air, if properly cooled before induction.

Finally, the NOx-soot trade-off worsens at high rates because EGR offers more sites for particles formation and agglomeration while the insoluble carbonaceous fraction increases together with the risk for the ATS to clog up, being these unable to burn it and thus requiring periodical mechanical regeneration from ashes.

4.2. HC/CO

Owing to the different chemical composition of Diesel fuel with respect to gasoline, HC produced by CI engines will be heavier than those emitted by SI ones and this might be an issue in testbed applications: the exhaust gas to be analyzed passes through pipes up to 10 m long before reaching the corresponding measuring instrument which is usually stacked on others outside the laboratory in a different room and it might happen that during this travel, if the temperature falls below 100°C, some water condenses dragging in the heaviest molecules of HC that won't thus be detected by the analyzer.

The measure will consequently be an underestimation and deposits will instead build up along the lines whose evaporation, during following experiments at higher loads for example, will again spoil the measure this time by overestimation.

To prevent contamination and erratic measurements, ducts and analyzer are permanently heated to 190°C while pre-filters located within 1.5 m of the exhaust pipe sampling point block PM particles, which may damage the equipment.

The two main formation mechanisms are *undermixing* and *overmixing*; as shown in Figure 4, at the tip of the jet along the spray axis, the trend ramps down to zero quite steeply but there still shall be a portion of mixture so lean to be below the flammability limit and thus eligible as the main source of HC in CI engines.



Figure 4 – Spatial evolution of the mixture' strength prior to autoignition

As the start of the premixed-burn is postponed there is more time for penetration and mixing as demonstrated by distributions flattening on the horizontal axis implying that increasing portions of the mixture shift toward leaner ratios with major risks for the fuel at the boundary of the jet directly in contact with the air cushion to overmix, resulting in a poor combustion: PCCI conditions featuring extremely extended ignition delays are then naturally prone to produce lots of HC because of overmixing.

What happens instead during the transient the injector closes is to be ascribed to the phenomenon of undermixing: spray quality worsens as the needle completes its downward stroke and the liquid fuel trapped in the sac (a little capacity upstream the nozzle holes is required to guarantee maximum uniformity of the 8/9 jets usually employed) eventually drips in the combustion chamber drew by the piston on its way to the BDC when the pressure approaches values similar to those during induction; obviously the situation becomes critical at high loads when the quantity to be injected increases and the transients duration prolongs as well to allow the maximum needle lift, for maximum fuel flow rates.

Nonetheless, undermixing is also linked to wall-impingement: increasing the injection pressure turned out to affect only marginally the liquid penetration length as the expected increase in spray velocity simultaneously causes the fuel to entrain more air boosting mixing and allowing evaporation in a reduced amount of time thus compensating for the first effect and resulting in no net disadvantage.



Figure 5 – Piston bowl designs comparison for reduced HC/CO

Other ways to impact on the HC emissions consist in optimizing the fluid-dynamics of charge macro-motions inside the cylinder. One of these is the *squish*, strictly related to the geometry of the bowl and distinguished between *direct* when the piston is at the final stages of the compression stroke and *reverse* when it starts the expansion one.

As far as HC is concerned, the latter is the one to focus on: it takes what remains in the bowl at the end of combustion and brings it above the piston top and in the proximities of the cylinder head and walls where a consistent quantity of almost pure air might help in the burnout of possible products of partial combustion.

Great care must be taken to ensure that such a displacement doesn't take place while combustion is still occurring as the lower temperature in the squish volume would immediately quench the flame.

As shown in Figure 5, a bowl of the *reentrant* type is the one more suited to accomplish the task improving at the same time *swirl* and direct squish during the upward stroke: in particular, it is possible to notice that it is the most beneficial under retarded timings because the reentrant lip delays the reverse squish to avoid that the burning charge is brought away from the combustion chamber into cold zones.

4.3. PM

PM, as visible in Figure 6, is an aerosol composed of an undefined mix of chemical species strongly dependent on the sampling and measuring technique: for this reason it is necessary to define an *operational definition* usually contained in standards.



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In Figure 7, circle dimensions indicate the quantity of PM emitted engine-out, green portions indicate the SOF while the white ones the solid components and sulfates.



Figure 7 – PM engine-out yields and SOF shares over the engine map

Ashes come from non-combustible substances: lube oil additives, engine and exhaust line wear, catalyzers like Cerium in solution within the fuel to help the DPF regeneration, etc.; SOF derive instead more from the lubricant than from the fuel while sulfates are agglomerates of water and sulfuric acid resulting from *heteromolecular nucleation* that, being a low temperature mechanism, is probably taking place somewhere in the exhaust system before the discharge into the external environment.



Figure 8 shows the risks of *sample alteration* if the temperature isn't kept constant until the measurement is complete. It is also possible to see that in order to burn soot, it is required the permanence in an oxidizing environment at above 650°C for more than 30 minutes, which could suggest how to perform a filter regeneration, even if there is no way to remove ashes that will end up clogging the filter and asking for a mechanical regeneration.

Standards define as PM: anything remaining trapped on a filter kept at the fixed temperature of 52°C.



Figure 9 – PM density alteration as a function of the sample exposure to the ambient air

When the exhaust stream is eventually disposed into the atmosphere, dilution starts to occur; as far as PM is concerned, Figure 9 shows that up to DR = 5 the cooling caused by the external air drives the condensation of organic species into SOF while beyond this threshold the partial pressure of unburned hydrocarbons lowers to the point to let part of the SOF evaporate, implying that besides all the issues already discussed, PM measurement is also influenced by the dilution with external air that shall then be reproduced with great accuracy: the common practice establishes a range for the DR spanning from 3 to 20.



Figure 10 – PM size distribution and main features of the nuclei/accumulation mode particles

Here is a brief discussion of the PM mass-size distribution shown in the upper portion of Figure 10:

- Nuclei Mode: these particles, for the most part lacking of the primary carbon fraction as shown in the scheme in the lower portion of the same figure, form when temperatures are particularly low i.e. in the exhaust line or in the external environment – namely after the DPF, which provides the main constituents after its own regeneration – or even in the sampling/measuring apparatus itself: such particles are highly unstable and therefore very difficult to analyze;
- Accumulation Mode: the solid carbonaceous substrate marked in grey in the picture on the right tells that such particles are almost entirely produced during the combustion process with later deposition by gas adsorption or vapor condensation of the SOF;
- *Coarse Mode*: not related to the combustion process but to the engine/exhaust system wear.

Starting from EURO 6, it has been decided to regulate in addition to the mass (PM) also the number (PN) of particulate emissions: this is mainly due to the fact that although the mass share of Nuclei Mode particles is almost negligible, they still represent the vast majority of the total and are responsible for the most powerful effects on human health being nanoparticles able to penetrate the deepest in the respiratory system.



Figure 11 – HACA: Hydrogen Abstraction C₂H₂ Addition direct mechanism of PAH growth into soot particles

Figure 11 refers to the HACA – Hydrogen Abstraction C_2H_2 Addition – direct (faster) mechanism of PAH growth into soot. Acetylene C_2H_2 is the simplest alkyne and the triple – quite unstable – bound between its two carbon atoms makes it very prone to react: at the end of the cycle a benzene ring is added and the process fast forwards until soot agglomerates are born.

Nevertheless, a second *indirect* (slower) mechanism also exists which takes place on the diffusive flame rather than on the premixed one and is basically an HACA variant in which alkanes and alkenes too are involved in the process together with the alkynes – all aliphatic compounds can then participate – since temperatures are so high to be able to destroy any bound: this mechanism is the one responsible for the grains thickening at the boundary of the plume on the rich side of the diffusive flame.

Once soot particles start to accumulate as a result of the two mechanisms just described, collisions take on the task to build ever bigger structures upon which deposition of organic compounds occurs to give rise to particulate units developed to their final form.

As said earlier, EGR reduces NOx and increases soot but some boost might prove helpful in loosening the trade-off: the inlet charge density increase leaves indeed free space for additional EGR (to reduce NOx) or extra air (to lower soot); turbocharging also enhances EGR speed causing it to be admitted at higher temperatures because of less time available for heat exchanges implying reduced EGR density and leaner lambda ratios for better soot oxidation.

Post injections remain however the best way to cut soot and HC before the exhaust stream disposal in the atmosphere.

4.4. Noise

Combustion noise is evaluated as follows: an high-frequency in-cylinder pressure sensor, usually mounted in a hole drilled through the engine head or in place of the glow-plug, records a signal over time as the one in the upper portion of the following Figure 12 which is then processed by means of FFT to give a spectrum like that in the lower portion of the same figure.



Figure 12 – Combustion noise evaluation starting from the high-frequency in-cylinder pressure trace

To the final outcome of this operation is hence applied a series of filters:

- *CAV Filter*: representative of the acoustic dampening opposed by the engine structure/material to the sound waves propagation;
- A Filter: representative of the human ear sensitivity;
- *LP Filter*: to cut-out the resonance frequencies of the gas bulk vibrating inside the piston bowl like a drumhead.

The main contribution to the engine noise is given by the first derivative of the pressure signal, reaching its maximum values during the premixed-burn: once again, it is a matter of reducing the amount of fuel burning in premixed conditions by means of injection rate shaping, pilot injections and EGR.

In this regard, it may be interesting to evaluate the effects of varying the injection pressure: moving from 1000 bar to 2000 bar, for example, the fuel flow rate is increased by 40% as the injection velocity obeys the Bernoulli's principle and shall thus vary with the square root of the pressure drop experienced by the fuel during the discharge from the rail into the combustion chamber. However, if the engine point hasn't been changed, it is necessary to reduce the amount of time the injector stays open as the total quantity to be injected must be the same.



Figure 13 – Sauter Mean Diameter as a function of the fuel delivery pressure

As Figure 13 shows, at such high values of pressure, the *Sauter Mean Diameter* (SMD) – reference parameter for the evaporation process as the speed of the latter is inversely proportional to the magnitude of the former – dependency on the rail pressure is virtually flat implying that the reduction expected for it is totally negligible, as will be the effects on the mixing/vaporization time: the ignition delay turns out not to be affected at all and the timing of the switch from one combustion regime to the other remains the same, leaving the additional 40% of fuel enough margin to prepare and burn under premixed conditions, with great noise penalties. In logarithmic scale, 3 dB correspond to doubling the noise intensity in linear scale.

5. Exhaust gas analyzers

- NDIR: Non Dispersive Infra-Red Detector for CO and CO₂
- FID: Flame Ionization Detector for HC
- CLD: Chemi-Luminescence Detector for NOx
- Paramagnetic instruments for O₂

NDIR are in theory able to serve all measuring purposes but their use as all-rounders would require a series of technical expedients, to ensure the adequate experimental consistency, hard to realize.

Furthermore, legislation prescribes the use of specific equipment for a given pollutant as a prerequisite just to guarantee the homologation validity under a legal point of view.

The entire set of analyzers is placed inside a cabinet with modular design manufactured by AVL under the name *AMA i60 Emission Bench* which is located outside the testbed to allow for safe manual operation by the personnel thanks to a control computer embedded on the front end.



Figure 1 – AVL AMA i60 Emission Bench

5.1. NDIR

A gas exposed to infrared radiation absorbs only certain wavelengths according to the characteristics of its atomic structure.



Figure 2 – Typical functional layout of an NDIR

The one in Figure 2 is the sketch of a common layout for NDIR applications: chamber B is filled with N_2 or any other inert gas transparent to the infrared radiation while the exhaust stream is routed to chamber A and, coherently to what has just been said, the pollutant to be investigated contained in it starts absorbing the wavelengths typical of its species; chambers C and D are filled with the same gas comprising a precise concentration of the pollutant under testing.

At this point it should be clear that while the entire energy flux is transmitted from the source to D, some part of it is subtracted instead along the travel to C by the pollutant in A whose concentration is to be determined: less energy means lower pressure increase resulting in the flexible membrane M deforming and protruding inside C; being this membrane one of the two plates of a capacitor armature, its stroke causes a capacity change that translated into a voltage or current will finally provide the desired outcome.

$$E_a = E_i \left(1 - e^{-kcL} \right)$$

k = coefficiente di assorbimento del gas;

- c = concentrazione della specie di interesse;
- L =lunghezza della cella.

Se il prodotto kcL è molto inferiore rispetto all'unità:

$$kc_{\pm}L \ll 1 \rightarrow E_a = E_i (1 - e^{-kcL}) \approx E_i kcL$$

Figure 3 – Operating principle' fundamental relation linearization

The energy absorbed by the pollutant in A is a function of the one emitted by the infrared source and has the expression shown in the upper portion of Figure 3; if the product kcL is much lower than unity, it is possible to linearize as shown in the lower portion of the same figure: in order to ensure that the condition is met, high expected concentrations call for very short cells while longer ones should be employed otherwise to increase the sensitivity of the instrument, even if it costs linearity and loss of measure reactivity during engine transients.

To avoid any compromise, *two* NDIR are usually devoted to a *single* pollutant if its concentrations vary significantly depending on the engine point as in the case of CO and CO_2 ; actually, CO_2 is always quite prominent at the exhaust but it might happen, as in the case of PCCI, to run the engine with such high EGR rates to be necessary to monitor the quantity of CO_2 in the intake manifold to prevent misfire, for example.

The software managing the device is anyway able to acquire and store up to four different ranges, each extending the previous one.

To operate properly an NDIR there are routines to follow at the startup and shutdown. For example, before the start of a working shift: an inert gas synthetically brought to top-tier levels of purity is circulated as a *zero gas* through chamber A in order to locate the 0 on the measurement interval (it is clear that it must be the same gas used for chamber $B - N_2$ with reference to the previous example); then the procedure is repeated with a *span gas* containing the maximum pollutant-to-test concentration reasonably expectable in solution with N₂ in order to adjust the full-scale range of the instrument.



Figure 4 – Progressive instrument calibration by means of increasing strength span gases

At this point, however, further actions are required before starting the measurement campaign: even though it is possible to linearize the relation between energy absorbed and energy emitted as previously seen, if the output of the system is given in terms of voltage, it shall be remembered that in a capacitor the relation between voltage and capacitance is still not linear; this means that a few intermediate checkpoints have to be acquired to translate correctly voltages into concentrations over the whole range of values.

The final characteristic curve will consist of a set of 10 (at least) linear segments obtained by employing a pollutant/ N_2 mixture of ever increasing concentrations of the former so as to begin with a zero gas and finish with a span gas, as shown in Figure 4.

Only at this point the instrument is *calibrated* and ready for use. It must be said though that intermediate mixtures maintenance is scheduled once a year while zero and span procedures must be performed on a daily basis.

As a matter of fact, the actual NDIR construction scheme features, in addition to the functional components seen at the beginning of this paragraph, a *chopper* and a *filter*.

These add-ons are required because CO and CO₂, for example, have absorption spectra partially superimposing and since both of them are simultaneously present in abundance, each of the two NDIR will largely overestimate its reading by erroneously identifying some CO as CO₂ and vice versa: to solve this issue, a filter *cell* containing CO or CO₂, depending on which of the two is responsible for the parasitic absorption to eliminate, is placed between chambers A and C; an *optical* filter can alternatively be used in place of the cell to absorb the wavelengths of the parasitic species.

The *chopper* is instead a disk with circumferential gaps rotating at a frequency of 3-4 Hz that, clearly, chops the infrared radiation in order to have a pulsating voltage at the instrument output to ensure a sufficiently high signal-to-noise ratio (SNR) after its amplification, demanded by the microscopic movement of the membrane M.

Obviously, the walls of chambers C and D are kept at the same constant temperature to avoid again mistakenly ascribe effects driving the measure (gas pressure variations) to parasitic causes (temperature variations in the external environment).

As a final remark, it is essential to stress the fact that NDIR do not tolerate water in any form: vapor's absorption band is once more overlapping with those of CO and CO_2 and its condensation could cause drops deposition on the quartz glass of chamber A jeopardizing the whole operating principle; upstream of the instrument are then placed refrigerators inside which exhaust gases are dried-out by cooling down to allow for water vapor condensation.





Figure 5 – Typical functional layout of a FID



The combustion of carbon compounds produces ions in a number proportional to the total number of hydrocarbon molecules (THC) present in the sample, on the base of which is anyway impossible to go back to the individual species contributing to it, unless further equipment is employed, as it will be shown later.

As revealed by the scheme above, the flame to burnout any HC present in the exhaust stream is sustained using hydrogen as combustible to ensure that the instrument doesn't produce ions by itself, spoiling a priori the measurements; beside the flame, two oppositely charged poles collect the ions and let them flow through an electric circuit producing a current whose magnitude is proportional to the HC concentration in the exhaust.

Similarly, the air used to burn the hydrogen and light the flame up is obtained synthetically to avoid any carbon contamination and is a pure O_2/N_2 mix in the proportions 20%/80% also known as *chromatographic air*.



In the expression reported in Figure 6, $n[C_nH_m]$ is the total number of carbon atoms in the sampling volume estimated by counting the ions, Q is the exhaust gas flow rate varying according to the engine point and α is constant for a given HC species: in order to have the output current depending on $n[C_nH_m]$ only and thus make the instrument reading always *direct*, it is necessary to have both Q and α constant regardless of the engine point and the compound chemical composition, respectively.

Molecola	Risposta FID (normalizzata rispetto al propano)		
Alcani	0.97-1.05		
Aromatici	0.97-1,12		
Alchini	0.99-1.0		
Alcheni	1.07		
Ossigeno (negli alcoli)	0.23-0.68		

Table	1-	αν	alues	for	different	hydrocarbon	compounds
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Luckily, Table 1 shows that α is practically constant for any compound with the exception of the *oxygenated* ones: the reason is that, in this case, there is a bond O-C whose strength is high enough to prevent that specific carbon atom from releasing ions; the problem can be solved by using instead of pure H₂ as a flame combustible, a mix with helium.

The linear relation above suffers however the following limitations, causing some of the ions to slip away without being intercepted by the electrodes:

- Water vapor favors the recombination of free electric charges with opposite sign: to contrast the phenomenon, FID housing and feeding lines are kept at 190°C – as seen for NDIR – to avoid water condensation;
- Turbulence created by the flame itself might push some charges away from the detection point;
- High charge density hampering the electric field between the electrodes;
- Charges multiplication in presence of a very strong electric field.

All in all, if the system is designed properly, the above facts should have minor influence on the instrument operation resulting in a current linearly varying with the total HC (THC) concentration in the exhaust; in this case, *zero* and *span* will then be enough for a complete calibration: the zero gas is given by chromatographic air while as a span gas is usually exploited a mixture of propane (C_3H_8) and N_2 whose full-scale value expressed in terms of C_3 can be converted into C_1 , as required by standards being the use of propane not mandatory, simply by multiplying by 3.

To establish the chemical composition of compounds contributing to ions production it is necessary to have a *mass spectrometer* but if it is enough, as prescribed by the most recent legislation, to distinguish between *methane* and *non-methane* species only, it is sufficient to rearrange the layout as displayed in Figure 7 by providing a second FID and a supplementary catalytic oven – the *cutter* – upstream of it with the function to completely burn all the HC except methane (CH₄): the difference between the outcomes of the two branches will finally give the *non-methane HC* (NMHC) share.



Figure 7 – Functional layout modification for MHC/NMHC simultaneous measurements

5.3. CLD

Chemiluminescence is due to photons emission during a chemical reaction.

When NO reacts with ozone (O_3) a molecule of NO₂ electronically excited is formed emitting light as it returns to a non-excited state.

The sequence of events is the described by the following set of reactions where the asterisk denotes a state of electrical excitation:

- 1. $NO + O_3 \rightarrow NO_2 + O_2$ 2. $NO + O_3 \rightarrow NO_2^* + O_2$
- 3. $NO_2^* \rightarrow NO_2 + photon \quad (\lambda = 0.6 3 \mu m)$
- 4. $NO_2^* + M \rightarrow NO_2 + M^*$

As the temperature increases, reaction 2 becomes much more frequent than reaction 1 while there is always the risk that reaction 4 happens in place of reaction 3; in this case, the electric charge is given to another molecule M instead of being emitted as light making impossible for the instrument to detect it as when M finally returns to a non-excited state, the photon is released on a wavelength specific of the M's chemical composition and not of the NO₂'s one, leading the device not to count the photon and thus underestimate the NO concentration in the exhaust.

The operating principle is indeed very simple: the CLD counts the number of photons at a certain wavelength in order to estimate the NO concentration in the sampling volume by making use of a very unstable molecule like NO_2 as an intermediary to speed up the electric charge absorption/release process and guarantee then a real time measurement.

In order to do so, the operating temperature must be fixed to regulate the amount of NO reacting according to 1 and 2 and the vacuum made inside the reactor to eliminate any parasitic M and reduce therefore the incidence of reaction 4, keeping however in mind that only some of the total NO is going to react to form NO_2 but also that, at a fixed temperature, it is known a priori which is the percentage prone to react; in particular, this is equal to about 10% at 25°C and increases by 1%/°C.

In Figure 8 is sketched the basic layout of a CLD analyzer.



Figure 8 – Typical functional layout of a CLD

The intensity of emitted light is directly proportional to the NO₂ concentration which is in turn proportional to that of the species that react together to form it, namely:

$$\mathbf{I} \propto \frac{[\mathrm{NO}]\mathbf{O}_3]}{\mathrm{M}}$$

Obviously, the concentration of O_3 – obtained by exposing O_2 to an UV lamp – must be kept constant and in large excess of the NO's one in order to make the light intensity dependent on the latter only.

Nevertheless, Diesel engines require to know about the NO₂ concentration too in order to maximize the efficiency of their ATS; in this regard, the *catalytic converter* depicted in the previous scheme is employed: it is a sort of oven kept at constant temperature in charge to operate the inverse reaction transforming NO₂ into NO.

In this way, part of the total NO detected by the CLD stems from NO₂ but unfortunately the catalytic converter is not a particle counter and thus, if the information about NOx is not sufficient, knowing the magnitude of NO₂ share requires placing a second CLD downstream of it and modifying the overall architecture as shown by Figure 9:



Figure 9 – Functional layout modification for NO/NO2 simultaneous measurements

The desired outcome is obtained by subtraction as in the upper branch the NO_2 already present in the exhaust stream is and remains in an unexcited state thus slipping away undisturbed without being intercepted by the instrument while in the lower one everything is turned first into NO and then into NO_2^* .

If it is not mandatory to distinguish between the individual contributions of NO and NO_2 to NOx in real time and simultaneously, the original layout can still be used by alternately switching from the upper (for NO measurement) to the lower (for NOx measurement) branch and vice versa.

As a final remark, CLD exhibit very linear characteristics and then, to calibrate them, it is enough to perform the zero and span procedures using N_2 for the former and a mix NOx/N₂ for the latter.

5.4. Paramagnetic instruments

 O_2 shows paramagnetic behavior because two of the electrons in the last orbital are not paired; as a consequence, when surrounded by an external magnetic field, its molecules develop their own field aligned with the inducing one: they are attracted by the external magnetic field.

There exist two different typologies: *thermo-magnetic* and *magneto-dynamic* analyzers; the second ones are the most diffused and the operating principle can be described making reference to Figure 10:



Figure 10 – Typical functional layout of a magneto-dynamic paramagnetic analyzer

The rocker arm features two diamagnetic spheres at its extremities and is suspended by a quartz filament whose torsion contrasts the repulsion developed by the interaction between the spheres and the two pole pieces generating the external magnetic field, until an equilibrium position is reached; when the exhaust stream is let flow through the instrument, the free O_2 contained in it undergoes a magnetization interfering with the pole's field and causing the rocker arm to reach a new equilibrium position: the angle displaced can be easily linked to the amount of oxygen in the exhaust.

5.5. PM analyzers

There exist two different ways to proceed with PM analysis:

- Collecting Techniques (CT)
- In-Situ Techniques (IST)

CT require a filter through which the exhaust gases are let flow building deposits that are later analyzed with methods like *gravimetry* etc. The problem with such practices lies in the fact that it is very difficult to control the physical/chemical evolution of the sample until the measurement is done which, as a consequence, is not performed in real time; conversely, IST allow for instantaneous readings but are not *direct* methods implying that the outcome is an *estimate* based on the *smokiness* or *opacity* of the exhaust stream.

Once the exhaust gases have been cooled down to 52°C by means of dilution with external air according to the DR values previously seen, CT prescribe they are routed into a couple of glass-fiber filters coated in Teflon to keep the sample temperature

unchanged until it is weighted on a ultra-micro analytic balance located in a temperature/humidity conditioned atmosphere: as said, it might happen during the time lapse between collection and analysis that deposits instability causes some species to condensate or vaporize, affecting the mass to be weighted.

Anyway, IST are realized by means of a *smokemeter* or an *opacimeter*.

5.5.1. Opacimeter

As seen in Figure 11, it consists of a light source and a photosensor coaxial to it: when the exhaust stream is let pass in between, the luminous intensity detected by the sensor is reduced due to the solid particles obstructing the optical path; *opacity* is defined indeed as the percentage of the emitted light intensity lost when crossing the exhaust *plume*, namely the bulk of gases.



Figure 11 – Typical functional layout of an opacimeter

The zero procedure is done using chromatographic air while for the span it is enough to switch off the light source and let the photocell measure in dark conditions; from time to time, it is also necessary to clean up the protective quartz surfaces screening the two ends of the optical path by pumping compressed air – the process is called *purge*.

5.5.2. Smokemeter

The description of the working principle is pretty simple: a light of intensity I_0 is directed against a paper filter that, depending on the level of dirt accumulated after a brief exposure to the exhaust stream, will reflect back a light of reduced intensity I: the fraction of light intensity lost in the process defines the *smoke number* (SN).



As shown by Figure 12, the piston slides back drawing in a defined volume of exhaust gases through a filter tape which is then moved by a robotized mechanism to proceed with the measurement, as previously described.

In this regard, it is important to notice that modern engines' ability to produce low emissions would cause the filter to remain virtually white at the end of the piston stroke making the measure not significant, lying this within the uncertainty range of the instrument: as a matter of fact, it takes many cycles for the paper to reach an appreciable level of fouling and since the engine point is also affecting the smoke loading of the exhaust, it is mandatory to provide the system with the ability to keep track of the total volume sampled for a single measurement.



Figure 13 – CT vs. IST analysis capabilities

As a concluding remark, Figure 13 shows why the opacimeter is the least preferred solution: it is characterized by a relevant uncertainty when it comes to assess the SOF component of particulate; furthermore, their light source usually runs on a wavelength sensitive to the presence of NOx whose light absorption misleads the photosensor, that ascribes by default the loss of signal entirely to the particulate, causing the instrument to overestimate.

6. The engine

The engine, visible in Figure 1, has been derived from the conventional FPT F1C EURO VI series production model, whose main characteristics are summarized in Table 1, by applying some important modifications to run in PCCI combustion mode up to a load of 9 bar.



Figure 1 – FPT F1C PCCI prototype derived engine

The engine delivers a rated power of 129 kW at a rated speed of 3500 rpm with a maximum torque of 430 Nm over the speed range 1600 - 2800 rpm.

Engine type	FPT F1C Euro VI diesel
Number of cylinders	4
Displacement	2998 cm ³
Bore / stroke	95.8 mm / 104 mm
Rod length	160 mm
Compression ratio	17.5:1
V alves per cylinder	4
Turbocharger	Single-stage variable geometry turbine type
Fuel injection system	Common rail injection system
Injector static flowrate	990 cm ³ in 30 s at 100 bar upstream pressure
Injector cone angle	139.8°

Table 1 – Salient features of the FPT F1C EURO VI series production donor engine

 The compression ratio was reduced from 17.5:1 to 14.6:1 with the aim of lowering the in-cylinder pressure and temperature conditions before the combustion event and the resulting flame peak temperature, which are the main parameters responsible for engine-out NOx emissions;

- A modified piston bowl was designed to be more suitable for fuel spray-air mixing for advanced injection timings;
- New injectors were installed with reduced cone angles (from 139.8° to 130° as seen in Figure 2) and static flowrate (from 990 cm³ to 750 cm³ in 30 s at 100 bar upstream fuel pressure) to reduce the risk of cylinder wall impingement, as the reduced compression ratio and the generally advanced injection timings tend to increase the liquid penetration of the fuel spray inside the combustion chamber;



Figure 2 – Piston bowl design and spray angle modifications

- The original EGR cooler (nominal maximum thermal power of 6 kW) was replaced by a larger one, from a production 11-liter displacement engine, to allow a greater cooling power (nominal maximum thermal power of 32 kW) for the high EGR mass flow rates that are usually involved in PCCI operations. Moreover, the EGR cooling circuit in the F1C PCCI was separated completely from the engine cooling loop so as to provide a precise control of the exhaust gas temperature at the EGR cooler outlet;
- The EGR poppet valve installed on the conventional hardware was replaced by a throttle valve, with the aim of increasing the maximum cross-section area and the resulting EGR mass flowrate toward the intake manifold;
- A smaller turbo-group was specifically designed and implemented for higher boost levels at lower engine loads;
- The ATS of the F1C PCCI engine was only endowed with a DOC but the presence of systems usually installed on the vehicle, including a DPF and a SCR, was simulated by means of a throttle valve placed at the cold end of the exhaust line causing a constant backpressure.

7. The engine control equipment at the bench

The ECU used for calibration purposes is not the same as that mounted on a series production vehicle but is directly derived from it by adding in the same housing a small electronic board called ETK (EmulatorTastKopf) which stands for "memory emulator", as shown in Figure 1.

The ETK allows the ECU to focus on its usual tasks by taking care of the communication with the testbed in order to avoid an extra overhead of computational power that would slow down the data transmission rate especially at high engine speeds, when the burden of information to transfer in the unit time becomes critical.



Figure 1 – ETK chip parallel installation on the main ECU board

Using an ETK/XETK development ECU, series-production software can be calibrated with great ease and subsequently verified with the production ECU without the need to change drivers in the platform software.

The most common layout is the *parallel* ETK: in this case, the ECU FLASH memory is replaced by a Dual-Ported RAM to reduce the time required for reading/writing operations and have no restrictions on the device lifetime in terms of number of writing actions allowed, which is simultaneously accessible by both the ECU, whose control actions require the reading of calibration maps, and the ETK, that is in charge to update the Working Page (WP) whenever the bench operator/automation system decides to modify the current calibration; the WP is always paired with a Reference Page (RP) which is a read-only file containing a safe calibration to switch immediately to in case of emergency.

However, the ETK is still provided with a dedicated power supply and a FLASH memory from which the ECU RAM, that being a volatile memory loses its content as soon as the power supply is interrupted when the ignition key is turned off, can download a coldstart calibration to warm up the engine safely even if the connection to the PC running INCA is missing.
The ECU speaks the binary code only and thus the ETK shall receive two files from INCA: a .hex containing the calibration itself in binary format translated by the software starting from datasets in physical format and a .a2l known as the *project description file* that helps the ECU to interpret the meaning and store correctly each 0 and 1 of the .hex file.

Also, there exist two types of ETK: simple ones and XETK; the second ones are more expensive because, not requiring further hardware (ETAS ES59x series interface modules visible in Figure 2) to communicate with the PC hosting INCA, are of course more complex from a construction point of view due to the need to incorporate an Ethernet port handling the proper protocol (XCP) for fast and reliable data exchange. On the other hand, XETK eliminate the expenditure related to the ECU interface hardware.



Figure 2 – ETAS main interface and networking modules

Other ETAS modules include: the ES600.2 (again in Figure 2), that makes it possible to interconnect and provide with power (sequentially during the startup to avoid load surges) a multitude of other ETAS compact units and to bundle and synchronize the data of the connected modules before the transfer towards the INCA PC or an additional Rapid Prototyping (RP) device upstream; the ES636 dual-channel Lambda module (Figure 3) supporting Bosch's LSU 5.1 broadband (it measures lambda values in the range between 0.6 and 16) oxygen sensor, with a TEDS code inside that allows the module to recognize the sensor type preventing improper operation and automatically detecting malfunctions.

The ES636 is also equipped with a high-performance sensor heater to enable the rapidacting LSU ADV sensor attain its operating temperature within 5 seconds by means of a heating current of up to 5 A in the range between 0 V and 18 V; to protect the measuring sensor, the sensor heater's operation may be allowed to continue beyond the point at which the measuring units within the modules have been shut off and can be triggered independently of the measurement function by an external signal (typically the ignition key turn-on) before the startup, to ensure engine controlled operation during the early stages of warmup.

Furthermore, the ES636 is capable of measuring atmospheric and exhaust gas pressures by means of dedicated sensors embedded in the housing in order to automatically compensate for the influence they have on the lambda value measurements: a pressure variation of 400 mbar in the engine's exhaust pipe causes a 10 percent deviation of the sensor pumping current from which the lambda value is derived.



Figure 3 – ETAS dual-channel lambda module

The operating principle is pretty simple and is based on the oxygen ion conductance of zirconia at high temperatures exploitable by means of the layout reported in Figure 4: in the case of wideband sensors, the oxygen concentration of the test gas inside the Nernst cell is set to $\lambda = 1$; this is accomplished by pumping – injecting or extracting – oxygen ions into or from the test gas thus making the size and direction of such a pumping current easily relatable to the oxygen content of the exhaust stream.

This number is of paramount importance for direct injection Diesel applications: such engines are operated with a lean combustion across a wide load range, with the NOx content of emissions steeply rising in conjunction with increasing temperatures; EGR is a means of reducing both but some residual NOx remains anyway.

Emissions control requires then NOx storage catalytic converters; these ATS are simply traps capable to hold the pollutants until they can be reduced to nitrogen during phases of rich combustion: the lambda number represents the set-point control value for both exhaust recirculation and catalytic exhaust gas scrubbing.

On diesel engines, the lambda number is also used as the basic value for setting the smoke limit at full load.



Figure 4 – Typical functional layout of a lambda measuring module

Due to their continuous measuring characteristics, these sensors enhance the dynamics of the classic lambda control loop. The LSU wideband sensors from Bosch work reliably at temperatures above 600°C. They are designed for a permanent duty cycle at exhaust gas temperatures of up to 930°C and short-term peaks of up to 1030°C. As the sensors incorporate an internal heater, they are operational with cold exhaust flow and functional within a few seconds of starting a cold engine with minimal influence of the exhaust temperature on the sensor signal.

The most interesting device of the lot, crucial for research activities, is probably the ES910 RP module visible in Figure 5 that allows validation of prototype implementations for the software components of new open-loop control, closed-loop control, and diagnostic functions under realistic ambient conditions; the individual parts of the software to be tested can be coded manually in C, MATLAB/Simulink or any other suitable tool with minor concerns about compatibility and integration thanks to INTECRIO, which makes it possible to combine them in a single .a2l file ready to use.



Figure 5 – ETAS main rapid prototyping and interface module

Moreover, INCA can implement the ES910 RP module for calibration, for acquiring control unit and bus signals, for flash programming and for diagnostics.

The ES910 RP module also supports automated calibration on the test bench: the usual setup uses INCA as a connecting link between the test bench-based and the ECU-based engine controls deploying for the task the conventional ASAP3 interface.

However, when testing modern engines featuring a large number of actuators, the data volume, especially in the case of high engine speeds, is so large that the bandwidth of the ASAP3 interface no longer suffices to guarantee stable test bench operation: this is where the ETAS INCA-MCE (Measurement and Calibration Embedded) engineering solution comes to play.

The data chain in this case evolves according to the red line in the following Figure 6. Real-time capable drivers connect the bench controllers via CAN with the indexing system and via EtherCAT or iLinkRT fast routing protocols directly to the ES910 RP, replacing the conventional ASAP3/INCA interface thus bypassing the main PC: this is possible because the most relevant INCA functions – above all the ability to convert from and to .hex format – are already featured by the MCE software.

The large computational power of the ES910 RP module ensures then the rapid exchange of measured values and calibration variables between the control unit and the test bench's automation unit, converting from address-based binary values to name-based physical quantities and vice versa according to the direction of the data flow.

For calibration under steady state conditions, test bench calibration setups utilizing today's ASAP3 and ASAM MCD-3 MC automation interfaces are well suited but, to investigate the highly dynamic processes typically experienced during road travels, the INCA-MCE becomes mandatory and the overall layout evolves in a way similar to that shown in Figure 6.



Figure 6 – Bench data communication paths

The last piece of equipment usually employed among those of the ETAS family, is a powerful Multi-I/O all-rounder visible in Figure 7 with several digital and analog input and output channels for signal recording and output; it can be used for rapid prototyping, testing and calibration.



Figure 7 – ETAS multi input/output analog/digital all-round module

The inputs can be operated either in a predefined, equidistant time raster or in an event triggered mode: in the latter case, a measured value is transmitted only when a specific event has occurred on a digital input; such an event, a falling signal edge for example, triggers the synchronous measurement of all inputs coupled to this event. In the ES910 RP module, these event-triggered input signals facilitate the synchronization of the model sequencing to external events.

Besides ETAS hardware, two more units have to be mentioned:

 UMS 10: the acronym stands for Universal Measurement System but it is simply a RP device (visible in Figure 8) that electronically commands the VGT, the EGR valve and the additional flap installed in the intake manifold to produce a vacuum and further increase the EGR flow rate when the EGR valve and the analogous flap mounted in the exhaust pipe (downstream of the turbine) have already reached their maximum range of motion.

The series production TC group has been replaced indeed with a smaller one because the engine features a short-route EGR and since huge amounts of it are required to allow PCCI conditions, the rotor dimensions must be reduced to keep speed and boost high also at low loads, when the inlet feed drops drastically due to the particularly high EGR levels; being the mechanism controlling the blades incidence angle pneumatically controlled in the dismissed piece, the ECU isn't capable of actuating the system in the new assembly even if this is actually electronically actuated because its software is missing the proper instruction: an external means is therefore required to change the turbine geometry once having excluded the original compressed-air control implemented on the ECU that isn't effective neither on the EGR valve, now (as will be explained) of the throttle type and electronically actuated too.

Similarly, moderate EGR levels are sufficient for conventional combustion and in fact the old intake manifold wasn't provided with any sort of throttle: no command was then told to the ECU about; its presence in the new configuration demands instead the intervention of the UMS 10 again.



Figure 8 – UMS 10 rapid prototyping and interface module

• MRU: the acronym stands for *Main Relay Unit*; it is displayed in Figure 9 and its task is to coordinate in a single network all the control units onboard the vehicle with the main one: the ECU; obviously, these are not all employed for testbed applications but the ECU still requires the MRU to operate properly.



Figure 9 – MRU bench control units networking and ECU-interfacing module

8. Testbed salient parts of the experimental apparatus

The bench is entirely equipped with AVL instrumentation whose operational control and coordination is performed by the software PUMA OPEN through the Front End Modules (FEM) relaying the bulk of wirings from each sensor to the main computer, while the indicating software INDICOM manages the high-frequency signals through amplifiers and INCA provides the interface for the calibration tasks.

The test bench is equipped with the AVL APA 100 dynamometer visible in Figure 1 whose main characteristics are reported in the Table 1 immediately below.

It is a reversible electric machine connected to the engine crankshaft able to impose the desired load at the set rotational speed; it behaves as a brake system whose braking effect is took out from induced eddy currents but can also be operated as a motor, allowing investigations on the organic losses, for example.

In addition, this specific model presents two output shafts permitting to test two engines although not at the same time – as they would rotate with opposite sense of rotation, given the relative positioning, putting under torsion the shared machine' shaft – without the need to dismount any component except the link between the crankshaft of the engine not to be run and the dynamometer flange.



Figure 1 – AVL APA 100 dynamometer

Table 1 – AVL APA 100 main characteristics

Technical Data	
Maximum power [kW]	200
Maximum torque [Nm]	525
Maximum rotational speed [rpm]	12000
Inertia moment [Kgm2]	0,32

For the continuous measurement of the engine fuel consumption, the cell is equipped with the AVL KMA 4000 displayed in Figure 2.

Inside the device, the fuel is intaken and sent to a bubble separator; a pump then drives it through a filter, an heat-exchanger, a density sensor and a volume flow rate meter; the pressure is maintained constant by means of relief valves.



Figure 2 – AVL KMA 4000 fuel balance

A secondary pump pushes the fuel in a temperature regulator and finally into the engine; the return line of the injection system is connected upstream of the secondary pump after the fuel has been purged by a bubble separator.

The fuel consumption measurement is therefore operated on the basis of density and volume readings and corrected with respect to temperature.

An external hydraulic circuit filled with common tap water manages the various cooling systems of the engine: engine coolant heat-exchanger, intercooler and EGR heat-exchanger.

The engine to be tested is fitted on the bench without the type of radiator usually installed on a regular vehicle; for this reason, it is connected to a cooling system that is mainly composed by a heater, a heat-exchanger, a circulation pump and a thermostat. While production cars feature in most cases an air/liquid device, the CoolCon (visible in Figure 3) shows a conditioning system made by a liquid/liquid heat-exchanger where the coolant fluid is the water directly arriving from the water piping.



Figure 3 – CoolCon engine cooling management system' hydraulic layout

The coolant coming out from the engine first enters the heater where its temperature is brought towards the desired operating value to accelerate the warm-up if the engine has just been started; the coolant then enters the heat-exchanger where a PID controlled solenoid electro-valve regulates the water flow rate to make sure the engine coolant is at the right temperature before it is reintroduced within the engine again. The system is also provided with a level sensor and a degasser to prevent from air bubbles formation in the piping.

Furthermore, a second piping is directly connected with the engine intake downstream the engine thermostat: through this piping the engine cooling system is fed with cool water in order to fasten the cooling down operation after the engine has been run on the bench: this solution allows a sensible reduction in time between two consecutive tests.

As far as the intercooler is concerned, it is composed by a water-air heat-exchanger whose water circuit is independent from the one of the engine coolant.

The task of the intercooler is to cool down the intake air compressed by the turbocharger in order increase its density and thus improve the engine volumetric efficiency; the cold water entering the heat-exchanger is controlled by means of a solenoid valve operated by a PID control that allows setting the desired intake air temperature.

A picture of the intercooler employed at the bench is displayed in the left part of Figure 4.



Figure 4 – Compressed intake air intercooler (left) and EGR cooler (right)

In conclusion, the EGR cooler, in the right part of the same figure, exploits a heatexchanger that is similar to the one of the intercooler, the only difference lying in the fact that it uses the same standard cooling circuit of the engine: it can be controlled through the same valve that controls the engine coolant temperature.

9. Results

9.1 Double injection patterns: second injection timing sweep at increasing load – BSFC

The curves in Figure 1 are obtained keeping the engine stationary at the point 1400×27 (in terms of engine rotational speed and torque, where the units are rpm and Nm, respectively) and enlarging the dwell-time (DT) between the first and second injection by maintaining the former fixed at a certain SOI_{1st} (start of injection of the first injection) and postponing the latter only.

Each color corresponds to a specific combination of the calibration parameters among which the ones relevant to the discussion are reported in the legend while the vertical axis has been normalized for confidentiality reasons against the values recorded for the same quantity, when running under an early single-pulse PCCI strategy (see the horizontal solid line denoted in the legend "PCCI Single injection - map calibration" in the following graphs) optimized for the engine under test over a region of the map including all the points of interest for this study.

Instead, the dashed horizontal line highlighted in the legend as "conventional diesel combustion" provides indications about the values recorded when running under a triple-pulse conventional diesel combustion in the F1C production engine this prototype is derived from.

What has been just stated above is valid almost for all the graphs that are going to be shown in the following and it is then needless to say that no further explanations will be added once the discussion will move to them, unless specifically required for a correct interpretation.



Figure 1 – BSFC as SOI_{2nd} is postponed keeping SOI_{1st} fixed for double patterns at 1400×27

With reference to Figure 1, retarding the injection of the second pulse keeping the first one fixed causes a slight reduction of fuel consumption: in-cylinder conditions oppose more and more to the spray penetration and thus increase the combustion efficiency by reducing the amount of fuel impinging on the walls and enlarging the percentage of it that will participate in the combustion event.

The heat release phasing towards the expansion stroke also reduces the adverse pressure gradient typical of earlier timings that contrasts the piston motion with negative effects on the thermal efficiency.

In agreement with the considerations just made, the figure shows how better results are obtained when placing the first pulse closer to the TDC or increasing the fuel splitratio in favor of the second one: being close to the TDC or directly moving fuel from the first to the second injection means in both cases intake in-cylinder conditions less prone to wall impingement causing less impact on the thermal efficiency with a slight increase of the power output.

As it is possible to see, multiple-pulse strategies feature values not too distant from the one of the single-pulse scheme but quite higher than that of a conventional combustion: the lower compression-ratio due to the bowl design modifications, the lower combustion efficiency related to never negligible wall impingement phenomena and the higher induction temperatures due to the presence of higher EGR rates that further reduce the volumetric efficiency and thus the amount of air brought into the cylinder, all contribute to the higher BSFC that, nonetheless, keeps decreasing as usual as the load increases and the engine moves towards zones of the map characterized by higher efficiency.

It is worth stressing that the correct oxygen intake is fundamental to achieve the last stages of the oxidation process, late in the cycle: this aspect becomes crucial considering that besides the air already displaced by the EGR itself, a further reduction is to be expected due to its high temperatures.

The sequence of figures below shows how the same considerations made for Figure 1 apply also at higher loads, with multi-pulse strategies going considerably above the horizontal red line in Figure 4 probably because of the slightly over-advanced timing of the pilot.



Figure 2 – BSFC as SOI_{2nd} is postponed keeping SOI_{1st} fixed for double patterns at 1000×94



Figure 3 – BSFC as SOI_{2nd} is postponed keeping SOI_{1st} fixed for double patterns at 2000×120



Figure 4 – BSFC as SOI_{2nd} is postponed keeping SOI_{1st} fixed for double patterns at 2000×161



Figure 5 – BSFC as SOI_{2nd} is postponed keeping SOI_{1st} fixed for double patterns at 2000×200



Figure 6 – CO as SOI_{2nd} is postponed keeping SOI_{1st} fixed for double patterns at 1400×27

With reference to Figure 6, retarding the injection of the second pulse keeping the first one fixed causes an increase of CO emissions.

In-cylinder conditions are responsible for ever increasing amount of un-vaporized fuel and reduced mixing when the second pulse starts burning, and the diffusive flame, lighting up earlier and earlier because of the shorter premixed phase determined by smaller quantities of accumulated fuel, further deteriorates the preparation of the remaining portion of fuel that is being injected in the meanwhile, with higher CO emissions, partially arising from the modest peak temperatures too.

Additionally, retarding the second injection, the targeting point of the spray – optimized for the largely advanced timings of the first injection – sprays the fuel directly on the dome of the piston bowl causing a liquid film deposition that will combust only partially thus resulting in increased emissions of products of incomplete combustion.

The shift of the heat release towards the power stroke eventually causes a dramatic reduction of the oxidation rates due to the downward motion of the piston, decreasing in the meanwhile the time available before the discharge for post-combustion reactions that although supported by the higher temperatures of the exhaust gases coming from a postponed combustion, still barely take place because of the temperature and pressure drops during the expansion.

As can be seen indeed in Figure 7, when passing from the 1400×27 to the 1000×94 engine point, the reduction in speed possibly allows this extra-time for a re-burning of the incomplete products of combustion before the exhaust stroke as shown by the descending trends that, to be rigorous, do not apply for the 50/50 split-ratio condition probably because the increased amount of fuel burning in the second injection combustion in a mostly diffusive way yields to an amount of CO so large to be unaffected by the speed reduction.



Figure 7 – CO as SOI_{2nd} is postponed keeping SOI_{1st} fixed for double patterns at 1000×94

Actually, this is exactly the reason why the 75/25 split-ratio strategies seem to be in the majority of the cases the most convenient ones. In these cases, there exists an optimal timing for the first injection called *sweet-spot* – as defined by Opat and Ra in [9] – strictly dependent on the current engine working point for a given combustion system, as will be explained in detail later, that is a compromise between the impingement caused by an over-advanced injection command and the under-mixing determined instead by a too retarded one causing the DT before the second injection to shrink to the point to trigger negative interactions between the two pulses.

All in all, it is better to put more fuel in the first injection so as to let it burn in a premixed way with a higher degree of vaporization and mixing exploiting the peak temperatures and availability of pure air in the surrounding to bring to completion the oxidation of possible products of partial combustion, without mentioning that the following burn of the second pulse will still be beneficial to the purpose.

The second injection combustion is instead followed by the expansion stroke only and has a reduced amount of time, before temperature drops and burnout mechanisms freeze, to try to completely oxidize any residual CO prior to the exhaust phase.



Figure 8 – CO as SOI_{2nd} is postponed keeping SOI_{1st} fixed for double patterns at 2000×120



Figure 9 – CO as SOI_{2nd} is postponed keeping SOI_{1st} fixed for double patterns at 2000×161



Figure 10 – CO as SOI_{2nd} is postponed keeping SOI_{1st} fixed for double patterns at 2000×200

As the load increases, following the sequence from Figure 8 to Figure 10, the CO decreases because of the higher quantity of fuel introduced, heat released and values of temperature and pressure able to reduce wall impingement and enhance reactions completeness for maximum combustion efficiency.

Obviously, numbers are far above those of a conventional combustion being the augmented CO/HC emissions one of the main drawbacks of PCCI, and in general of every LTC method, that works such as the one presented in this thesis intend to mitigate but improvements seem to happen with respect to single-pulse strategies especially at higher loads, when the quantity of fuel to be injected is so large and the intake in-cylinder conditions are so favorable to autoignition to cause most of the fuel to be sprayed directly inside the flame with very poor vaporization/mixing and intensity of the premixed phase, resulting in vast amounts of products of incomplete combustion that have also to face the lack of free oxygen due to the decreased lambda.

Comparisons of in-cylinder CO distributions between multiple and single injections [18] show that higher concentrations of CO are located along the cylinder axis with multiple injections and at the cylinder walls for the single injection: under similar intake oxygen concentrations, a higher rate of fuel injection signifies relatively fuel rich equivalence ratios in the single injection case.

Thus the presence of CO along the cylinder axis and bowl path regions due to the overlean mixtures with a multiple injections is not observed with the single injection, whose CO rises instead from HC films on the walls.

9.3 Double injection patterns: second injection timing sweep at increasing load – Soot With reference to Figure 11, retarding the injection of the second pulse keeping the first one fixed causes a negligible amount of soot increase at very low loads because, for each one of the examined calibrations, the levels of pressure and temperature inside the cylinder are so low to allow an extension of the ID so substantial to achieve remarkable levels of mixture preparation before the autoignition.

Low load means less fuel to be injected but also quite high O₂ concentrations at the intake for a more uniform distribution of the fuel inside the charge whereas the long ID allows to accumulate most of it before autoignition, giving rise to a predominant premixed phase.



Figure 11 – Soot as SOI_{2nd} is postponed keeping SOI_{1st} fixed for double patterns at 1400×27

The U-shaped smoke trends – clearly visible especially in Figure 12 – obtained by varying the SOI of the second pulse, can be explained with reference to [10]. The increase in smoke obtained at the shortest DT has to be ascribed to the inhibited mixing/oxidation of the rich flame formed by the second fuel injection, which cannot use the air in the center of the piston cavity.



Figure 12 – Soot as SOI_{2nd} is postponed keeping SOI_{1st} fixed for double patterns at 1000×94

The closed curve in Figure 13 (a) denoted "Rich-Flame" simulates the in-cylinder situation in the latter part of an early PCCI cycle for a stationary engine point almost equivalent to that of Figure 12: 1300 rpm and 3.7 bar; the dotted rectangle circumscribes the area of in-cylinder conditions required for a fast burnout of the residual CO; each dot in Figure 13 (b) refers instead to a single elementary cell of the 3D combustion chamber discretization preliminary to the numerical simulation of the post-combustion phase on which the mentioned study is based and the level of grey-shading is directly proportional to the CO concentration in that cell.



Figure 13 – [10] Kamimoto-Bae diagrams depicting the in-cylinder temperature/stoichiometry conditions in the latter part of an early PCCI cycle for an engine point stationary at 1300 rpm/3.7 bar

As the CO oxidation proceeds in the cells whose temperature and stoichiometry fall inside the dashed rectangle, the corresponding dots whiten-out splitting the Rich-Flame in Figure 13 (b) in two zones: one characterized by temperatures too low and the other by a shortage of oxygen, both inhibiting from different sights the Rich-Flame oxidation.

A visual check of the same figure is enough to tell that the majority of cells is included in the Rich-CO subgroup and actually this is to be expected as the load increases and the lambda goes consequently down, as can be seen in Figure 9 of the previous paragraph on CO for 50/50 split-ratio strategies, obviously more prone to such issues being higher the quantity of fuel injected in the second pulse.

Although arising from CO discussions, the phenomenon has a major impact on soot too: with an over-advanced timing, the Rich-Flame of the second injection travels across the combustion chamber thanks to conditions favorable to spray penetration and the aerodynamic drag reduction offered by the wake left by the previous pulse, until reaching the side walls and burning there together with it.

This implies an underutilization of the air in the center of the piston bowl and a slowdown of the process to the point that CO/soot distributions remain almost unchanged until the mixture is finally exhausted.

As about to see in the following, the single-peak history of the HRR suggesting the unified burning of the two pulses, is the prime suspect for the noise rise experienced in this case.

Over-retarding the second pulse injection, instead, the soot increases again due to the flame of the first injection combustion propagating through the second pulse jet up to the near-nozzle region, before the latter can even vaporize.



Figure 14 – Soot as SOI_{2nd} is postponed keeping SOI_{1st} fixed for double patterns at 2000×120

In light of what has just been said, Figure 14 shows how the situation obviously worsens in terms of soot emissions when the fuel split-ratio of the second pulse is increased and although, by definition of PCCI, these are much lower than those of the

conventional combustion, the single-pulse strategy seems to be always better, with the exception of the points at higher loads.

As can be seen following the sequence from Figure 14 to Figure 16, the horizontal red line referring to the single-pulse PCCI starts climbing along the vertical axis to end up quite above the multi-pulse strategies, for a correct selection of the DT, in Figure 16.

In fact, regardless of the longer charge preparation time extending the ID and accumulating more mass for the rapid burn under premixed conditions, the switch to the diffusive flame regime advances earlier and earlier as the load increases because the in-cylinder conditions become more favorable to autoignition, causing ever increasing portions of liquid to be injected directly into the flame and making the single-pulse strategy less suitable for high loads.



Figure 15 – Soot as SOI_{2nd} is postponed keeping SOI_{1st} fixed for double patterns at 2000×161



Figure 16 – Soot as SOI_{2nd} is postponed keeping SOI_{1st} fixed for double patterns at 2000×200

Increasing the load means a reduction of lambda and then an increase of smoke as a general trend in the series of figures above.

To conclude this paragraph, it is worth remarking that while experimental observations have suggested that soot is formed and accumulates at the fuel spray tip of a diesel engine, by splitting the fuel injection pattern, there is a substantial decrease in the amount of smoke because rich regions (conducive to soot formation) are no longer being fuel-replenished during the time interval between injections.

In addition, the use of split injection schemes is an effective mean to improve the fuelair mixing and lean out the mixtures thanks to the augmented air entrainment following the fuel delivery throttling but the correct choice of the DT remains crucial.

Choosing the correct timing for the second injection not only decides to which extent the high temperature region lasting from the previous combustion will impact on the next combustion event but, symmetrically, how much the turbulent effect induced by the injection of the second pulse will enhance the mixing and re-burning of partially oxidized species coming from the burn of the first pulse.

9.4 Double injection patterns: second injection timing sweep at increasing load – HC

HC trends are similar to those pertaining to CO and therefore the same considerations given before can be held also in this case with the only exception of the fact that, as far as HC are concerned, it is better to move the first injection closer to the TDC and to increase the fuel split-ratio in favor of the second one.

HC molecules deposit as an actual liquid film on the combustion chamber walls and are therefore heavily ruled by the impingement phenomenon which can be contrasted by reducing the spray penetration length (first injection delay for adverse intake incylinder conditions) or the amount of fuel injected in the first pulse (lower jet inertia and less mixture to prepare in the same amount of time).

As a prove of what has just been said, in Figure 17 a comparison is made between the HC and CO plots relative to the same engine point: 1400×27; the same can be said for all the other points under test but, for sake of brevity, further comparisons are omitted.



Figure 17 – HC vs. CO as SOl_{2nd} is postponed keeping SOl_{1st} fixed for double patterns at 1400×27

Obviously, in light of what has been explained above, the advantages of split-strategies are clear with respect to single-pulse ones as are the disadvantages of LTC methods in general with respect to the conventional combustion (still from the HC perspective) as better shown by the following figures.



Figure 18 – HC as SOI_{2nd} is postponed keeping SOI_{1st} fixed for double patterns at 1000×94



Figure 19 – HC as SOI_{2nd} is postponed keeping SOI_{1st} fixed for double patterns at 2000×120

It is interesting to notice how trends flatten-out as the load increases, as it is possible to see when switching from Figure 17 to Figure 18 and so on up to Figure 21 for all the set of calibrations investigated during the experimental campaign: this is clearly representative of the great influence of in-cylinder conditions on the wall impingement phenomenon but also of the marginal control exerted by the second injection timing on this pollutant species, as seen for CO.

As a matter of fact, the sequence of graphs considered so far also features two more curves referring to calibrations employed for analogous investigations on DT made by keeping the SOI of the second pulse fixed and advancing the one of the first injection only, more and more away from the TDC back into the compression stroke.

Such analysis is postponed to another section of this chapter and any explanation is therefore omitted for now but it is worth noticing how the timing of the first injection plays instead a determinant role in the control of CO and HC emissions.



Figure 20 – HC as SOI_{2nd} is postponed keeping SOI_{1st} fixed for double patterns at 2000×161



Figure 21 – HC as SOI_{2nd} is postponed keeping SOI_{1st} fixed for double patterns at 2000×200

Again with reference to [18], in multiple injections, the measured HC concentration is 2.3 times lower as compared to single injection and HC are located primarily along the cylinder axis near the injector, which could be due to the contribution from over-lean regions associated with increased mixing at the end of injection.

The absence of higher HC concentration regions along the cylinder wall in multiple injections indicates that the crevice originated HC is negligible, which means that the majority of the injected fuel is well mixed and participates in complete combustion before it reaches the crevice, which is also augmented by the shorter ignition delay.

Thus, the crevice originated HC is the major source in the single injection case while the over-lean regions along the cylinder axis contribute to the HC emission in multiple injections.

The two main formation mechanisms are indeed undermixing and overmixing.

As described in [5], early injections cause the start of the premixed-burn to be postponed and, as a consequence, there is more time for penetration and mixing implying that increasing portions of the mixture shift toward leaner ratios with major risks for the fuel at the boundary of the jet directly in contact with the air cushion to overmix or even fall below the flammability limit.

PCCI conditions featuring extremely extended ignition delays are then naturally prone to produce lots of HC because of overmixing.

What happens instead during the transient the injector closes is to be ascribed to the phenomenon of undermixing: spray quality worsens as the needle completes its downward stroke and the liquid fuel trapped in the sac (a little capacity upstream the nozzle holes is required to guarantee maximum uniformity of the 8/9 jets usually employed) spills in the combustion chamber drew in by the piston on its way to the BDC.

Obviously the situation becomes critical at high loads when the quantity to be injected increases and the transients duration prolongs to allow the maximum needle lift for higher fuel flow rates.

9.5 Double injection patterns: second injection timing sweep at increasing load – Noise



Figure 22 – CN as SOI_{2nd} is postponed keeping SOI_{1st} fixed for double patterns at 1400×27

As seen before talking about soot, when the timing of the second injection is too advanced for a given SOI of the first one, the jet of the former traverses the combustion chamber under the influence of intake conditions more favorable to spray penetration and of the aerodynamic wake left behind by the first pulse after its passage.

This causes the jet of the second pulse to make a scarce utilization of the air in the central portion of the piston bowl, compromising not only the mixture but also the

combustion quality as it reaches the flame of the first pulse and burns with it giving rise to a single-peak history of the HRR.

The unified burning of the two pulses at the outer boundaries of the piston cavity might explain then the noise rise experienced in this case, as shown by Figure 22.

The trends then descend – enlarging the DT – not only because of the reduced interaction between the two pulses but also because of the milder combustion of the second injection as its timing approaches the TDC: the in-cylinder conditions at the time of its SOI are increasingly favorable to autoignition and thus shorten the ID, reducing the duration and intensity of the premixed burn and as a consequence shifting earlier and earlier the switch to the diffusive combustion regime.

The body of heat release is then postponed from the compression into the expansion stroke causing a further drop of the pressure rise rate and the extension of the combustion duration beyond the TDC when the gas-dynamic expansion happens during the geometric expansion of the combustion chamber volume as the piston makes its way down to the BDC.



Figure 23 – CN as SOI_{2nd} is postponed keeping SOI_{1st} fixed for double patterns at 1000×94

In light of what has just been said, Figure 23 proves that lower levels of noise are obtained delaying the first pulse and increasing the fuel split-ratio in favor of the second injection in order to reduce the amount of premixed combustion and shift the HRR center of gravity away from the compression towards the expansion stroke.

With the proper DT, the advantages with respect to the single-pulse strategy are substantial although this might be not so evident from the figures: values, as already said, have been normalized with respect to those obtained with a single-pulse PCCI calibration optimized for the portion of the engine map of interest for the scope of this work, but the vertical axis in the charts above still remain expressed in logarithmic scale, meaning that +/-3 dB translate into doubling/halving the corresponding noise intensity level in linear scale, respectively.

The following figures show the magnitude of CN reduction at higher loads with a proper choice of DT.

As usual, the consistently higher values in Figure 25 for the 2000×161 engine point might be due to an excessive advance of the first injection shifting its heat release way too back into the compression stroke and causing the steep pressure rise rate coming from an extended premixed burn to collide against the volume reduction induced by the piston.



Figure 24 – CN as SOI_{2nd} is postponed keeping SOI_{1st} fixed for double patterns at 2000×120



Figure 25 – CN as SOI_{2nd} is postponed keeping SOI_{1st} fixed for double patterns at 2000×161

Experimentally, it is observed that although the single injection has a higher percent of rich regions, their occurrence is earlier for multiple injections which could be due to a shorter charge preparation time owing to a smaller injected fuel mass at any given time.

The early occurrence of fuel rich regions results in early ignition and thus a shorter ignition delay in multiple injections; the high temperature regions grow also more gradually for the multiple injections signifying a gradual progress in combustion while it is much more rapid for the single injection.

From the delayed start of combustion and the subsequent wider distributions of high temperature regions in the single injection, it is evident that larger amounts of premixed charge mass participate in the combustion in a shorter interval of time and thus result in a higher pressure rise rate and higher combustion noise.



Figure 26 – CN as SOI_{2nd} is postponed keeping SOI_{1st} fixed for double patterns at 2000×200

Thus, the benefit of lower combustion noise by the use of multiple injections, particularly evident under the high load conditions displayed in Figure 26 above, comes primarily through the shorter ignition delay, resulting in lower premixed combustion: this is made possible due to the better charge preparation owing to a lower mass of pilot injected fuel; further, favorable in-cylinder conditions in terms of a higher pressure and temperature during the subsequent injections would also contribute for a better charge preparation in multiple injections enhancing the diffusion phase over the premixed one.

Clearly, the noise increases with the load and regardless of the improvements with respect to single-pulse strategies just described, the gap to the numbers typical of the conventional combustion remains significant: the latter features indeed still a triple injection scheme and the timing is intended to have most of the heat released after the TDC for maximum torque and efficiency, and not in the compression stroke as tends to happen in PCCI.

As a final remark, what prevents the PCCI combustion mode from being exploited at high loads/speeds, when its use would reveal the most effective to contrast the unacceptable increase in NO_X and soot observed for conventional combustions, are the noise levels so huge to make any application on regular vehicles basically unfeasible.

9.6 Double injection patterns: second injection timing sweep at increasing load – NO_X Retarding the injection of the second pulse keeping the first one fixed causes a slight reduction of NO_X which is one of the most common practices to move the soot-NO_X trade-off in favor of lower emissions of the latter.

Anyway, there is not a winning strategy among those examined since, for any of the engine point under test, NO_x levels are so low to be virtually negligible, as expected by definition of PCCI.

EGR rates are always so relevant to prevent in-cylinder temperatures from reaching values high enough to trigger the first reaction of the Zeldovich' sequence with a significant temporal/spatial frequency.



Figure $27 - NO_X$ as SOI_{2nd} is postponed keeping SOI_{1st} fixed for double patterns at 1400×27



Figure 28 – NO_X as SOI_{2nd} is postponed keeping SOI_{1st} fixed for double patterns at 1000×94

The situation is particularly evident with reference to Figure 27 and Figure 28.

Comparing single against multi-pulse strategies while keeping in mind the extremely low values dealt with, it turns out that the former shows better results: this is maybe due to the fact that a large number of injections spread the fuel distribution and consequently the heat release over an extended portion of the cycle prolonging the duration of high-temperature conditions.

Thus, irrespective of the much lower peak values if compared to those experienced after the prominent premixed phase of a single-pulse combustion, multi-pulse strategies still offer more time at higher temperatures and hence higher probability for NO_x formation.

This starts to be visible in the three figures below, from Figure 29 to Figure 30, as the load increases and the NO_X production, although always confined to very low levels, becomes more significant.



Figure 29 – NO_X as SOI_{2nd} is postponed keeping SOI_{1st} fixed for double patterns at 2000×120 and 2000×161



Figure 30 – NO_X as SOI_{2nd} is postponed keeping SOI_{1st} fixed for double patterns at 2000×200

NO_X obviously increase with the load and it is clear from the vertical axis range of values how drastically they are being cut under LTC conditions with respect to the conventional diesel combustion mode, highlighted as always by the horizontal dashed line in the figures.

9.7 Double injection patterns: first injection timing sweep at 2000 rpm/120 Nm

The following graphs have already been presented in the previous section of this chapter but the discussion made so far has been focusing only on the effects caused by a timing sweep of the second injection.

Indeed, in the same figures are also reported trends obtained for the same engine point by sweeping the first injection timing keeping the SOI of the second one fixed at the value highlighted in the legend: it is noticeable how acting on one or on the other, quantities result significantly affected or insensitive at all.

From now on the focus is then moved on the red and cyan curves unless differently specified during the discussion.



Figure 31 – BSFC as SOI_{1st} is advanced keeping SOI_{2nd} fixed for double patterns at 2000×120

Advancing the first injection keeping the second one fixed causes a slight increase in BSFC, as can be seen in Figure 31, due to the fact that spray targeting is altered and some of the fuel misses the bowl before impinging on the walls, thanks also to the incylinder conditions opposing very little resistance to the spray penetration.

Furthermore, this reduction in combustion efficiency is accompanied by the additional reduction in thermal efficiency due to the increased amount of heat released early during the compression stroke.

All in all, more fuel is required to deliver the same torque because of the two phenomena just described.

There is not a winning calibration and values are again similar to those obtained with a single-pulse PCCI and quite higher than the ones of a conventional combustion.

Starting from the CO graph visible in Figure 32, it is possible to see how moving the first is much more influential than moving the second injection: in the former case, trends exhibit a sudden increase beyond a certain DT value – approximately equal to 1500 μ s and corresponding to an advance of 35° for the first injection – while in the latter, curves seem to remain relatively flat.



Figure 32 – CO as SOI_{1st} is advanced keeping SOI_{2nd} fixed for double patterns at 2000×120

The above mentioned ramping up for increasingly advanced timings of the first injection is due to the ever larger portions of fuel missing the bowl and ending up in zones of the cylinder characterized by poor combustion (CO/soot) or directly on the walls (HC).

In addition, the cyan curve also shows a tendency towards higher CO emissions for excessively retarded timings of the first injection which, being brought closer to the second pulse, starts suffering of bad mixture preparation because of shorter ignition delays coming from temperature and pressure conditions more favorable to combustion and altered targeting on the bowl, being the spray angle optimized for early timings only.

The combination of all these factors affects the local mixture strength and air utilization and intensifies the mutual interferences of each pulse' combustion over the other.



Figure 33 – [9] Validation of experimental results obtained in a single-cylinder LD diesel engine running under LTC conditions at the point 2000 rpm/5.5 bar by means of a 3D CFD simulation of the combustion process

In Figure 33 are reported the results of a study [9] aiming at the validation of experimental results obtained in a single-cylinder LD diesel engine running under LTC conditions at the point 2000 rpm/5.5 bar (almost coincident to the point 2000 rpm/5.02 bar under focus) with a 3D CFD simulation of the combustion process coupled with the heat release analysis of measured in-cylinder pressure data and a further phenomenological model simulating the in-cylinder thermodynamics in such conditions.

The purple curve is obtained on the basis of the experimental results while the red ones represent the simulations outputs: as can be seen, the overall experimental trend is validated by superimposition of the simulations outputs and not only the shape but also the timing of the sweet-spot closely recall those in Figure 32.

Very similar trends are also found for soot and HC, as seen in Figure 34 and Figure 35, although the impact exerted on the former by the timing of the first injection is marginal if compared to that of the second one; a closer look gives also evidence of the change in slope for DT larger than 1500 μ s in the HC graph where the expected increase before the said threshold can't be appreciated due to the lack of data.



Figure 34 – Soot as SOI_{1st} is advanced keeping SOI_{2nd} fixed for double patterns at 2000×120



Figure 35 – HC as SOI_{1st} is advanced keeping SOI_{2nd} fixed for double patterns at 2000×120

Finally, the same considerations about fuel split-ratio and pros/cons with respect to single-pulse and conventional strategies given in the previous paragraph may be applied in this case to the last three quantities analyzed.



Figure 36 – CN as SOI_{1st} is advanced keeping SOI_{2nd} fixed for double patterns at 2000×120



Figure 37 – NO_x as SOI_{1st} is advanced keeping SOI_{2nd} fixed for double patterns at 2000×120

The noise, seen in Figure 36, seems to be unaffected by the timing selection for the first injection probably because the increased impingement with very early timings compensates for the SOC shifting more and more towards the compression by subtracting some of the fuel theoretically intended for heat release and driving it into the walls, from where it is later scraped and exhausted as HC.

Again, the two calibrations analyzed look comparable.

As far as NO_X is concerned, seen in Figure 37, the levels of this pollutant are so low to be virtually negligible, as expected by definition of PCCI: EGR rates are always so huge to prevent in-cylinder temperatures from reaching values high enough to trigger the first reaction of the Zeldovich' sequence with a significant temporal/spatial frequency.

The main question of this paragraph hasn't however been answered yet, namely: why is the first injection timing much more effective on the control of CO/HC levels whereas the second injection timing rules the trends of soot and noise?

Saying for completeness that BSFC and NO_x are instead depending mainly on the mode of combustion adopted – PCCI in this case – the answer might simply be that it cannot be otherwise: the typical pressure/temperature conditions under which they take place, the fact that the first pulse is followed by the injection/combustion of the second pulse while the latter is followed by the expansion stroke only, and the different uses the two have been intended for – combustion improvement from one side and power generation from the other – make some species *characteristic* of the first rather than the second injection and almost completely dependent on its tuning in terms of timing and fuel split-ratio.

9.8 Double injection patterns: timing sweep for a fixed DT at 2000 rpm/120 Nm

The following plots are obtained by rigidly moving both the injection pulses keeping unchanged the DT between them (the last timing option with only two pulses available) at a value of interest selected after investigating the results of previous measurement campaigns; for sake of simplicity, the SOI of the first injection has been taken as a reference and put on the horizontal axes.

The values on the vertical axes are, once again, normalized against those obtained with a single-pulse PCCI calibration optimized for the low-medium load/speed portions of the engine map.



Figure 38 – Main effects of a rigid shift of both pulses towards more advanced timings for different double patterns at 2000×120

In light of what has been said at the end of the previous paragraph, as expected, moving both injections causes substantial variations either on CO/HC (the characteristic species ruled by the first injection) and on soot/noise (the characteristic species ruled by the second injection) as seen in Figure 38.

All explanations given so far for trends, optimal calibrations among those in the set investigated at the bench and pros/cons with respect to single-pulse and conventional combustion strategies still apply here taking care to consider the minor changes due to the second injection this time being advanced too and not retarded as seen in the first paragraph.

9.9 Double injection patterns: exhaust flap sweep at 2000 rpm/120 Nm

To regulate the amount of EGR sent to the engine, it is possible to act on the exhaust flap placed downstream of the turbine and upstream of the main DOC: in this way, it is possible to create a counter-pressure that propagates backwards up to the point where the exhaust stream is split between the turbine inlet feed and the EGR shortroute path so as to increase the EGR mass flow channeled into the latter.

The values on the vertical axes are normalized in the usual way and the focus on this engine point only is made only for the sake of brevity as trends for other points of the map are very similar.



Figure 39 – BSFC as the exhaust flap is opened for double patterns at 2000×120

Reducing the throttling across the exhaust flap (from hereinafter it will be said for simplicity "opening" the exhaust flap) the counter-pressure decreases, so does the EGR flowrate, the intake air flowrate increases and the lambda value with it: moving from left to right on the horizontal axes corresponds then to opening the flap.

The BSFC seems to lower a little bit, as shown in Figure 39, thanks to the reduced dilution and higher combustion temperatures increasing the burning rate and shortening the heat release duration with benefits on the thermal efficiency.

In addition, the increased amount of cooler and more dense intake gases opposes a stronger resistance to the spray penetration, allowing a reduced impingement and
consequently a higher percentage of the total injected fuel mass participating in the oxidation process, with improvements of the combustion efficiency too. There is not a calibration to be preferred over the others.



Figure 40 – CO/HC/soot as the exhaust flap is opened for double patterns at 2000×120

With reference to Figure 40, as far as CO soot and HC are concerned, more air and increased in-cylinder temperatures for the reasons just explained, are the main drivers to achieve the final stages of oxidation during but especially after the combustion event, when the last pockets of rich stoichiometry struggle to find some available air before the exhaust.

As already said in the previous sections of this chapter, a fuel split-ratio in favor of the first injection is more beneficial to CO and soot emissions as the extended duration of the ID due to the larger quantity of accumulated fuel allows to achieve a superior mixture uniformity that, alongside the values of pressure and temperature usually reached under premixed conditions, results in a more stable and complete burn.

Unfortunately, more the fuel to be injected early in the compression stroke higher the amount of impingement due to the favorable in-cylinder conditions at the SOI with increased HC emissions that, as said, is indeed the pollutant on which the first injection tuning influence reaches the maximum extent.



Figure 41 – CN as the exhaust flap is opened for double patterns at 2000×120

The combustion noise, whose trend is displayed in Figure 41, starts dropping considerably for a value of lambda equal or higher than about 1.3 as less dilution means higher charge reactivity and thus lower ID and less accumulated mass to feed the premixed burn.

The premixed phase then shortens in favor of a longer diffusive burn which protracts after the TDC when the gas-dynamic expansion is no more contrasted by the rising piston motion; obviously, lower noise levels are to be expected with a fuel split-ratio in favor of the second injection as the body of heat release is shifted away from the compression phase.

Since in almost every LTC strategy EGR rates are deployed to the maximum possible extent in order to dilute the mixture until the combustion temperatures decrease is enough to avoid triggering the Zeldovich' sequence for NO_X formation, it naturally follows that the production of these species shall increase at higher values of lambda.

As shown in Figure 42, indeed, since less O_2 is displaced away from the flame front by the EGR molecules, higher temperatures must be reached on its surface as the combustion zone is no longer broadened by inert particles absorbing heat and reducing the reactivity of the mixture.

Likewise, letting more fuel to burn in the pilot means sustaining even more the temperature rise and the NO_X production.



Figure 42 – NO_x as the exhaust flap is opened for double patterns at 2000×120

9.10 Double injection patterns: rail pressure sweep at 1400 rpm/27 Nm

The last section of this chapter is covered by the analysis of the influence of the injection pressure on the set of quantities of interest for this work, so far confined to double-pulse strategies.

Once more, the vertical axes have been normalized as previously done and the focus on this engine point only is made for the sake of brevity as the corresponding graphs from other points of the map show trends close to the ones displayed here.





Figure 43 – BSFC/CO/HC as the rail injection pressure is increased for double patterns at 1400×27

Higher fuel delivery pressures means higher velocities and thus deeper spray penetration with increased probability of wall impingement especially at lower loads, like the one under examination.

This reflects on BSFC, CO and HC emissions, whose trends are all increasing as shown in Figure 43.

As higher injection speed means augmented air entrainment as well as more time for charge preparation, for a given engine speed, the fuel at the outer boundaries of the jet risks to lean-out so much to pass the flammability limit and incur in the overmixing phenomenon or to reach the most remote zones of the squish volume, where the combustion is poor and very likely to remain incomplete.

With reference to Figure 44, as far as soot and noise are concerned, the mechanism at the roots of their behavior is the same: the expected reduction of the Sauter Mean Diameter (SMD) – an aerodynamic parameter of the spray inversely proportional to the mixture preparation time – still takes place for increasing values of the injection pressure but, being the starting point of the sweep already substantial in magnitude, the entity of this effect is totally negligible and the atomized fuel droplets undergo minor physical changes with very little relevance from the charge preparation point of view.



Figure 44 – Soot/CN as the rail injection pressure is increased for double patterns at 1400×27

This causes the ID to remain practically the same but the mass accumulated before ignition is now much larger because of the enhanced flow rate, meaning that most of the injected quantity will burn under premixed conditions, known for their heavy impact on soot and noise control: the former decreases and the latter increases, particularly at the fueling rates typical of the high load/speed portion of the map.



Figure 45 – NO_X as the rail injection pressure is increased for double patterns at 1400×27

 NO_X turned out to be practically unaffected, as they are consistently low and close to the usual PCCI levels, as shown in Figure 45.

9.11 Triple injection patterns at 2000 rpm/120 Nm

The figures below show what happens advancing either the first and second injections or the whole set of three injections keeping the DT between two consecutive pulses fixed, in a triple injection scheme, at the 2000×120 engine point.

The trends aren't much different with respect to those obtained for a double-pulse strategy following the same method of rigid shifting away from the TDC (towards the compression stroke).



Figure 46 displays the whole series of calibrations explored during the experimental measurement campaign.



Figure 47 – BSFC as the first two or all the three pulses are advanced for triple patterns at 2000×120

As far as the BSFC is concerned, curves in Figure 47 are in general a little bit higher along the vertical axis compared to the analogous experiment made with double injections probably because, besides all the already exposed drawbacks in terms of combustion and thermal efficiencies related to further adding another pulse very early during the compression stroke, the increased number of injector actuation transients starts having a role.

In fact, as described in [5], the more the injector is opened/closed the higher the slip of extra-fuel beyond the quantity required by the current engine point due to constructional limitations extending actually to the injection system as a whole: pressure waves propagation in the pipes, delays, capacities replenishment/emptying etc. all contribute to alter the nominal quantity of fuel commanded by the ECU for the given engine point.

Regardless the great number of tests made, there seems not to be a winning calibration.



Figure 48 – CO as the first two or all the three pulses are advanced for triple patterns at 2000×120

The CO graph in Figure 48 is virtually superimposable to the corresponding one obtained with double injections and again there isn't a best calibration prevailing over the others.



Figure 49 – Soot and HC as the first two or all the three pulses are advanced for triple patterns at 2000×120

With reference to Figure 49, the *U* describing the soot behavior this time evolves into an *hyperbola-like* shape probably because, even if a larger portion of the fuel injected within the first pulse is missing the bowl given the huge advance, for the conservation of the total quantity – the engine point is the same – the fuel delivered each time is less than before, as proven by the HC plot. HC curves are indeed slightly below those recorded with double injection patterns due to the reduced impingement favored by smaller and less penetrating jets: liquid layers all deposit as raw fuel and then, depending on the location on the walls where they impinge, during the cycle each will undergo a substantial, partial or no combustion at all to finally evolve into soot, CO and HC, respectively, at the exhaust.

Smoke production decreases keeping the DT between the pilots confined within 700 and 800 μ s while there are no appreciable differences between calibrations for the HC.



Figure 50 – CN and NO_x as the first two or all the three pulses are advanced for triple patterns at 2000×120

Noise, whose curves are visible in the upper portion of Figure 50, slightly improves thanks again to the lower instantaneous quantity of injected fuel, characterizing triplepulse strategies with respect to double injection patterns, that causes a longer less intense HRR sustaining for more time the persistence of high-temperature conditions – although at reduced peak values – helping to bring to completion the oxidation of soot and to extend the probability of NO_x formation.

As proven indeed by the relative graph on the lower portion of the same figure, curves are a bit above those of the double scheme and exhibit a tiny peak too for the value of advance *characteristic* for the first injection at the given engine point and for the specific combustion system under deployment.

The subgroup of calibrations highlighted before is now the one responsible for the higher noise levels while no difference can be seen with reference to NO_x.

The set of graphs displayed starting from Figure 52 up to Figure 54 shows instead the effects caused by changing the timing of the second or of the third injection keeping the others fixed.

To explore such effects on the usual group of quantities of interest for a proper assessment of the quality of combustion, it has been decided to compare the series of calibrations reported in Figure 51.





Here it can be seen how the second injection impacts almost exclusively, but nonetheless quite heavily, on the emissions of CO and soot only as seen in Figure 52 and Figure 53: these are rapidly decreasing as the DT between the first two pulses is shortened to increase again as quickly once the interval becomes too small, giving rise once more to U-shaped trends – not entirely visible though as the sweeps interrupt unfortunately right after the sweet-spots where the change in slope occurs.



Figure 52 – CO as SOI_{2nd} or SOI_{3rd} are advanced for quadruple patterns at 2000×120



Figure 53 – Soot as SOI_{2nd} or SOI_{3rd} are advanced for quadruple patterns at 2000×120

The sweet-spots represent the optimal timings at which the second pulse burns at its best avoiding any negative interaction either with the flame of the first or with the jet of the third pulse, keeping in mind that spraying liquid fuel on a burning flame causes the production of big amounts of soot and a bad preparation for the mixture that avoids the flame as well, with lots of CO at the end.

The third last injection keeps ruling soot and noise levels also in this case, as seen in Figure 54 but also making reference again to Figure 53.



Figure 54 – CN as SOI_{2nd} or SOI_{3rd} are advanced for quadruple patterns at 2000×120

9.12 Quadruple injection patterns at 2000 rpm/120 Nm

The figures below show the effects of advancing the entire sequence of four injections back in the compression stroke keeping fixed the DT between two consecutive pulses. To bring to an end the experimental effort required for the last section of this chapter and therefore of this project, Figure 55 lists the calibrations put under test in order to investigate quadruple-pulse strategies and eventually get a quite comprehensive view over the world of multiple injections.



Figure 55 – Set of calibrations under test for quadruple patterns investigations at 2000×120

As far as the BSFC is concerned, the trends in Figure 56 are increasing as seen with the triple scheme and the numbers keep getting higher for the same reasons provided in that case about penalties in combustion and thermal efficiencies due to adding a fourth injection almost midway through the compression stroke and to the fuel slip beyond the ECU computed quantity coming from the iterated actuation of a non-ideal injector and feeding system as a whole [5].

There isn't the prevalence of a calibration over the others.



Figure 56 – BSFC shifting all of the four pulses toward more advanced timings for quadruple patterns at 2000×120

The CO trend in Figure 57 loses its typical U shape for a decreasing behavior starting very high along the vertical axis compared to triple strategies – up to a multiplication factor of 5 – as an indication of the mixing issues stemming either from an overcrowded sequence of pulses suffering undermixing because of the too short DT making difficult the air utilization and from the fuel being delivered into in-cylinder conditions so adverse to ignition to cause the jets to travel across the combustion chamber for a time long enough to mix upon falling below the flammability limit.



Figure 57 – CO shifting all of the four pulses toward more advanced timings for quadruple patterns at 2000×120

Lots of CO are also formed from the rich pockets of extra-fuel spilling into the cylinder during the numerous injector' opening/closing transients [5]: the optimal calibration is a compromise between a sufficiently long DT between two consecutive injections to avoid undermixing and a not too large advance for the pilots – leading instead to overmixing – which is the unavoidable consequence for a satisfactory realization of the first requirement.



Figure 58 – Soot shifting all of the four pulses toward more advanced timings for quadruple patterns at 2000×120



Figure 59 – HC shifting all of the four pulses toward more advanced timings for quadruple patterns at 2000×120

The trends of soot and HC in Figure 58 and Figure 59 are very similar to those seen with triple injection schedules although values are somehow higher for the latter, probably because of a slight increase in wall impingement: regardless of the deployment of tinier jets, the longer spray penetration possible under such ambient conditions isn't completely compensated by the adoption of less inertial pulses thus causing the probability of impingement to increase, although very weakly. Even in this case there is not a winning calibration.



Figure 60 – CN and NO_x shifting all of the four pulses toward more advanced timings for quadruple patterns at 2000×120

Finally, noise and NO_x in Figure 60 remain flat during the sweep and at comparable levels for all the calibrations in the set under testing but a more noisy combustion is experienced with respect to triple-strategies probably because of the increased quantity of injected fuel and the shift of the heat release towards more advanced timings.

Also in this case there is not a best calibration prevailing over the others.

10. Conclusions

Multi-pulse injection strategies actually proved to dampen the main problems related to excessive engine-out concentrations of products of incomplete combustion and noise with minor penalties in terms of soot and NO_X whose abatement levels, encountered during the single-pulse PCCI optimization of the engine map, remain indeed unmatched: this is mainly due to the larger overall quantity of fuel that burns under premixed conditions thanks to the much longer ID required for charge preparation and to the absence of additional injection commands.

Subsequent-injections interactions in multi-pulse schemes determine indeed the degree of in-cylinder air utilization and the amount of premixed burn of the subsequent pulse as the *thermal atmosphere* [16] left by the combustion of the preceding pulse promotes with its high temperatures the ignition over increasing portions of the newly injected fuel, whose mass can't then accumulate and mix properly due to the reduced ID.

This leads to the formation of products of incomplete combustion and soot unless an optimal DT between two consecutive injections is established and mutual interactions are avoided allowing to achieve CO/HC/soot values very close to those of the single-pulse PCCI.

Noise intensity obviously lowers increasing the number of pulses in a multi-injection pattern while NO_X levels aren't actually affected much as their main factor of influence remains the EGR rate, always large.

Nevertheless, the fuel consumption slightly increases with the number of pulses deployed and at high speeds/loads not only the above described situation further worsens but also some *feasibility* issues arise: the system actuation isn't fast enough to cope with the ever reducing cycle time, making strategies with more than two or three pulses unsustainable or very difficult to calibrate and overall really unstable and not cycle-to-cycle repeatable, while the larger and larger quantities to be injected at each pulse, other than producing the same effect as well or causing the *hydraulic fusion* of subsequent shots of fuel, put again into play the problem of mutual interactions.

All in all, the triple scheme proved slightly better than double-split strategies although the comparison is confined to the 2000 rpm/120 Nm engine point and advantages might turn out not so remarkable at higher speeds/loads to justify the increased calibration effort and hardware capabilities required to deal with the series of inconveniences just exposed.

Also, the investigation on quadruple injection patterns has been done for sake of completeness as the level of complexity soon revealed barely sustainable for practical applications.

To conclude, the improvements offered by the use of more injection pulses in PCCI applications are substantial but not enough to allow the attainment of perfectly *dry*-*EGR* i.e. recirculated exhaust gas ideally containing negligible concentrations of products of incomplete combustion and particulate uncovered by deposits of soluble organic fractions.

The enormous recirculation required under PCCI conditions demands for a cooldown prior to the induction into the cylinder to guarantee adequate levels of volumetric

efficiency which translate into the engine's ability to intake air; the heat-exchanger devoted to such a task shall then be able to operate properly – without pressure and efficiency drops or the need for periodic mechanical regenerations – for the whole cycle-life of the engine/vehicle so as to reliably assist the calibration work of the ECU always striving for a higher combustion quality.

Even if the heat release proceeds relatively smoke-free, the continuously high flow rates inside the heat-exchanger make it prone to foul and clog anyway and the magnitude of the phenomenon amplifies as species like CO, HC and SOF (depositing on the carbonaceous skeleton of soot particles during and after the exhaust phase) are present, even moderately, in the exhaust stream since they promote the particulate sticking on the walls at high temperature thus accelerating the performance degradation of the cooler.

A possible solution to be investigated in future works consists in installing a mini-DOC immediately upstream the heat-exchanger in order to obtain something approaching the concept of dry-EGR at its inlet and avoid the risk of flow obstructions – at least for a longer amount of time – which would dramatically compromise the consistency of PCCI conditions, whose foundation are built indeed on the intensive use and control of EGR.

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