POLITECNICO DI TORINO

I Facoltà di Ingegneria

M.Sc. in Automotive Engineering “Propulsion Systems Development”

Master’s thesis

Effect of Hot Exhaust Gas Recirculation "EGR" on low load performances on PCCI diesel engine.

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“Life is a perpetual resistance to inertia trying to sabotage our deepest wills. Who are tired to will, want nothingness”

Friedrich Nietzsche

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Abstract

This thesis is the resume of my 6 months' engine calibration activity done at DENERG, the department of energy at Politecnico di Torino.

Thanks to that I learned to manage an engine test bed with all the instrumentation and help the team into post-processing activity of the data come from the tests.

This activity is focused on the study of how change the performances of unconventional combustion PCCI in diesel engine using hot EGR, so with by-pass of the EGR cooler.

To execute this research, we used a FPT F1C modified to exploit better all the advantages of the PCCI combustion and also to explore higher load, in facts with the conventional engine works in PCCI mode the maximum load reached was 2 bar BMEP, now we reach 8 bar BMEP.

In the first part of the thesis I explained all the characteristics of diesel engine, the pollutant formation mechanisms and all the after treatment devices.

Going more into details, I explained the PCCI combustion mode so the advantages and disadvantage of this new type of combustion.

However, the core of the thesis is the study of how the erg temperature affects the performances of the engine at low load, so how changes the pollutants emissions, the fuel consumptions, the torque demand and the noise if I by-pass the EGR cooler. This is done because during previously experimentation we found some problems about fouling of the EGR cooler at low load, where the exhaust recirculated speed is low.

I report in the last chapter all the results by engine speed vs BMEP scatter graph for each parameter analyzed.

1 The diesel combustion

The diesel combustion exploits the auto ignition of the fuel by compressing the mixture and so increasing the temperature to reach the point of ignition.

Therefore they cannot be mixed with air and compressed into the cylinder because otherwise the combustion process will start spontaneously during the compression stroke. The only way to control the start of combustion is to mix fuel with air just immediately before the desired start of combustion.

1.1 Diesel combustion phases

The diesel combustion has different phases:

- Ignition delay (AB)
- Premixed phase (BC)
- Mixing controlled phase (CD)
- Late combustion phase (DE)

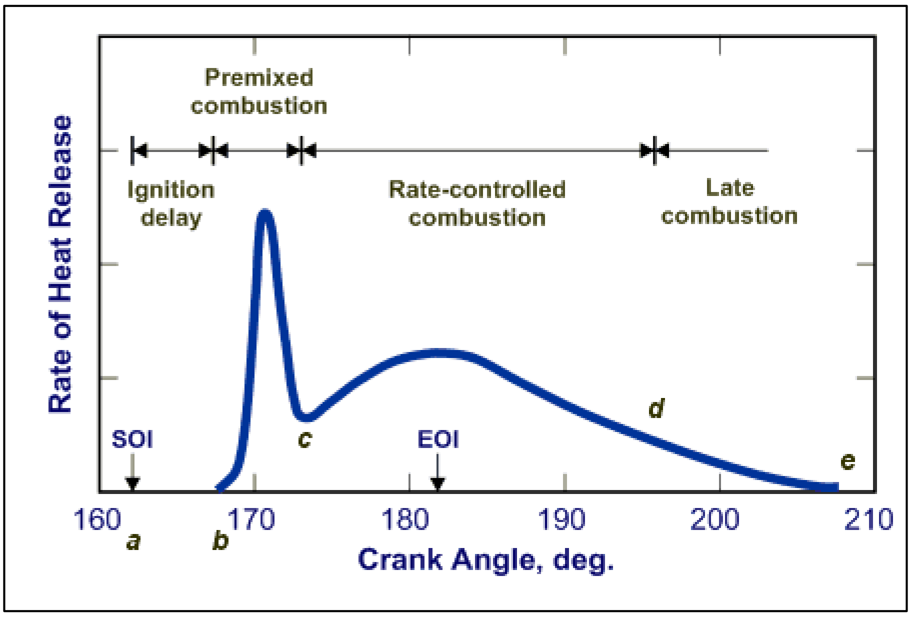

Figure 1 - Rate of Heat Release "www.DieselNet.com"

In the Figure 1 is showed the Heat Release Rate "HRR" where are present the four phases of the combustion.

The HRR is calculated starting from the pressure signal acquired inside the chamber.

These 4 different phases happen at very different thermodynamic conditions. Compared with the SI engine, where there are only 2 region, one behind the flame-front with burned gases, and the zone in front of the flame-front with fresh mixture, in CI is more complex because there is the presence of a fuel jet with a liquid core close to the injector nozzle, followed by a vapor plume which is mixing with air, and as soon as air and fuel are mixed, there is the start of the combustion process accompanied with the generation of burned gases; this hinders the understanding of when and where are the pollutants are formed.

1.1.1 Ignition delay

The ignition delay is the time between the SOI[1] and the SOC[2]. In this phase, it is caused by physical and chemical reasons.

The former, related to the fact that the fuel liquid jet has to undergo an atomization process, meaning that the liquid jet must be broken into smaller droplets by means of the aerodynamic interaction with the surrounding air, and then, the small droplets need to evaporate and mix with the surrounding air. The physical phase then is made of atomization, evaporation and mixing.

The latter is represented by chemical reactions, which are preliminary reactions that lead to the last step of the reaction change

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[1] SOI= Hydraulic Start Of Injection

[2] SOC= Start Of Combustion

1.1.2 Premixed phase

During the ignition delay, the fuel is accumulated inside the chamber and when the first droplet that have been introduced have finished its physical and chemical interaction the combustion is started.

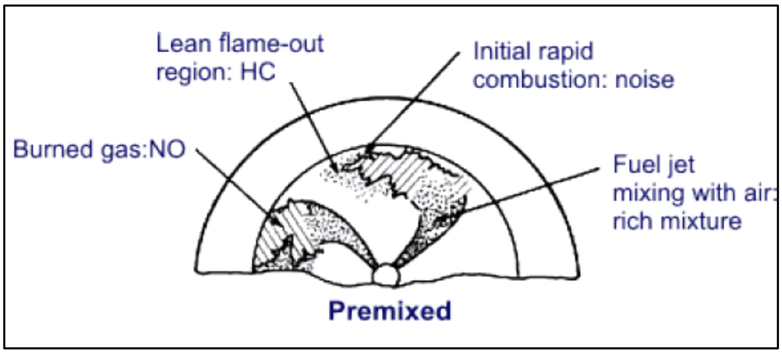

Figure 2 – Premixed phase "Spessa's notes"

When a first fuel mass fraction is burned, there is the increasing of the pressure and so also the HRR is increasing.
This energy release increases the temperature inside the chamber that accelerate the vaporization of the fuel and also the chemical reaction, so the ignition delay of the droplets injected after the first combustion is shorter.

This bring the accumulated fuel to burn all together creating high pressure first derivative so close to the ideal constant volume combustion.

This is good for the thermodynamic efficiency because how we know from the literature a constant volume combustion has high efficiency.
However, in terms of noise and NOx emission this is detrimental. For this reason, we search always to break down this peak in the HRR.

This phase is controlled by the ignition delay, more it is and more will be the fuel accumulated. So for reducing the peak in the premixed combustion we have to reduce the ignition delay.

1.1.3 Mixing controlled phase

After the premixed phase there is the mixing controlled phase in which the HRR has a moderate peak and the fuel burns with the burning speed which is related to the injection rate.

During this process, NOx continues to form in the high temperature region inside the chamber and also soot may form due to lack of oxygen in the combustion chamber regions, where there's extremely rich fuel concentration, with a local lambda below 0.5 (air-fuel ratio lower than 50% of the stoichiometric value).

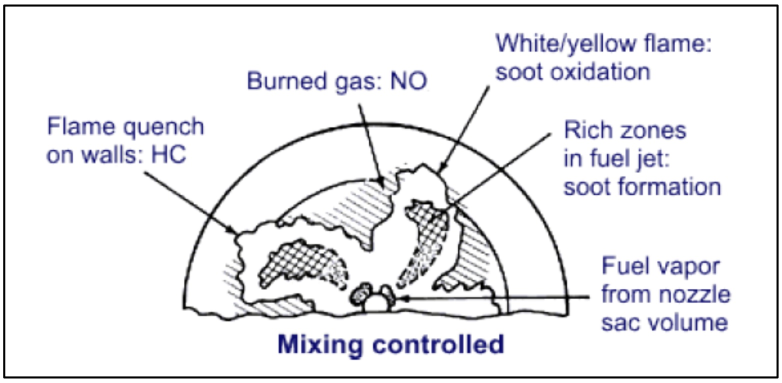

Figure 3 - Mixing controlled phase "Spessa's notes"

1.1.4 Late combustion phase

This is the last part of the combustion phase, where the combustion process continues and there will be air entrainment inside the plumes of fuel, air and burned gases that are developing during the combustion process. This involvement of new air in the combustion process is leading to soot oxidation, thus, part of the formed soot can be oxidized and the oxidation process will continue during the last portion of the combustion process, which corresponds to the ending tail of the HRR.

1.2 Spray model

For understating all the pollutant formation mechanism and where they are formed inside the chamber, is fundamental to know the spray model. In this way we can manage better the engine parameters that affect the pollutant formation.

For investigating the combustion inside the chamber, and so describe the new spray model they needed a specific instrumentation showed in Figure 4.

It consists in a single cylinder with replaced parts for inspection and mirrors to direct the laser jet.

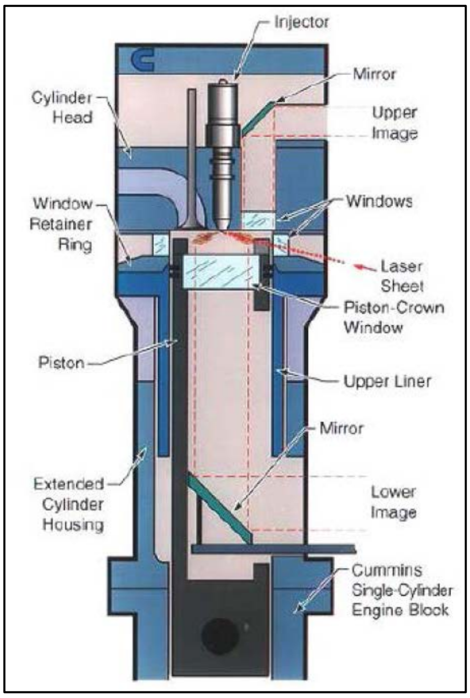

Figure 4 – Dec's single cylinder "Spessa's notes"

1.2.1 Old spray model

The model that I'm discussing below is completely wrong, and this is discovered thanks to the laser inspection.

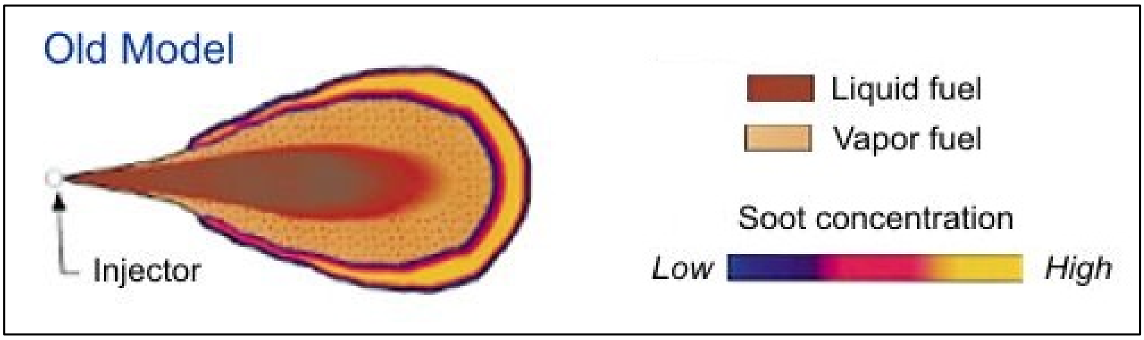

Figure 5 - Old spray model "www.DieselNet.com"

In the old model (Figure 5), the injection, mixing and combustion process were supposed to be as follows: The liquid jet was supposed to penetrate significantly inside the chamber, this was explained with a kind of wake effect.
The first droplets to be injected are pushed by the following injected droplets which benefit from the wake effect, so quite all the drag is kept by the first droplets and so the other droplets exploit the wake for penetrating more.

Therefore, the evaporation was supposed to start at the jet periphery were the first droplets were ending.

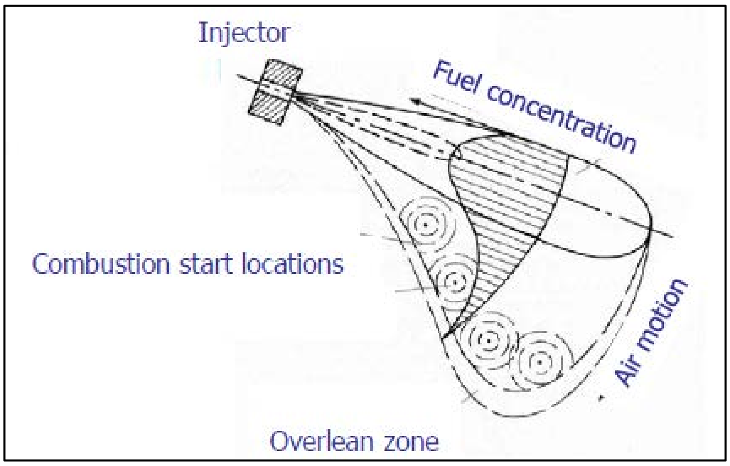

Figure 6 - Fuel distribution "Spessa's notes"

It was believed that the combustion process was going to start when fuel vapor was mixed sufficiently closed to stoichiometric conditions with the surrounding air, therefore, it was thought that the combustion starting location was at the jet periphery and from this shell-like region, the combustion was going to develop. It was also believed that the fuel concentration was following a Gaussian like distribution (see figure), with extremely high fuel concentration in the spray axis, with almost zero air to fuel ratio. Then, the air-fuel ratio is progressively decreasing from the spray axis towards the spray periphery, where the stoichiometric region is located, and then a region with just air

Actually, the soot formation process is typically occurring in extremely rich condition, with temperatures above 1300K, while in the jet core, were there is the extremely rich region, there is no combustion, because combustion was believed to occur just at the periphery, where composition was close to stoichiometric; thereby, from the concentration point of view, and according to the old model, soot was supposed to be formed in the central region of the spray, where the highest fuel concentration was present, but from a thermodynamic point of view, this is not possible because the center of the spray is relatively cold.
The old model was not capable to properly explain soot formation in a consistent way, and soot was supposed to be formed in the region close to the periphery where the temperature and fuel concentration were high enough to lead to soot formation.

All these hypotheses made with the old model was, successively not confirmed by the experimental analysis used for creation of the new model.

1.2.2 New spray model

The new model has completely difference behavior, the ignition actually occurs at multiple points across the downstream region of the jet. In normal burning, the liquid length is very short compared to the old model.

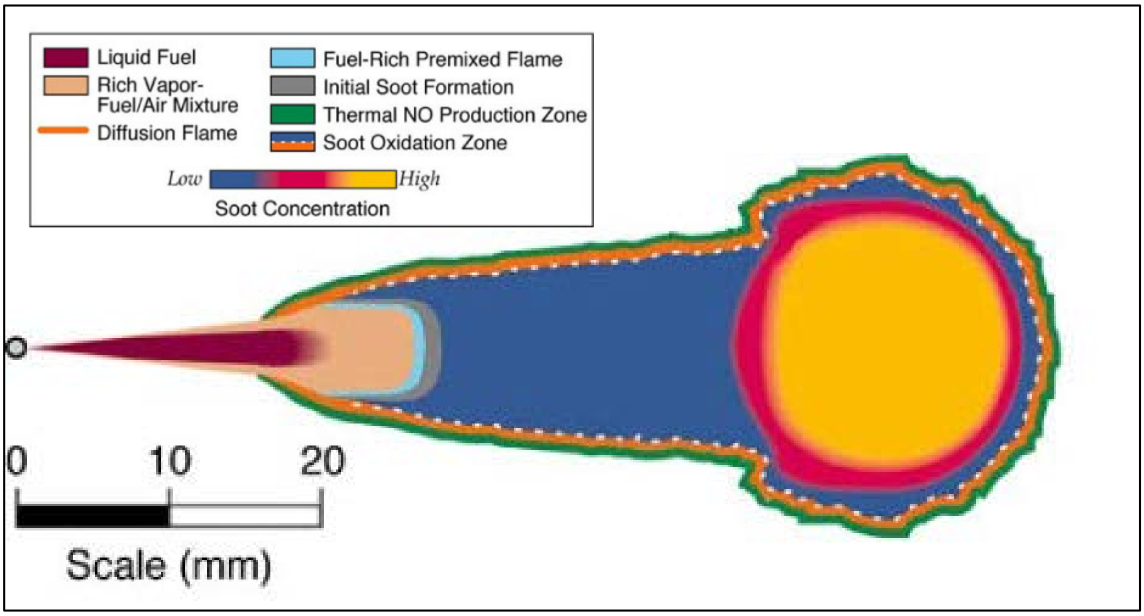

Figure 7 - Quasi-steady diesel combustion plum "Courtesy Dr. John E. Dec (Sandia national laboratories)"

In the new model, since the all model is different, also the pollutant formation will be different. The NOx is formed only where the temperature is high enough and also we have the sufficient oxygen availability so only on the diffusion flame.

The soot is formed at the end of the liquid length, where we have the first premixed combustion, in the middle of the plum we have the products of this combustion and also the creation of PAH that at the end of the combustion became soot, so the soot concentration is very high in the last part of the plum.

1.2.2.1 Spray evolution

For going more in the detail with the process of the combustion we can use the Figure 8 that represents the different physical and chemical processes that occur during combustion process and the figure has color representations showing what happens during the injection, mixture formation and combustion process.

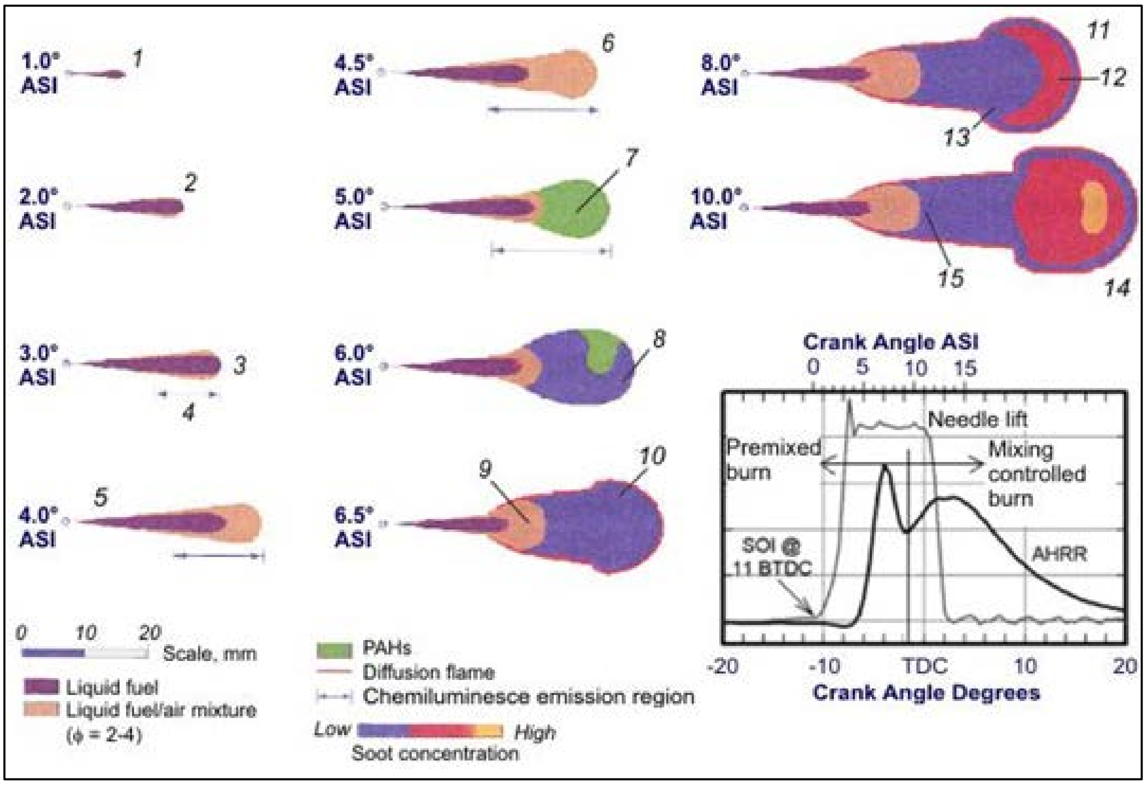

Figure 8 - Combustion plume evolution "www.DieselNet.com"

At the beginning we have a fuel injection that starts to propagate inside the chamber, the fuel jet will be over a cone surface, with the cone vortex in the injector nozzle.

The SOI is at 11°BTDC and the first evaporation is showed at 3°ASI, where in the picture is present a little plum in orange color. The liquid penetration becomes stable after 4°ASI and doesn't increase anymore, due to evaporation. It should be highlighted that the limited liquid-phase fuel penetration is not a result of combustion heating, it is due solely to the evaporation induced by entrainment of hot ambient air (heated by compression) into the fuel jet, therefore, is the vapor plume what continues to penetrate inside the combustion chamber. The liquid penetration depends on the droplets diameter and so, increasing the nozzle diameter we have also higher dimension of the droplets, these need more time to evaporate and at the end the liquid length is increased.

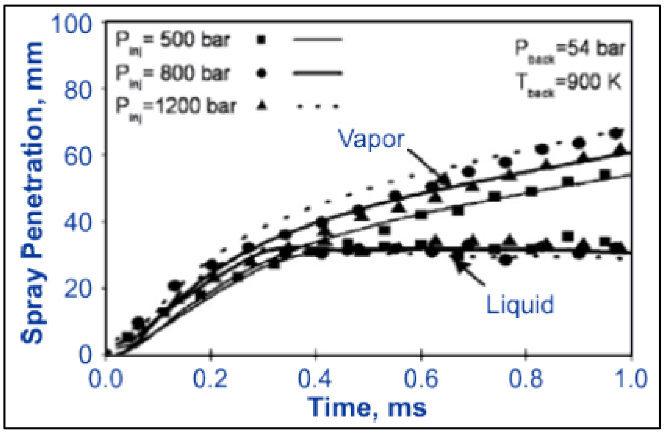

Figure 9 - Spray penetration “Spessa's notes”

Another parameter that we can take into account for the liquid penetration is the p_{rail}. From the left side graph is possible to say that the injection pressure doesn’t affect the liquid penetration but only the evaporation phase penetration. The higher the injection pressure, the higher the jet velocity, and the longer will be the vapor phase penetration; on the contrary the liquid phase will penetrate at the same distance regardless of the injection pressure.

Always on the liquid penetration we can check the dependence on the inside chamber temperature, for different charge density.

It can be seen that the liquid length penetration decreases by increasing the charge temperature. During cold start, some problems could be present because cold air will be intake and will be compress, leading to a lower combustion chamber temperature at TDC. The lower temperature will lead an increased liquid length penetration, and considering the typical size of the piston bowl for light duty applications (passenger cars diesel

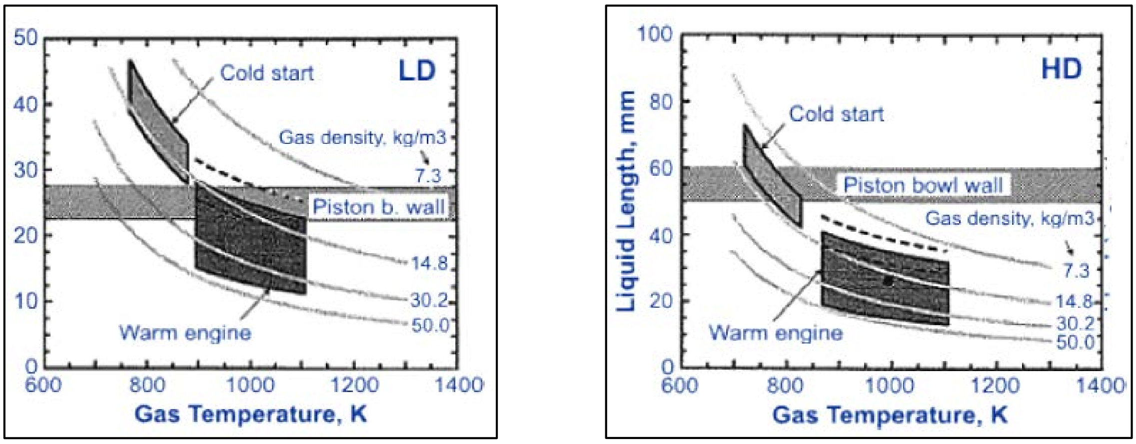

Figure 10 - Liquid length for LD and HD applications “[3]”

engine), it might be present a length larger than the piston bowl radius, hence, the liquid jet may impact on the bowls' wall. This is something that should be avoided because fuel impacting on wall produces extremely rich region that will have a delayed evaporation and will lead to soot formation. The liquid fuel hitting on the piston bowl walls is a problem that is present during cold start operation for heavy duty engines (HD), but can be significant, even during normal operation, in light duty engines (LD) which have a smaller combustion chamber.

In summary, liquid fuel penetration is limited by evaporation, independently from injection pressure, and injection pressure only leads to higher penetration for the vapor phase, but not of the liquid phase.

Another important point is that the combustion will start some crank angle degrees after the stabilization of the liquid spray penetration.
In this case, combustion starts at 5° after SOI, this value is measured by the detection of some chemical species "PAHs[3]" (green areas in previous figure) and with the use of the HRR diagram. The surrounding air is hot enough to make the fuel evaporate without the need for waiting for combustion.

Differently from what was believed with the old model, there's almost no region of pure fuel surrounded by a decrease in fuel concentration, but instead, there's a plume which is relatively uniform and quite rich, because has an air-fuel equivalence ratio $\lambda = 0.25$ to 0.5 (2 to 4 times richer than stoichiometric).

After the SOC, from 5° to 6°ASI, during the premixed burn peak of the HRR, there will be extremely rich mixture conditions, which means, favorable conditions for soot formation. Instead, there will be almost no NOx formation due to the lack of oxygen.

At 6.5° ASI, as shown in the figure, there's a large region in the plume's head where there's high soot concentration, and on the periphery there's the occurrence of the diffusion flame. In the diffusion flame, soot is going to be oxidized and fuel oxidation process will occur, leading to CO2 and water vapor, but at the same time, since there's

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[3] PAHs=Polycyclic aromatic hydrocarbons

oxygen available and high temperature, NOx is also going to be formed. This is basically what is going to occur during the mixing controlled burn phase. higher will be the premixed peak, higher will be the temperature that leads an higher production of NOx during onset of the diffusion flame, so for reducing the NOx we have to act on the premixed phase, also if in this part of combustion there isn’t a NOx formation.

It must also be considered that, at the end of the combustion process, the velocity of the last portion of fuel injected is progressively decreasing as the end of the injection phase and the closure of the injector are approached. This low velocity fuel may have problems in penetrating and reaching the leading edge of the fuel jet, where the combustion process is occurring, thus, this last portion of fuel may significantly contribute to soot formation because it may have problems to reach the region where soot oxidation occurs.

1.3 Design of engine

With the always more stringent regulations about the pollutant emissions, it’s very useful to spend a lot of time for optimizing the design off all engine components. In this thesis I will discuss only the part related to the combustion system.

The main parameters that affect the combustion are diameter “D”, stroke-diameter ratio “C/D”, compression ratio “rc” and the K-factor.

Before to start talking about these parameters we have to remember that the nozzle of the injector is placed on the center line of the cylinder and the combustion system is basically made up of a bowl (Figure 11). It can be seen that the fuel jets are usually directed into the cavity.

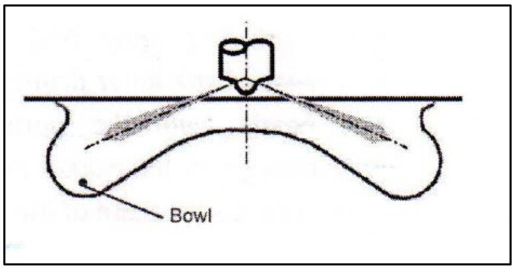

Figure 11 - Injection inside the piston bowl “Spessa's notes”

1.3.1 Piston Bowl

Using the piston bowl as combustion chamber and also an injector centered can have some benefits:

- With the piston bowl as a combustion chamber, there is the possibility to have a flat cylinder head surface, which increases the mechanical strength of the combustion system, that is important, especially for diesel engine, in which we have higher peak firing pressure, especially at high load;
- Having the injector centrally located, and with the jet radially entering the combustion chamber, allows an axial-symmetric fuel injection, which allows the chamber geometry to be well utilized in order to generate a gas motion that supports the combustion process.

There are at least two types of combustion chambers for an in-piston bowl combustion chamber:

- Open combustion chamber;
- Re-entrant combustion chamber

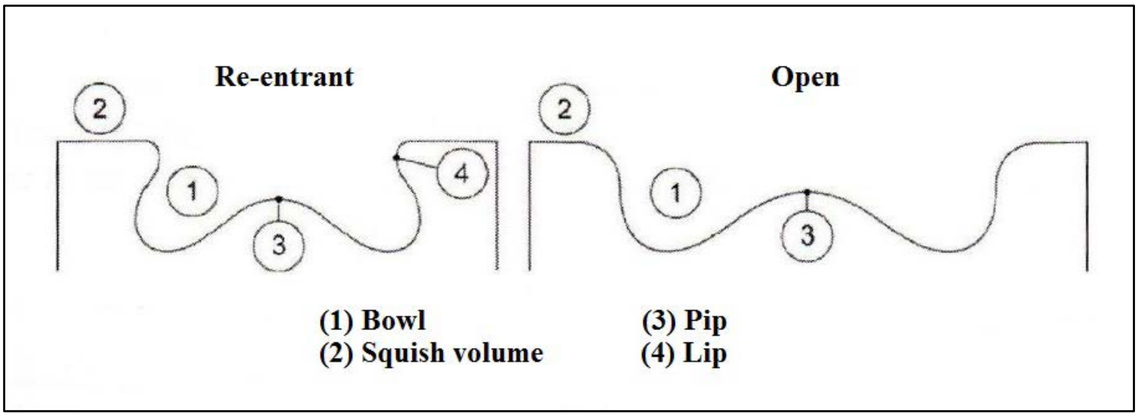

Figure 12 - Piston bowl types "Spessa's notes"

The squish volume (2) plays a role that can be positive or negative:

- *Direct squish*: during the compression phase, very close to the combustion event, there is some air that is pushed from the squish region inside the cavity and this macro vortex delivers kinetic energy to the turbulence flow field, generating micro vortex that can promote the combustion event itself;
- *Reverse squish*: the piston moves down during the expansion, there is air that is extracted from the bowl and goes back to the squish area. This can be positive if there is soot inside the bowl, because extracting it there is more possibility to have a soot burnout thanks to the high oxygen availability, or can be negative, if there is the extraction from the bowl a part of the combustion plume that is still burning, taking it in a colder area that can create bulk quenching.

The other geometric surface is the pip (3) that is used for having a more strength of the piston because the combustion occurs in the periphery of the bowl, so there isn't need to have space for the mixture in the center.

The Re-entrant bowl is used more in LD engines[4] because it can be optimized for low and medium load, that are the point in which the engine works under the type approval procedure.

We know that these engines work with short and multiple injection, so we have no significant turbulence related to the spray.

The turbulence created by the injection is close to the injection tips, because the spray has a relatively short penetration and decays quickly during the combustion event. After the end of the injection, when you still have the combustion event, there is no significant turbulence related to the spray, and in case it is present, it isn't located where the combustion happens. This means that the spray cannot deliver the kinetic energy required for driving the mixing process, because is in the wrong time and position. This kinetic energy can be delivered by the combustion system, that have to be capable of promoting the mixing during combustion, therefore, also during late cycle mixing.

In the case of HD engines[5] the situation is completely different because they usually work at high loads meaning that the injection phase is longer than LD engines and the combustion occurs, very often, at the same time of the injection, and lasts after the injection so the spray can deliver the kinetic energy exactly when it is required.

For this reason it is preferable to use an open combustion chamber also because eliminating the lips means eliminate a sharp edge that increase the mechanical strength of the piston that is subjects to high load.

Another important point is that, considering that the combustion chamber has to support the mixing and the turbulence creation in less way, so we can have a chamber with a reduced swirl level.

This is beneficial for the volumetric efficiency that will be higher .

In addition the reduced swirl level has also a positive impact on the heat transfer losses that will be lower due to the reduction of gas motion.

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[4] "Light Duty" engines= all the engines mounted on passenger cars and vans.

[5] "Heavy Duty" engines= all the engines mounted on the truck, buses and coaches.

1.3.2 Piston diameter "D"

In this chapter I'm discussing all the features regarding the changing of the bore[6] "D".

Having a larger bowl has 2 positive effects:

- Better smoke limit and enhancing the power density;
- Reduced wall wetting phenomena.

Analyzing the first aspect, having a larger bowl we have a better air utilization that improve the smoke limit. This is related to 2 facts:

- The distance between injector and wall is longer with a larger bowl, so at high load, where the injection pressure is higher and also the injection is longer, the combustion occurs far from injector tips and having a larger bowl can help to have the air in the right position.
- Having the larger bowl increase the K-factor that is positive for having good air utilization. Reminder the definition of K-factor as follow:

$$K - factor = \frac{V_{bowl}}{V_{bowl} + V_{head} + h\pi\frac{D^2}{4}} = \frac{1}{1 + \frac{V_{head} + h\pi\frac{D^2}{4}}{V_{bowl}}}$$

The improvements of the air utilization doesn't have a positive effect on the smoke limit but also in the enhancing the power density of the engine as we can see in Chart 1:

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[6] Bore D= piston's diameter

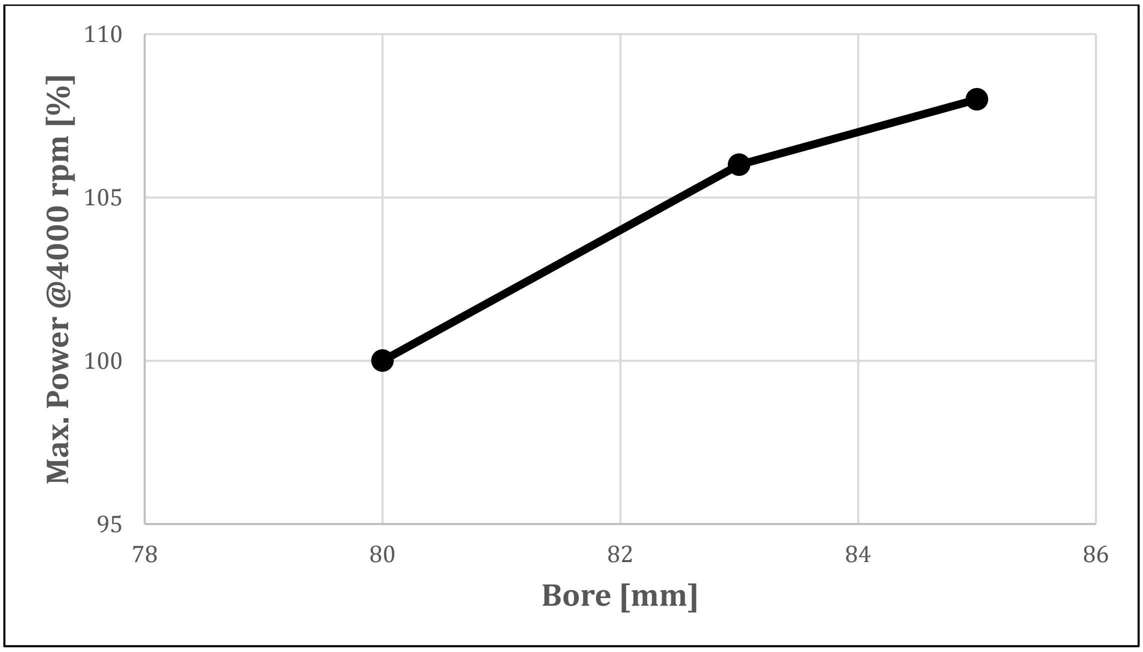

Chart 1 - Bore Vs. Max. Power

At rated speed we can see how the maximum power change, the graph suggests increasing the diameter D and as a consequence the diameter of the bowl have a great impact of the maximum power.
However, a certain point the increasing is derate due to the reduction of the ratio C/D that has a negative effect on the K-factor.
Anyway, larger bores achieve higher power density.

Talking about the second positive effect, the reduced of the wall wetting phenomena helps to reduce the HC emission at cold start because in cold condition the penetration of the liquid phase is longer and therefor it is clear that can impact on the wall. If the bore diameter is higher, we have more distance between the injector tips and the walls that allows to reduce the impingement.

Other advantages of larger bowl diameters are reported here:

- More tolerant to advanced injection timing;
- Help prevent oil dilution;
- Tend to increase the tolerance of the combustion system to variations in spray targeting;
- Rend to reduce the heat load on the piston due to a more favorable surface-to-volume ratio;
- Higher mechanical strength.

All these advantages are done increasing the bore diameter without taking care about the other constraints present during the engine design.

1.3.3 K-factor and stroke to bore ratio "C/D"

For talking about this 2 parameters we have always change the bore diameter but now, we consider also constraints.
So, increasing the bore diameter and fixed the engine displacement we have to reminder the relation between the stroke "C" and the bore "D":

$$iV = i\pi \frac{D^2}{4} C = i\pi \frac{D^3}{4} \frac{C}{D}$$

We can see that if we increase the bore D and we keep V fixed, we need to decrease the stroke to bore ratio C/D.

Now we add another constraint that is the fixing of compression ratio expressed with the following formula:

$$rc = \frac{V + V_m}{V_m} = 1 + \frac{V}{V_m} = 1 + \frac{V}{V_{bowl}} \cdot "K - factor"$$

Where K-factor is defined as:

$$"K - factor" = \frac{V_{bowl}}{V_{bowl} + V_m}$$

And the dead volume express as V_m

$$V_m = V_{bowl} + h\pi \frac{D^2}{4} + V_{head}$$

This dead volume is composed by 3 contributions and we can assume that it remains constant.

Express the K-factor as follow, we can notice that increasing the bore diameter the K-factor is reduced, because we have to reduce the V_{bowl} to keep the V_m constant

$$"K - factor" = \frac{V_{bowl}}{V_{bowl} + h\pi \frac{D^2}{4} + V_{head}}$$

At the end, if we want to increase the bore D, we have to change the stroke to bore ratio and at the same time, we have to reduce the K-factor, if we want to take fixed the compression ratio and the engine displacement (usually these 2 parameters are fixed because come from the performance requirements).

So, reducing the K-factor has a negative impact on the air utilization, instead to have the air inside the combustion bowl, the air moves in another region like the squish area.

Another impact that we may have is on the BSFC[7] because the friction usually scale as a function of C/D, it is reasonable to expect that by increasing the bore D, due to the reduction of the ratio C/D, we will also have a reduction of FMEP[8] that leads a reduction in BSFC.

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[7] BSFC="brake specific fuel consumption" expressed as follow: $bsfc = \frac{Fuel\ consumption\ rate\ [g/_s]}{Power\ [W]}$

[8] fmep="friction mean effective pressure" expressd as $fmep = pmep + rmep + amep$

In Table 1 is present an example of increasing the bore D on a diesel engine

Bore [mm]	*80*	*83*	*85*
ISFC [%]	*100*	*100*	*100*
FMEP [%]	*100*	*94*	*88*
BSFC [%]	*100*	*99*	*99*

Table 1 – Effect of changing bore on BSFC

The impact of the FMEP is more effective a part load.

Finally, the change in the bore to stroke ratio C/D has also an impact on the turbulent fluctuations u' because it has the following dependence:

$$u' \propto (C/D)^{2/3}$$

So, the decrease of C/D has also a decreasing on the turbulent fluctuations u' that has negative effect on the combustion because it reduces the importance of the turbulence generated by the piston with respect to the turbulence generated by the spray, especially in the case of LD diesel engines.

Where pmep="pumping mean effective pressure"

rmep="rubbing mean effective pressure"

amep="auxiliary mean effective pressure"

Every mean effective pressure is evaluated by $p_{me} = \frac{W}{V_d}$

1.3.4 Compression ratio “rc”

Now I would discuss the effect of increasing the compression ratio rc.

Increasing the compression ration increases the ideal efficiency of the engine. The advantages of a high geometric expansion ration may not always be achieved, however:

- We might have some practical limits related to peak firing pressure or NO_x because an higher PFP can produce a problem in the mechanical resistance of the engine.
  In order to control the NO_x emissions the combustion should be delayed, thus reducing the effective expansion ratio, since, delaying the combustion means also delay the expansion. This has a negative efficiency on the efficiency.
- If we have higher pressure due to the higher rc, we also have higher FMEP because there are some terms that depends on the pressure inside the chamber. This reduce the efficiency of the engine.
- Increasing the rc, we usually have a reduction of the S/V at the TDC and this is detrimental in terms of heat losses.
- With higher rc has negative effect on the unavailable air as is showed by the figure

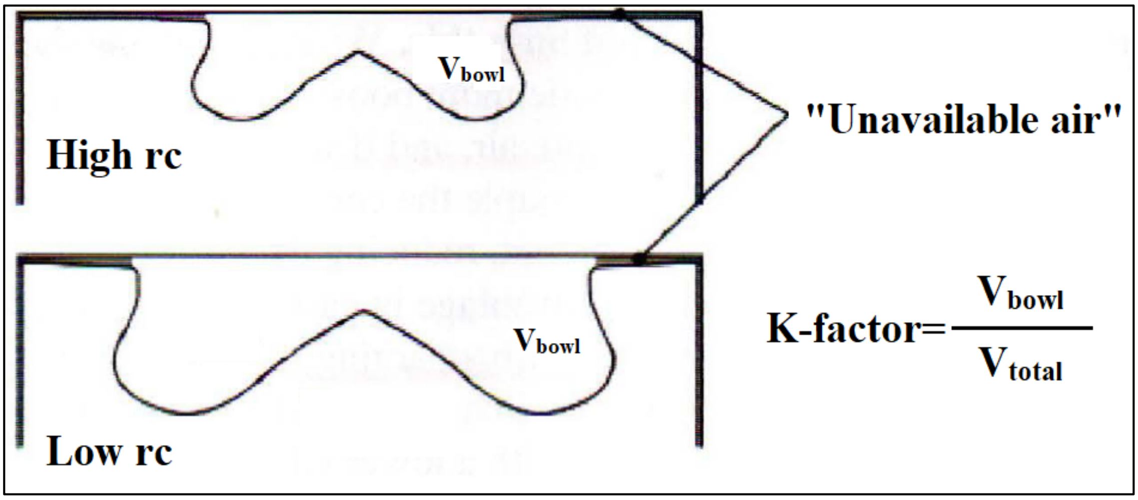

Figure 13 - Bowl shape as a function of rc

Instead of increasing the compression ratio, now I discuss the advantage of *reducing the rc*:

- It allows greater specific power at a given peak firing pressure because there is the possibility to advance the SOI and also increase the boost that leads and increasing of air available in the intake manifold.
- It lowers engine-out emissions by lowering compression temperature because a reduction of rc from 17:1 to 14:1 will decrease the near-TDC compression T by approximately 50 K). This is favorable for the NO_x control because we don't need to further delay the combustion process if we have already reduced the compression ratio.
- Due to lower expansion, exhaust gas T are generally higher:
    a) More energy extraction by the TC[9], and higher boost levels can be achieved at low rpm, improving the low speed torque characteristics;
    b) The efficiency of the DOC is higher with higher exhaust gas temperature. However, the production of the HC and CO is higher due to the higher ignition delay, so there is a sort of compensation;
    c) The catalytic light-off time is reduced;
    d) The slower cooling during expansion provides more time for oxidation of soot and other products of partial combustion;
- The increasing of ignition delay and the slower of the cooling during expansion can leads of reaching the soot emission targets with lower injection pressure. If we reduce the injection pressure, we also reduced the power needed by the injection system so the BSFC is reduced too.
- K-factor can be increased because the volume of the bowl is increased like the figure

[1][2][3]

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[9] TC=Turbocharger

2 Pollutant and noise formation mechanisms

In this topic I would to discuss the formation mechanism of all the pollutant produced by the combustion as NO_x, CO, PM and HC. In addition, also the noise will be treated in this chapter because it is something that we would to avoid, for having a more comfortable for the driver, and also because it is related to the pressure inside the chamber.

2.1 NO_x formation mechanism

The NO_x is formed during the diffusion burning but the premixed burning has a relevant role on the production because higher peak of the pressure means higher temperature that leads a more suitable environment during the diffusion burning for the NO_x formation.

The use of pilot injection can reduce the ignition delay of the main injection, reducing the amount of fuel that burns all together in the premixed burning as well as the injection rate shaping that means to inject a lower amount of fuel during the first portion of the injection process and then, injecting fuel with a higher injection rate later.

Once NOx are formed, unfortunately, the quite rapid expansion which occurs in the combustion chamber leads to a sudden temperature decrease, thus, the NOx concentration remains more or less frozen as shown in Figure 15.

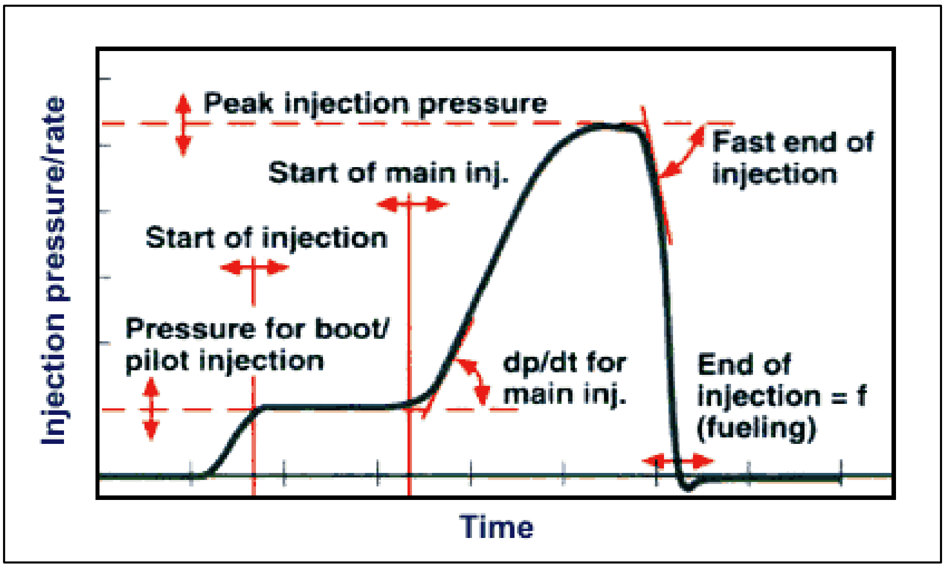

Figure 14 - Injection pressure shape “www.DieselNet.com”

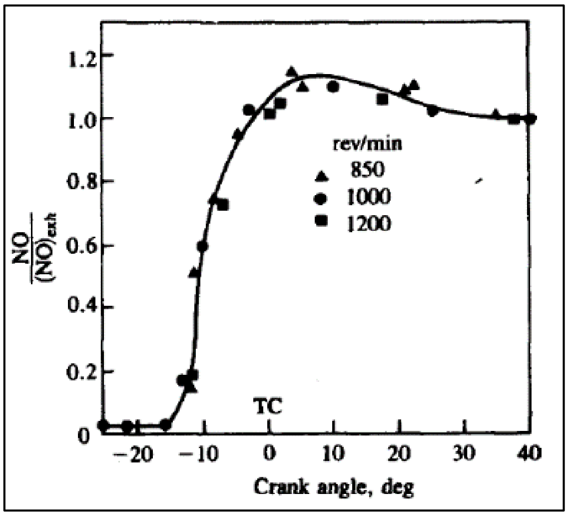

Figure 15 - NOx rate Vs CA “Millo's notes”

I want to underline that in the CI is very important the NO_2 fraction, because can affect the efficiency of some catalytic converters.

2.1.1 Engine parameters that affect the NO_x formation mechanisms

For analyzing how the NO_x formation change in the engine, we have to focus on the impact of the engine parameter as:

- Air-fuel ratio
- Injection timing
- EGR rate

2.1.1.1 Air-fuel ratio

Looking at the graph NO_x concentration Vs. equivalence ratio[10], there is a significantly different trend of what was observed for SI engines.

In diesel engines there is a broader range of variation of λ values but on the shown figure the average value is used. It must be considered that inside the combustion chamber there is a wide variety of air-fuel ratio values.

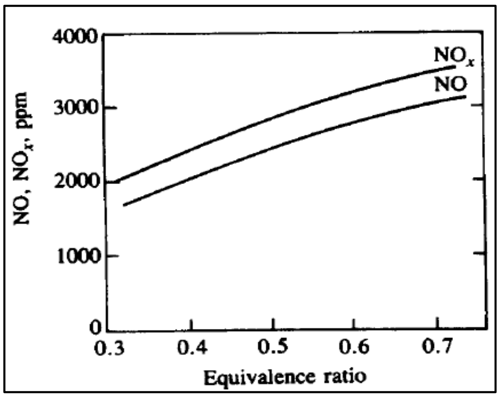

Figure 16 - NOx Vs Equivalence ratio “Millo’s notes”

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[10] Equivalence ratio $\Phi = \frac{Fuel}{air}$ so it is rich $\Phi > 1$ and lean $\Phi < 1$

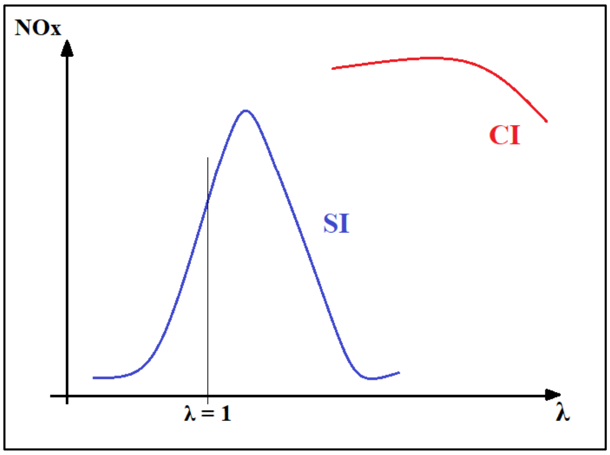

Figure 17 - NOx Vs λ compared SI and CI engines "Millo's notes"

NOx emissions for CI engines decrease with increasing λ because when the average A/F ratio increases there will be, on average, lower temperature, thus lower NOx formation. But as seen in the figures, the sensitivity to air-fuel ratio is different from the one of SI engines; this is due to the uniformity of the λ values for SI engines.

In the CI engines we have zone with very rich mixture and other zone of very lean mixture, so the average value of λ doesn't represents a right distribution inside the chamber, it can give only an information about how much fuel is injected, so increasing the fuel the NO_x emission will be higher. For this reason, at higher load, where we injected more fuel, so the temperature will be higher, also the NO_x emission at engine out will be higher

2.1.1.2 Injection timing

Now I'm discussing about the change of NO_x production changing the SOI.

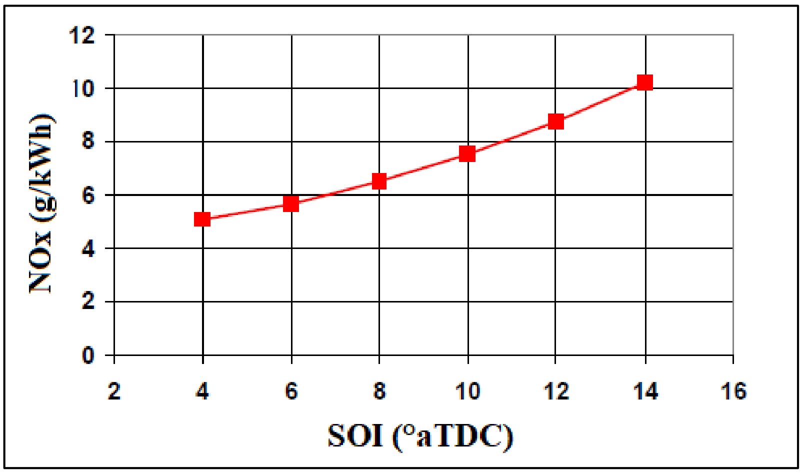

Chart 2 - SOI Vs NOx

The more advanced the injection is (with respect to the TDC), the longer the ignition delay will be. This happens because we start to inject fuel in an environment with lower air density and also the temperature is lower, so the atomization process will be longer.

If the ignition delay increase, we accumulate more fuel inside the chamber that will burn all together, creating an high peak in pressure and temperature. This is detrimental for the NO_x formation that increase the pollutant at engine out.

The Chart 2 show how the NO_x specific emission is so sensitive to injection timing. We can see that passing from a 4to 14 °aTDC, in these particular operating conditions, NOx emissions are doubled moving from 5 to 10 gr/KWh.

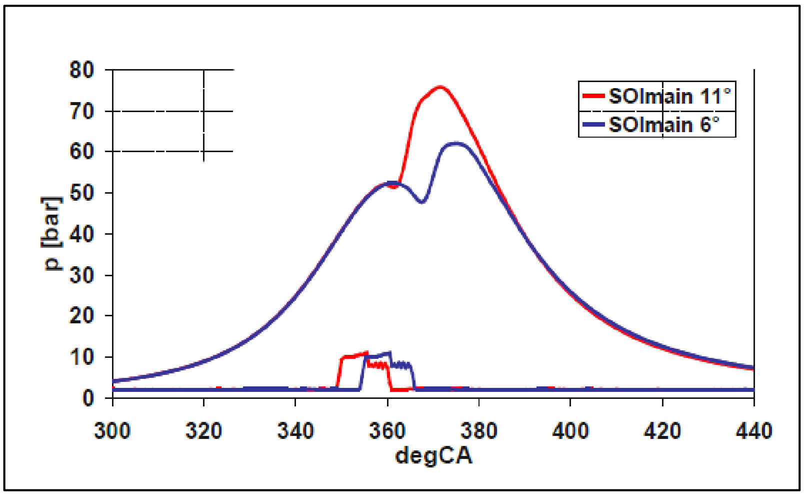

Figure 18 – Pressure signals as a function of SOI "Millo's notes"

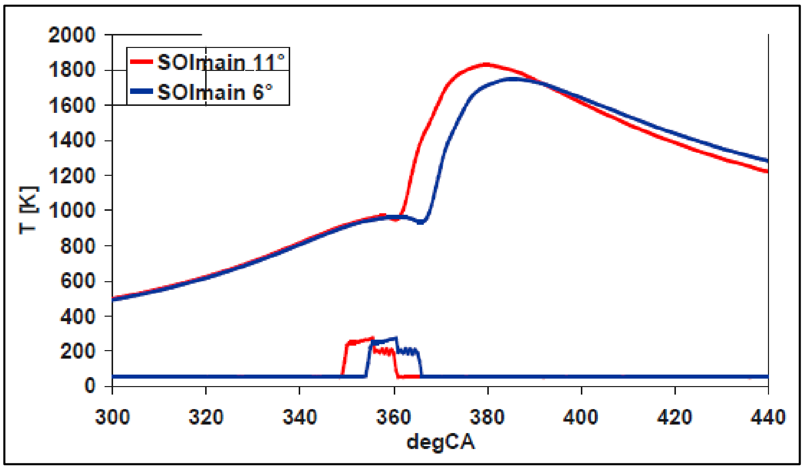

Figure 19 – Temperature signals as a function of SOI "Millo's notes"

This can be understood looking at the average temperature and pressure inside the combustion chamber.
In Figure 18 and Figure 19t here are pressure and temperature signals compared between two different timings, 6° and 11° before TDC.

By injecting 6° before TDC the ignition delay is significantly long and the combustion process will occur in the expansion stroke well after TDC, thus, combustion will start to develop when the pressure has already decreased significantly. Combustion will produce an increase in pressure, but the peak pressure reached is lower if compared with a more advanced timing as 11° before TDC; as seen, the pressure will start to rise pretty close to TDC and will reach a higher level. Obviously, if the same amount of fuel is injected in these two different conditions, it is expected that the red curve (11°) will produce more work because there's higher pressure level for a significant portion of the expansion stroke.

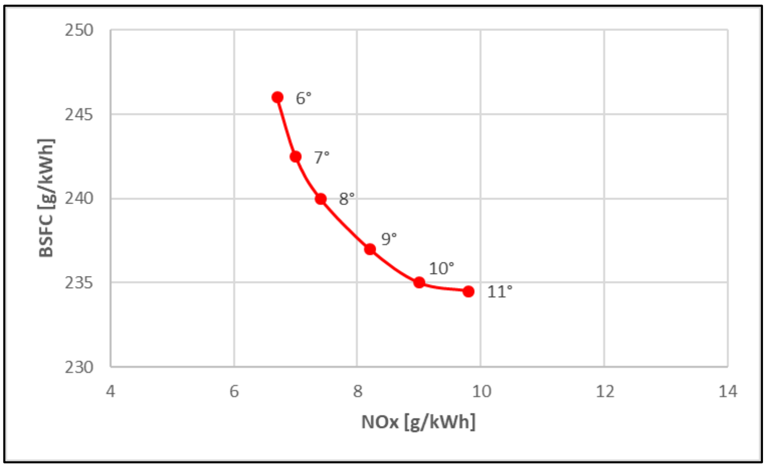

Chart 3 – NOx Vs. BSFC

Comparing these two points in terms of BSFC (Chart 3), this will be lower for the more advanced timing (11°) and higher for the more retarded timing (6° before TDC). If more advance is applied there's more favorable phasing of the combustion process with a lower fuel consumption, but accompanied with higher pressure and temperature that will lead to higher NO_x. By retarding the injection, peak temperature and pressure are reduced but there's less favorable combustion process leading to higher BSFC.

Similar remarks can be made for particulate matter (Chart 4), PM, increasing the injection advance, more fuel burns in the premixed phase, with better chances to be later oxidized, meaning that, a lot of soot will form, but there is also a good amount of time for the soot

burnout. On the other hand, if the timing of injection is retarded, the peak of the premixed phase will be smaller with lower NOx formed, but higher soot emissions.
Again, a trade-off diagram is obtained by plotting PM versus NOX specific emission and moving along the curve by changing the injection timing, the same hyperbolic trend is found.

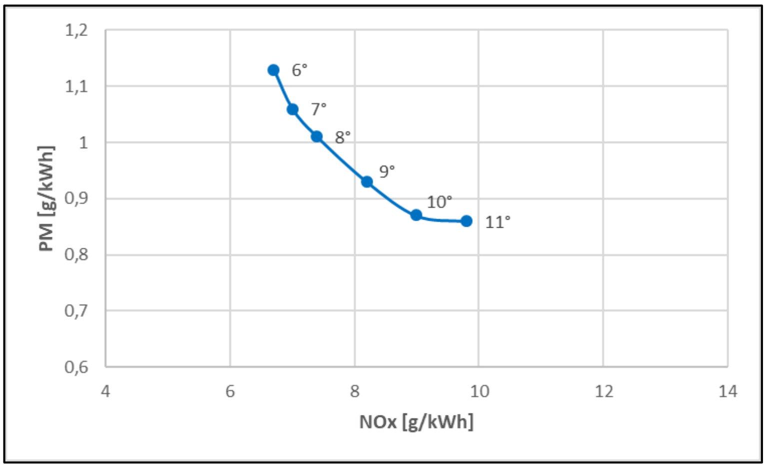

Chart 4 – NOx Vs. PM

There is never an optimal timing, it depends on the target, which could be, the reduction of NOx or the maximization of the performances, but it also depends on the after treatment technology available.

2.1.1.3 EGR rate

Burned gases behave as dilatants that absorb heat, contributing to cool down the peak temperature of the combustion process, thus, hindering NOx formation.
A diesel engine operating at part-load is the ideal engine for the use of the EGR, because there's no throttling, therefore, the engine intakes the same amount of air that intakes at full load, meaning that the engine will have a high volumetric efficiency; filling all the combustion chamber with air and with a small quantity of fuel.
The high amount of air in the combustion chamber is detrimental for NO_x formation due to the high oxygen availability, hence, replacing a part of the in-cylinder air with burned gases that do not carry any oxygen.

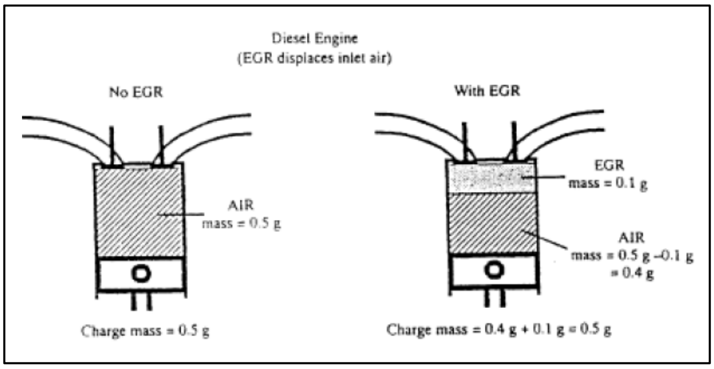

Figure 20 – EGR volume "Millo's notes"

For measuring the EGR rate in terms of concentration by volume, it is possible by simply making the ratio between: the CO2 measured in the intake system, after mixing the EGR with fresh air and, the CO2 measured in the exhaust system.

$$EGR\ [\%vol.] = \frac{CO_{2int\%}}{CO_{2exh\%}}$$

Now, I'm going more in details in the mechanism which make EGR so effective in reducing NO_x emissions.
There are 3 main mechanisms: dilution effect, thermal effect and chemical effect.

1 *Dilution effect*: it's the effect due to the replacement of some fresh air with the dilutant, inert. Considering a unit volume of mixture in which there must be a certain amount of fuel and a certain amount of air to burn this fuel. If air is mixed with burned gases, this will mean that each oxygen molecule will not come only with 79/21 molecules of nitrogen, but will also bring together other nitrogen, other CO_2, other water vapor, that are products of combustion. All these molecules share the energy released by the fuel combustion and, since the energy released by the fuel during combustion is always the same, the increase in temperature will be lower if this energy is spread among a higher number of molecules, due to the presence of this inert molecules.

2 *Chemical effect*: due to dissociation of CO_2 and water vapor. At high temperature, CO_2 dissociates into CO and water vapor dissociates into OH groups. Both these reactions are endothermic, so they absorb heat. If combustion products containing CO_2 are routed back, the dissociation of CO_2 will contribute to absorb heat and cool down the temperature, further reducing the NO_x emissions.

3 *Thermal effect*: it's almost negligible and it is due to the higher specific heat of burned gases in comparison to air. The specific heat of CO_2 and water vapor is higher than the specific heat of air, hence, the recirculated burned gases are absorbing more heat per unit of mass than air. The amount of CO_2 recirculated is very small, so the thermal effect is negligible. Reminding the example made at the beginning, considering an engine operating with A/F ratio double than stoichiometric, the concentration of CO_2 in the exhaust is about 6%. Even if the 50% of the exhaust is recirculated to be mixed with air, there will be, into the

intake, a mixture with only 3% of CO_2 and, even if CO_2 has a specific heat 20% higher than air, there will be 97% of other species such as nitrogen or oxygen and just 3% of species with higher specific heat such as CO_2.

I can use a synthetic EGR for separating the 3 effects and look which is the most important.

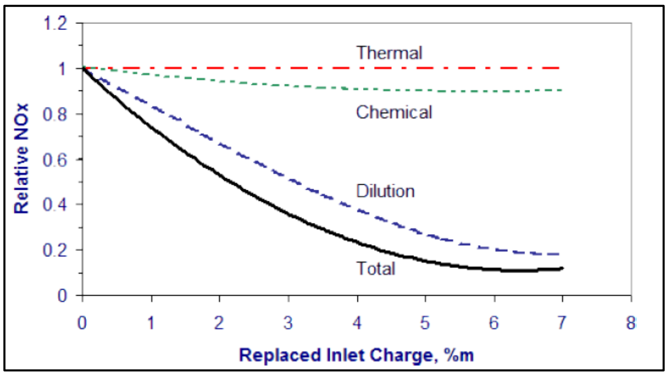

Figure 21 – Rate of NOx reduction depending on different phenomena “Millo’s notes”

Figure 21shows what is get when some dilutants are recirculated, plotted as a function of the fraction of the replaced inlet charge by percentage of mass. Attention must be paid because percentage by mass and by volume are extremely different in this case because, when exhaust gases are recirculated they are hot so have very low density in comparison with fresh charge.

$$d[NO]/dt = [6 \cdot 1016/T^{0.5}]\ x\ exp(-69090/T)\ x\ [O_2]^{0.5}\ x\ [N_2]$$

			O_2=23.3%	T=2280K
O_2 [%]	Peak Temp [K]	d[NO]/dt	d[NO]/dt	d[NO]/dt
23.3	2280	1.00	1.00	1.00
22.1	2220	0.44	0.45	0.97
19.8	2160	0.18	0.19	0.92
17.5	2035	0.02	0.03	0.87

Table 2 – d[NO]/dt as a function of O_2 and T

It is possible to understand this remembering the typical relationship that gives the reaction rate for a chemical reaction, such as NO formation. The reaction rate is obtained by the Arrhenius law (shown below in Table 2), where there is the product of the concentrations of the reactants, in this case, concentration of oxygen and nitrogen.

Since this is the stoichiometric proportion between oxygen and nitrogen, it can be seen that oxygen concentration is the square root, but, what is really important is temperature, since the reaction rate is related to temperature by means of an exponential reaction. A small change in temperature dramatically changes the reaction rate. To better understand this, you can compare the different results obtained in Table 2, changing the temperature and the oxygen availability.

However, it should be always remembered that, in a diesel engine, differently from SI engines, not all the EGR were made equal. Diesel engines may be operated with very different values of the A/F ratio, from the stoichiometric ratio up to 5-6 times leaner than stoichiometric.

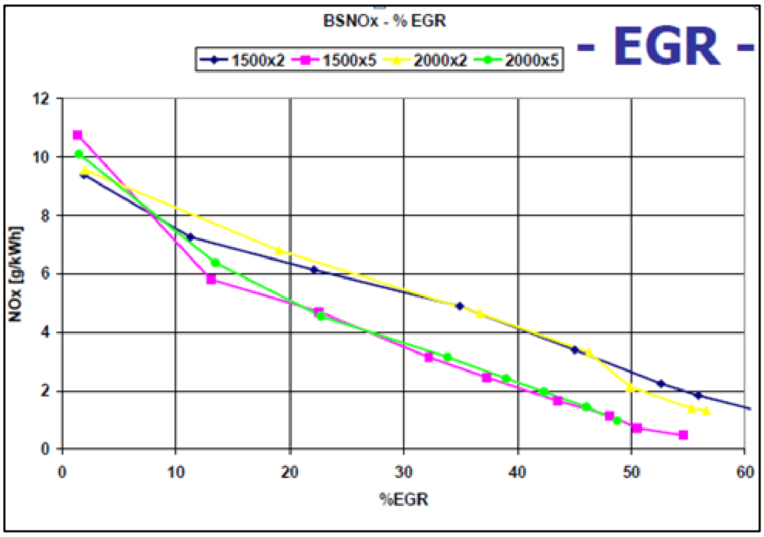

Chart 5 - %EGR Vs. NOx

Taking a very high A/F ratio, for instance, 6 times the stoichiometric A/F ratio, in these operating conditions, exhaust gases will contain a small amount of burned gases and a huge amount of air, since there is an excess of air. What it's called "recirculated exhaust" is actually a mix of a small portion of burned gases, which are dilutants and a huge portion of air, thus, it's not a true dilutant and it cannot be expected that is as effective as EGR when close to stoichiometric, that is when most of the exhaust gases are burned gases, since there's a very few amount of excess air. This means that the same EGR rate, at different engine loads, i.e. different A/F ratios, will produce different results. Chart 5 shows the NOx emissions versus EGR rate for different operating points, which are typical of the NEDC, they are: 1500 rpm at 2 and 5 bar of BMEP, and 2000 rpm with 2 and 5 bar BMEP. It is possible to observe that the blue and yellow curves, that are both at 2 bar BMEP, behave in the same way; the same can be said for the green and the purple curves, that are both at 5 bar BMEP.

There are also some drawbacks, if the charge is diluted, the oxygen availability is reduced, therefore, the oxidation of soot will be more difficult. By increasing the EGR rate the soot emissions will increase, but the soot formation does not change so much. The soot burnout (or soot oxidation) is affected by EGR and this will lead to a higher soot emission.

2.2 PM formation mechanism

Particulate Matter - usually abbreviated as PM - is a complex aerosol system. Contrary to gaseous diesel emissions, PM is not a well-defined chemical species. The definition of particulate matter is in fact determined by its sampling method. PM sampling involves drawing an exhaust gas sample from the vehicle's exhaust system, diluting it with air, and filtering through sampling filters. The mass of particulate emissions is determined based on the weight of PM collected on the sampling filter. It is quite obvious that any changes in the procedure, for example using a different type of sampling filter or different dilution parameters, may produce different results. Particulate matter, as specified by most standards and regulations worldwide, is sampled by filtering diluted diesel exhaust at temperatures not higher than 125°F (52°C). This cooling effect is typically achieved with laboratory dilution ratios in the range 3:1 - 20:1.

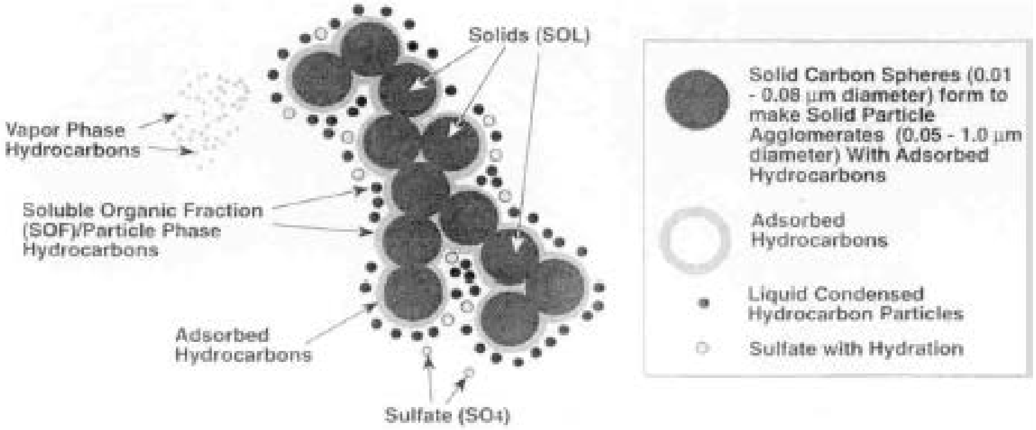

Figure 22 – Soot molecular "Millo's notes"

This leads to the agglomeration of solid carbon spheres (formed by nucleation of soot) and HC molecules, with heavy molecular weight coming from lubricant, fuel and sulphuric organic acids, can condensate and be adsorbed over the agglomerate itself.

2.2.1 PM Chemical composition

Diesel particulates are composed of elemental carbon particles which agglomerate and adsorb other species to form structures of complex physical and chemical properties.

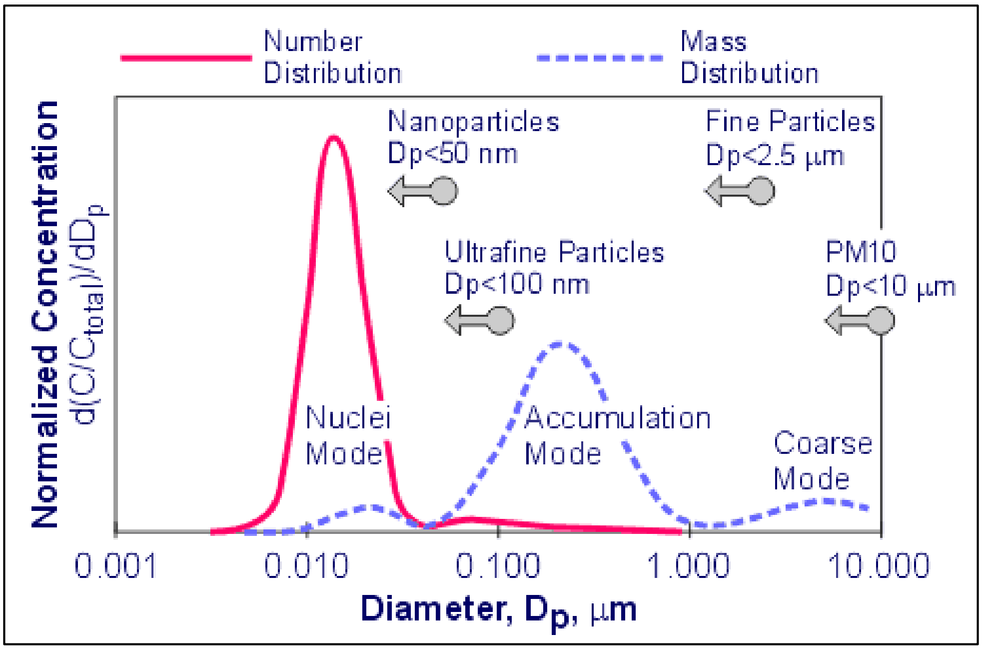

Figure 23 – Number and Mass distribution of PM "www.Dieselnet.com"

Diesel particulates have a bimodal size distribution. They are a mixture of nuclei mode and accumulation mode particles, schematically shown in Figure .

Nuclei mode particles are very small - their diameters are between approximately 0.007 and 0.04 µm. Nuclei mode particles are often referred to as nanoparticles, although these two terms are not the same. Nanoparticles are usually defined as particles below 50 nm in diameter (0.05 µm). Nuclei mode are primarily volatile and consist mainly of hydrocarbon and hydrated sulfuric acid condensates condensates which are formed from gaseous precursors as temperature decreases in the exhaust system and after mixing with cold air, be it in the laboratory dilution tunnel or in the ambient air. These volatile particles are very unstable; their concentrations strongly depend on dilution conditions such as dilution ratio and residence time. A small amount of nuclei mode particles may consist of solid material, such as carbon or metallic ash from lube oil additives.

Nuclei mode particles constitute the majority of particle number—on the order of 90%—but only a few percent of the PM mass

Accumulation mode particulates are formed by agglomeration of primary carbon particles and other solid materials, accompanied by adsorption of gases and condensation of vapors.
They are composed mainly of solid carbon mixed with condensed heavy hydrocarbons, but may also include sulfur compounds, metallic ash, cylinder wear metals, etc. Diameters of the accumulation mode particles are between approximately 0.04 and 1 µm with a maximum concentration between 0.1 and 0.2 µm.

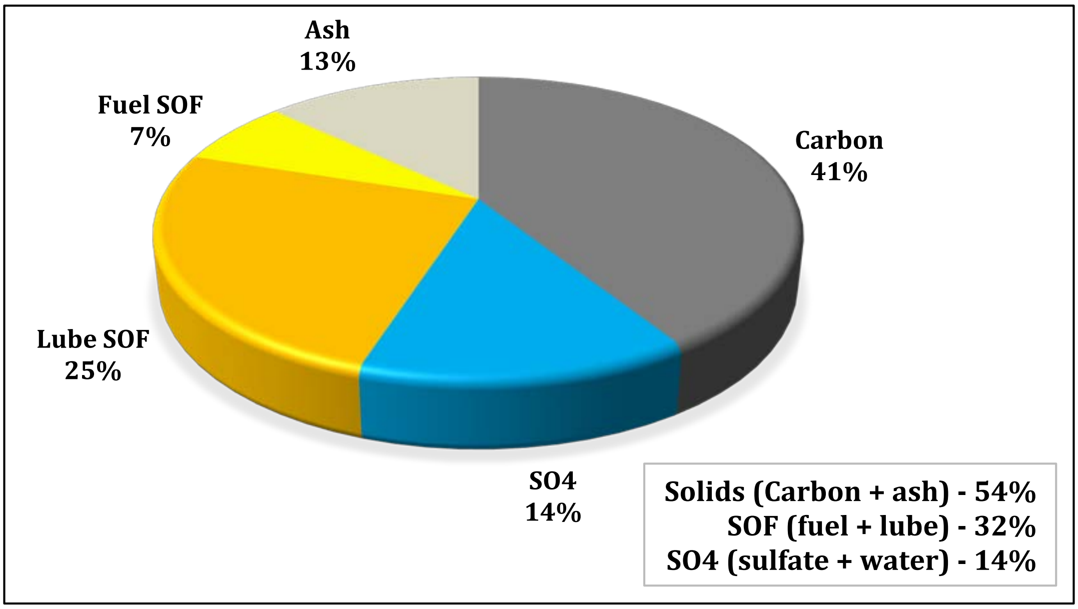

Chart 6 – PM compositions "Millo's notes"

PM is traditionally divided into three main fractions, as follows:

- **Solid fraction (SOL)**
    - Elemental carbon
    - Ash
- **Soluble organic fraction (SOF)**
    - Organic material derived from engine lubricating oil
    - Organic material derived from fuel
- **Sulfate particulates (SO4)**
    - Sulfuric acid
    - Water

In the followng Figure 24 is possible to see the different phases of PM structure.

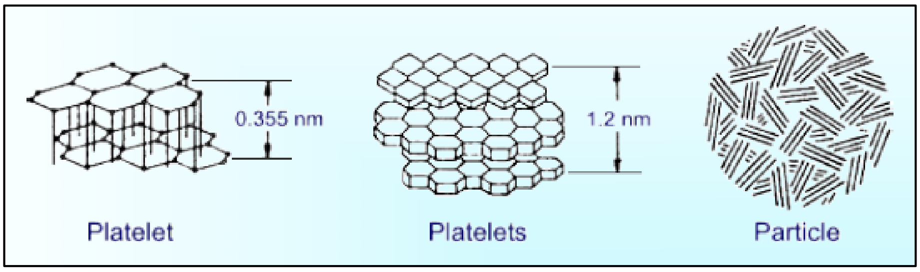

Figure 24 – PM structure "Millo's notes"

As it shows, the particles are made of agglomerates of smaller particles, which are coming from the agglomeration of platelets, that are originally formed from aromatic molecules (= hexagon rings that are the typical shape of aromatic hydrocarbons PAH such as benzene C6H6), so these are the building blocks of soot: a general rule is that the higher the aromatic fraction of hydrocarbons in diesel fuels, the higher the soot emissions, because when the burning occurs under severe lack of O2, this kind of hydrocarbons will lose the hydrogen atoms and the carbon rings will collapse together in this kind of well organized structure.

Some PAH are limited by legislation, because they are carcinogenic and could be dangerous for the people especially that work with fuels.

2.2.1.1 Ash

They are solid residuals of the combustion process, mainly coming from additives used to improve the performances of lubricant; these additives are used as:

- Detergents or dispersants to avoid that soot particles in the oil can create some large agglomerates of PM, thus worsening the characteristics of the oil;
- Acid neutralizers to avoid that sulphuric or nitric gases, that are formed by the condensation of oxides, can chemically attack the lubricant molecules;
- To improve the performances of lubricant under extreme pressure conditions, for example to avoid that the thin oil film is broken under very high pressures that we may have, for instance, between cam and tappet or also in other parts of the engine.

These additives contain oxides of calcium, zinc, magnesium and other kinds of metals, also sulphates and phosphates and all these substances significantly contribute to ash formation.

For eliminating the soot we can use the DPF[11] that traps the particles. The filter is periodically regenerated by burning soot accumulated on the filters, but even if we can regenerate the filter burning soot and burning the soluble organic fraction made of unburned HC, ashes can't be eliminated by burning, because ashes are solid residuals of combustion, meaning that ashes will remain trapped in the DPF and they will accumulate over and over.
For this reason, all the engines using a DPF require a specific lubricant defined "low Sulphur-low ashes" that has a specific formulation in which the additives have been replaced with different formulations of lubricant molecules, so that the amount of ashes is so small, that the DPF can be used for the entire life of the vehicle without requiring to be treated to remove these kinds of ashes.

2.2.1.2 Soluble Organic Fraction "SOF"

SOF is mainly composed by HC derived from lubricant oil rather than diesel fuels.

Its chemical composition is strongly affected by the operation condition of the engine, so also the effectiveness of the DPF conversion depends on this conditions.

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[11] DPF="Diesel Particulate Filter", it is a mechanical filter that it is mechanically trapping the particles tha are in the exhaust stream

The Figure 25 shows how much the SOF (in green) is and how much the solid fraction SOL + sulphate is (in white) in function of speed and load.

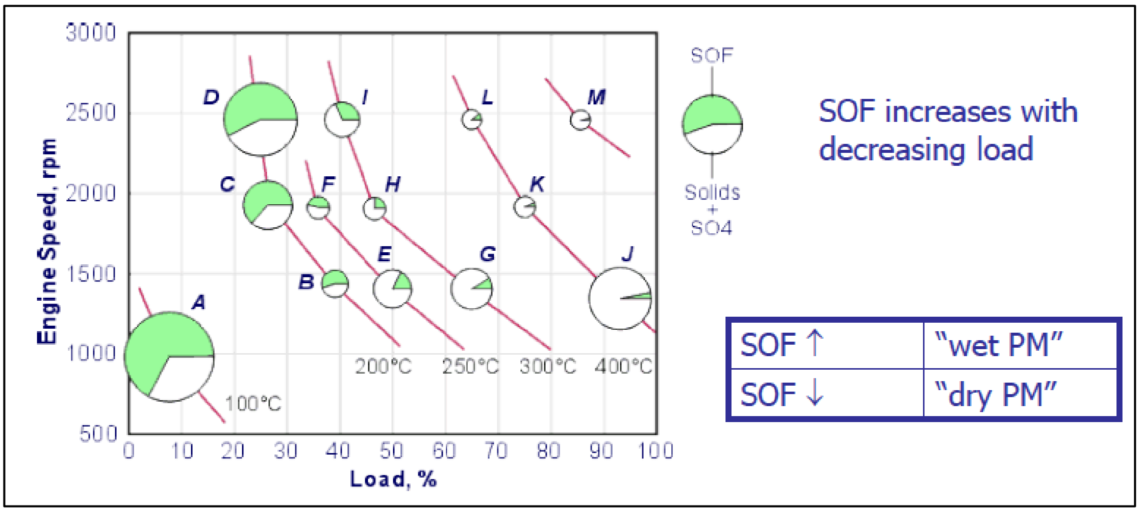


Figure 25 – Composition of soot as a function of load "www.DieselNet.com"

Dry PM: in the high speed-high load region: PM is mainly made of soot with no significant fraction of SOF, because in this operating condition, we usually operate close to stoichiometric A/F ratio and this operating condition makes more probable the soot formation. On the contrary, there is few formation of unburned HC.

Wet PM: in idle conditions and in low loads-low speeds: PM is mainly made of unburned HC (=SOF), which condensate and absorb on a relatively small soot fraction.

This is important in terms of aftertreatment system: diesel oxidation catalyst DOC can easily treat wet PM, because DOC can be quite effective oxidizing HC, before they can absorb and condensate.
So, composition of PM is important to understand the chances we may have to properly treat these pollutants in the aftertreatment devices.

2.2.1.3 Sulfate particulates "SO4"

These particulates come from the Sulphur present in the fuel. The most effectives way to reduce the SO4 particulates at the engine out is to limit the percentage of Sulphur in the diesel fuel.
The regulation changes based on the country in which you can sell the fuel.

2.2.2 PM formation process

The PM is going to be formed during the combustion process, which occurs in 2 stages:

- First stage: the fuel starts to burn in a very rich condition $\phi > 4$ ($\lambda < 0.25$, rich premixed reactions) at relatively low temperature (1600K) and fast formation process occurs.
  It involves aromatic molecules and leads to the formation of platelets, through the nucleation, and then of the initial soot;
- Second stage: during which the molecules, products of the rich premixed combustion, burn out (are further oxidized) through the near-stoichiometric diffusion flame at the jet periphery where the temperature is higher. Also aliphatic components can be transformed through pyrolysis and cracking to produce soot with an indirect and slower path.
  The soot-burnout is important too, because it determines how much soot we will have in the exhaust, because as the jet (made of fuel, air and combustion products) is entraining more and more air, so more and more O2 becoming available for the oxidation of the soot that has initially formed.

So it is also important to understand that the result in terms of soot emissions at the exhaust is the combination of two contrasting processes: the **soot formation** on one side and the **soot burnout** (or oxidation) on the other side.
So some techniques can be effective because they tend to suppress soot formation, some other techniques can be effective into enhancing the soot burnout.

2.2.3 Engine parameters that affect the PM formation mechanism

Also in this case, like the NOx formation, the main parameters that affect the PM formation mechanism are these following:

- A/F ratio;
- Injection timing;
- EGR rate.

So, now I’m going to discuss in more detailed way the 3 topics.

2.2.3.1 A/F ratio

If we plot soot or PM emission as function of the A/F ratio (Chart 7), considering for instance a NA engine, and we simply make a load shift from idle operating condition (no load), in which the A/F is high since the amount of injected fuel is small, to full load operating conditions in which, increasing the amount of injected fuel, we will operate the engine close to the stoichiometric value, we will see a trend like this:

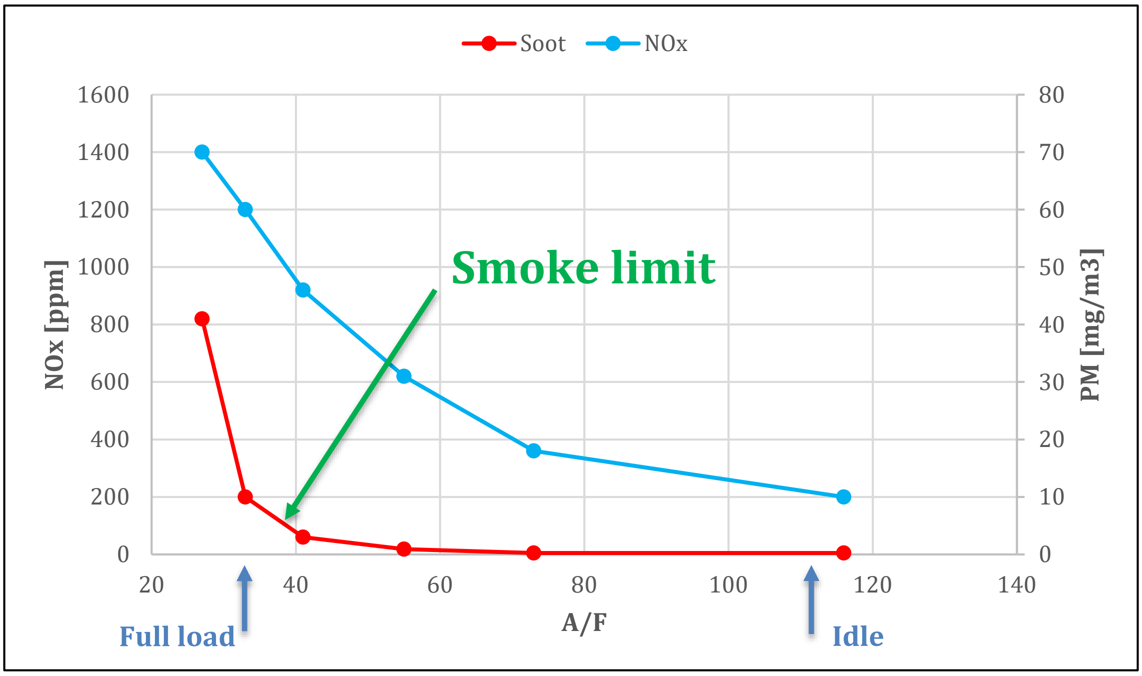


Chart 7 – NOx and PM Vs A/F

For very high fuel air ratio (at idle), we have a very small PM emission because we are injecting very small amounts of fuel, so we have few chances that this fuel is not able to burn.

For sure, we have some mount of soot formation but the high availability of oxygen can promote the soot burnout.

But then, approaching the stoichiometric value (A/F ratio decreasing), there is a dramatic change in the trend and PM emissions start to rise dramatically with a kind of asymptotic trend towards a vertical asymptote.
The point at which we have this abrupt change of the trend is called **smoke limit** (or A/F ratio limit or λ limit): it's the point, beyond which, if we further increase the amount of injected fuel, or further decrease the A/F ratio, we will have a huge increase in soot and PM.

The smoke limit in terms of A/F ratio changes depending on the engine revolution speed, because increasing the engine speed we have less time for the soot burnout but we have also more intense turbulent mixing between air and fuel.
Therefore, if we increase the speed, this limit moves to the left of the diagram, closer and closer to the stoichiometric value.

2.2.3.2 Injection timing

The PM emissions are extremely sensitive to injection timing, showing usually a significant increase with decreasing injection timing.
A NO_x – PM trade-off has therefore to be found.

The more the injection is delayed, the more difficult the soot burnout and this is the reason why, if we plot NOX and soot (PM) emission over a trade-off diagram for different injections timing values, the more delayed the injection is (so the smaller the injection advance is before the TDC), the higher will be the soot and particulate emissions.

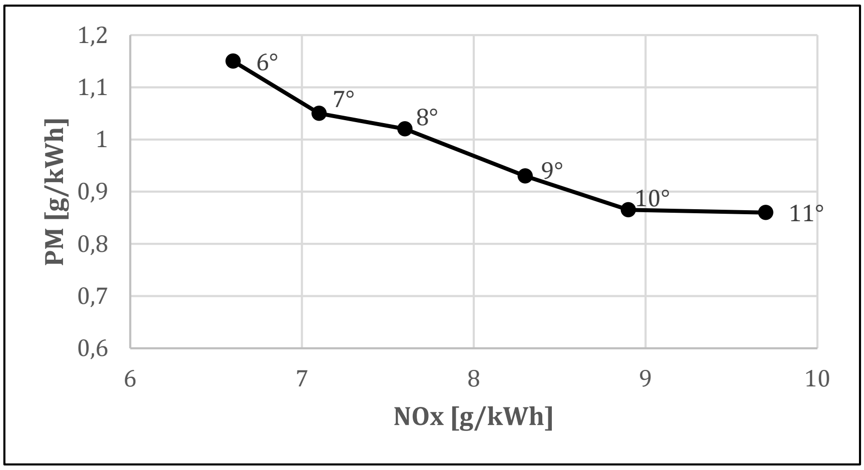


Chart 8 – PM Vs. NOx

If we delay too much the injection, soot burning that is still going on, when we already are opening the exhaust valve, so the combustion process will be too much delayed and soot burnout can't be completed before the exhaust phases starts.

2.2.3.3 EGR rate

The presence of dilutants in the combustion chamber can increase the soot emission for 2 reasons:

- lack of O2 during the combustion process due to this higher concentration of dilutants;
- soot burnout process is made more difficult because of the presence of dilutants.

We may start the exhaust phase, when we are still significantly in the soot burnout phase. We will prevent the complete oxidation of soot.

As general rule: what is good for NOX is not good for soot. There is the need a trade-off between action aimed to suppress NOX formation and, at the same time, suppress soot formation.

2.3 CO formation mechanism

Carbon monoxide typically does not present high relevance in conventional diesel engines, but it must be taken under control at EGR-operated part loads. Furthermore, for engines exploiting PCCI combustion, where high level of EGR rate is used, CO formation rate becomes relevant. CO forms from partial hydrocarbon oxidation, according to following flow chart:

Hydrocarbons → Radicals → Peroxides → Aldehydes → Ketones → CO

Finally, CO is oxidized according to the following reaction

$$CO + OH \leftrightarrow CO_2 + H$$

oxidation process of CO in CO_2 is slower than the one that forms CO. Moreover, at the temperature reached by the combustion process (≈ 2800 K), the reaction rate is high enough to achieve the chemical equilibrium.
A considerable amount of CO is therefore present in exhaust, despite the expected quantity calculated considering the chemical equilibrium.
This states that the phenomenon is controlled by the reaction chemical kinetics: a "freezing" effect prevents the further CO oxidation leading dissociation products concentration unaltered, as far as the temperature decreases.

2.4 HC formation mechanism

The HC pollutant emission in the CI engines is more complex than SI due to the different fuel that has a long carbon chain with different organic compounds.

In addition, during the combustion, the fuel is subjected by important pyrolysis reaction that gives solid particulates.

In the Figure 26 you can see different ways to HC formation.

So in the following chapter I will discuss in deep way the different mechanism for the HC formation that are:

- Overmixing/overleaning;
- Undermixing/overrich;
- Bulk quenching;
- Misfire;
- Spray impingement.

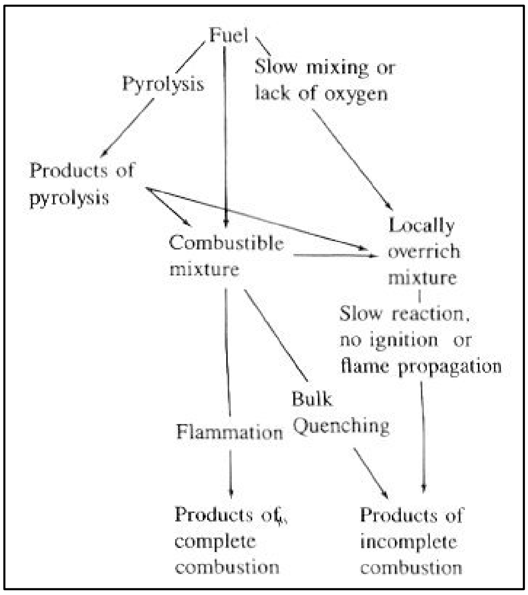


Figure 26 – Flow chart of HC formation mechanisms “Spessa’s notes”

The composition of the HC varies with the engine load, at low load the overmixing produces unburned molecules instead, at higher load, the undermixing creates a partial oxidation and pyrolysis compound.

2.4.1 Different HC formation mechanism

2.4.1.1 Overmixing

In this situation, we have an excessive local mixing with the air so the mixture fuel-air became to lean and so we can’t have the auto ignition of the fuel or the sustain of the flame propagation inside the chamber.

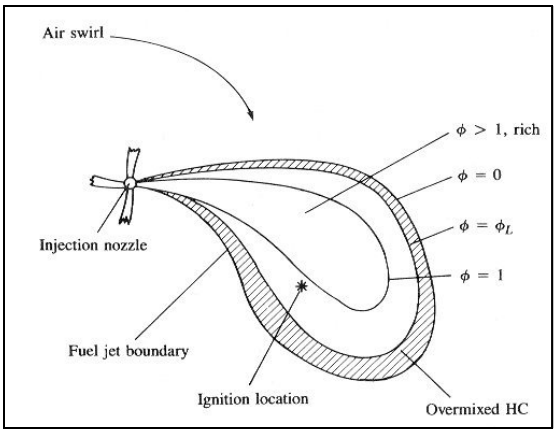


Figure 27 – Plum with description of equivalence ratio "Spessa's notes"

How you can see on the Figure 27, the leaner zone is in the boundary layer, with a equivalence ratio $\Phi_L \approx 0.3$ that is too low to auto ignite.
However we can have a oxidation thermic reactions, that happen in a slow and incomplete way.

The entity of the fuel in the overmixing zone, that it is exhausted in the environment without participating at the combustion process, is directly affected by the mixing phenomena with the air during the ignition delay and also by the ignitability limits.
These parameters depending on the injection law, on the fuel property, on the air motion inside the cylinder and also on the working condition of the engine.

2.4.1.2 Undermixing

This happens when there is a not sufficient local mixing between air and fuel that creates a rich pocket (overrich mixture) that can't burn or sustain the flame propagation inside the chamber, due to lack of oxygen.

This phenomenon has 2 main causes:

- Excessive fuel quantity injected during the combustion phase (Over fueling during the acceleration phases);
- Drip of the fuel from the injector sac.

The direct injection diesel engine has a limitation of the equivalent ration at 0.7 due to the high production of soot at the exhaust.

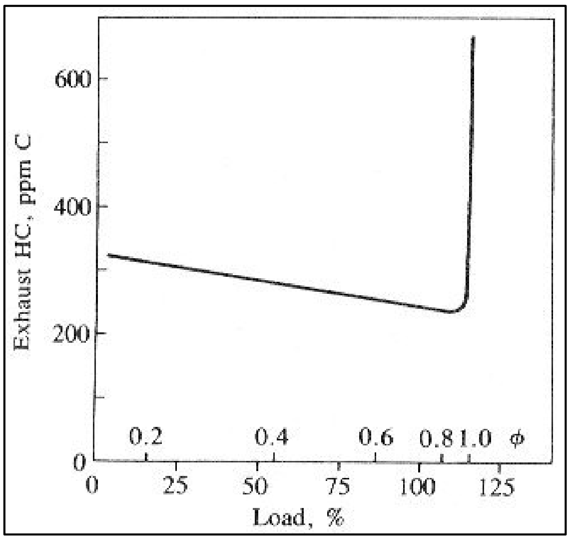


Chart 9 – Exhaust HC Vs. Load "Spessa's notes"

During the acceleration phase can happen the overfluing, the mixture is overall lean but the flame propagation can reach zones with rich task of fuel that produce partial oxidation products and unburned fuel.

Generally, the undermixing produces less HC emission than overmixing that happens at low loads and idle.

Talking about the HC emission due to the drip of fuel from injector sac, we can see, from the following graph, that also without a fuel inside the sac we haven't an HC emission level equal to zero. This means that also the fuel present in the nozzle holes contributes to the HC emissions.

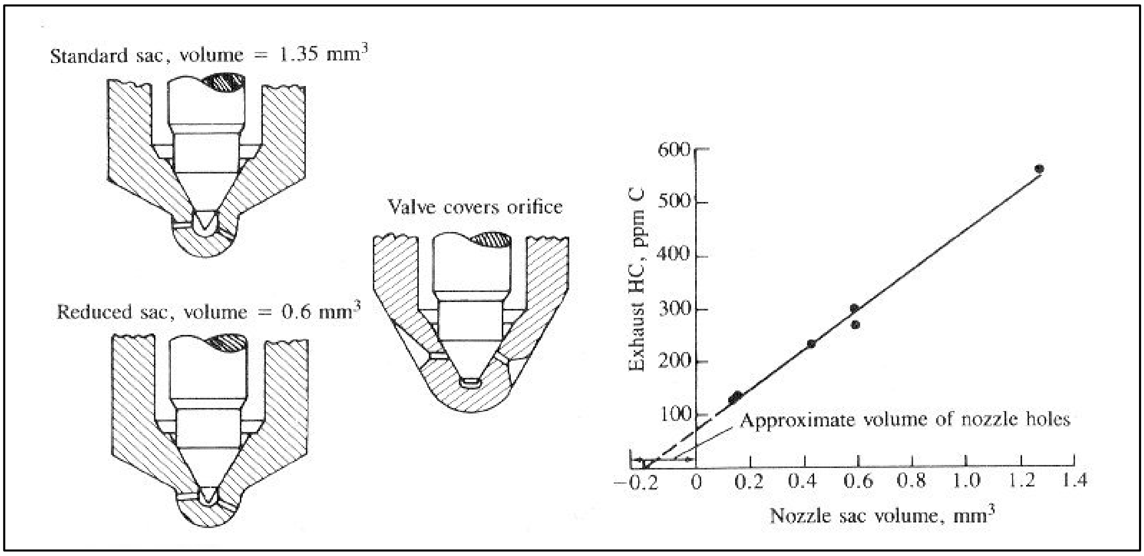


Figure 28 – Volume of nozzle holes "Spessa's notes"

In addition, we can observe that 1 mm^3 of sac volume contributes at the HC emission with 350 ppm C$_1$, instead 1 mm^3 of fuel could produce a 1660 ppm C$_1$, so this means that in the sac isn't present only the liquid fuel due to the cavitation that happens with a low injection pressure value when the injection is finished.

2.4.1.3 Bulk quenching

The bulk quenching is a physic process that determines incomplete oxidation of the mixture causes by the expansion phase that happens in the cylinder. This flame extinguishing is determine by the higher diffusive process velocity than the chemistry one.

In particular, this phenomenon can be enhanced by the rapid mixing of the mixture and also by the presence of thermal boundary layer.

From the experimental tests we can see an increase of HC emission about 30% if we change the oil and coolant temperature, from 90°C to 40°C, without changing the ignition delay.

The parameters affecting the bulk quenching are:

- High expansion velocity;
- Low equivalence ratio;
- Retarded combustion;
- High percentage of fuel during the injection, that impacts the walls (spray impingement)

2.4.1.4 Misfire

The misfire happens less frequently in the CI engine respect to the SI because the cycle to cycle variations of the combustion process are less pronounced. However, it can happens in a very unfavorable conditions as the low temperature and pressure in the combustion chamber during the compression or with a SOI retarded.

2.4.1.5 Spray impingement

This is the principal cause of HC emissions in the diesel engine.

This happen when we use injectors with high nozzle dimensions and when the distance between the injector nozzle and the walls is below certain value. In this condition, the evaporated fuel impact the cold walls and become liquid, in addition it also can be diluted into the lubricant so it doesn’t participate at the combustion process.

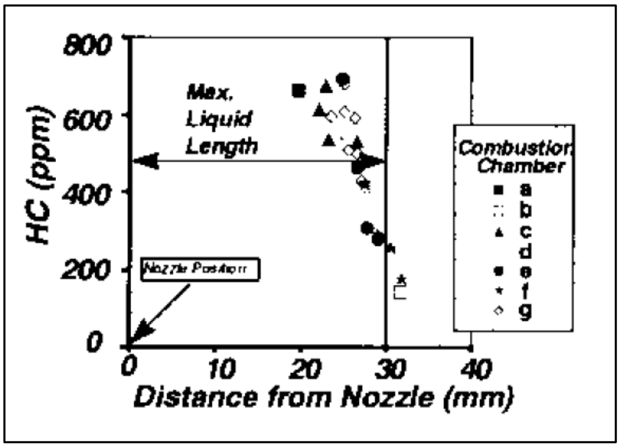


Chart 10 – HC Vs. Distance from Nozzle "Spessa's notes"

With the higher nozzle diameter, the liquid length is increased, so enhance the spray impingement.

How we can see in the Chart 10, the distance from the nozzle has to be higher than the liquid length penetration, in order to avoid the spray impingement.

2.4.2 Engine parameters that affect the HC formation mechanism

Now I'm discussing more in details the engine parameters affecting the HC formation mechanism that are:

- Ignition delay;
- Intake air temperature;
- Combustion chamber shape.

2.4.2.1 Ignition delay

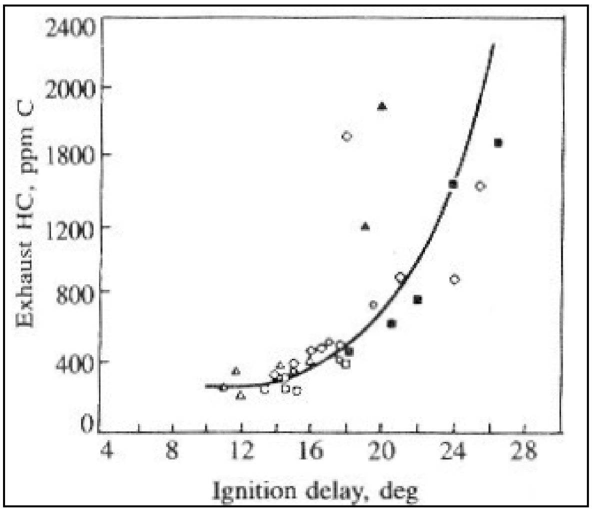


Chart 11 – HC Vs. Ignition delay "Spessa's notes"

This parameter is fundamental because affects directly the mixing between air and fuel.

Increase the HC emission due to the overmixing and bulk quenching, this can be visible in the figure by the increase of the curve slope at the increasing of ignition delay.

The HC emission with the ignition delay lower than the critic value depending mainly by the fuel come from the injector sac.

2.4.2.2 Intake air temperature

This parameter influences the overleaning and bulk quenching phenomena.

In fact, the ignition limit equivalence ration Φ_L increases with the decrease of the air temperature inside the cylinder in the moment of auto ignition, in turn decreases at the decreasing of the intake air.

This means at higher quantity of fuel in the jet area characterized by $\phi < \phi_L$ and so higher quantity of HC emission for overleaning.

In addition, a lower intake air temperature determines an increase of ignition delay, that I mentioned before, increases the HC emission by overleaning and bulk quenching.

At the end, a lower intake air temperature enhances the bulk quenching also in the anticipated phases of the cycle.

2.4.2.3 Combustion chamber shape

The adoption of bowl in the piston can improve the swirl motion at the end of the compression phase and also to generate a squish motion.

The bowl and the injector position determine the proportion of fuel that is divided between the bowl and the squish area. The problem consists into optimization of the injection parameters, the injector position and the chamber shape in order to obtain a fuel uniform distribution.

The mixture presents in the squish region is at lower temperature and so can determine an increasing of HC emission due to the quenching. The increasing is influenced by the impact point between the liquid fuel and the walls, by the injection and by the combustion chamber shape.

The diameter of the bowl, with the same rc, increases the swirl level at TDC. However, we have to take it under control because an excessive mixing can create HC emission by overmixing. We can also have the same effect with the same bowl diameter but increasing the deep of the bowl.

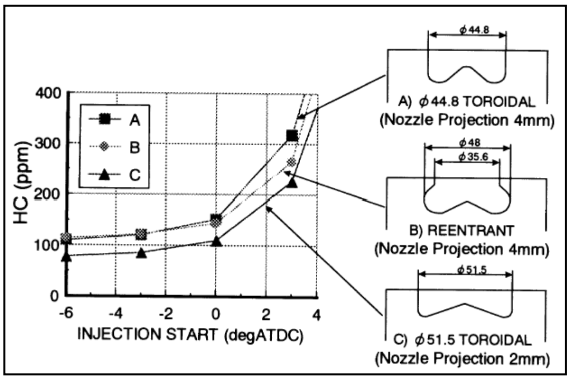


Figure 29 – HC influenced by different piston bowl shape "Spessa's notes"

In the case with injection before the TDC, the swirl prevents the dispersion of the mixture in the squish region. So the HC emissions are influenced mainly by the distance between the injector holes and the walls. The bowls "A" and "B" give higher HC emissions.

In the case with the injection after TDC, the swirl can create a revers squish motion that pull the air inside the bowl that can leads an overleaning or bulk quenching.

So, the re-entrant combustion chamber “B” allows to counteract the exit of the charge from the bowl, reducing the HC emission respect to the toroidal chamber “A”.

2.5 Diesel combustion noise

Noise in engines can be ascribed to different categories:

- Mechanical noise: that comes from impacts between different engine parts during normal operating conditions: impact of the valve against seat, or the impact between the needle of the injector and the needle seat, or also the piston slap when the piston impacts the liner close to the TDC;
- Gas dynamic noise: the engine is a reciprocating machine which is operating in unsteady conditions, so there are pressure waves both at the engine intake and exhaust. The result is the intake/exhaust noise: the exhaust noise is suppressed with the silencers.
  As general rule, volume kills the gas dynamic noise: if the muffler in the exhaust pipe has a big volume, the expansion inside the big volume will kill the noise, because it behaves as a constant pressure volume, killing the pressure waves travelling across the exhaust pipe.
  The same can be said of the intake: one important function of the intake air filter, besides the filtering and the cleaning of the intake air, is also to provide a big volume to kill the engine intake noise.
- Combustion noise: noise produced by the combustion process mainly in in diesel engines. In diesel engine we have a rapid pressure rise in the combustion process, with very high-pressure first derivative during the premixed burning phase. There is a sort of step variation of the mechanical load on all the engine components such as the piston or the crank, that are elastic components: a sudden change in load on elastic structures gives raise to vibrations in the structure itself, that will spread in the in the surrounding environment and will perceived as noise.

2.5.1 Input spectrum

In order to understand the origin of this combustion noise and how we can try to suppress or mitigate it, we have to analyze what is the input spectrum in our system.

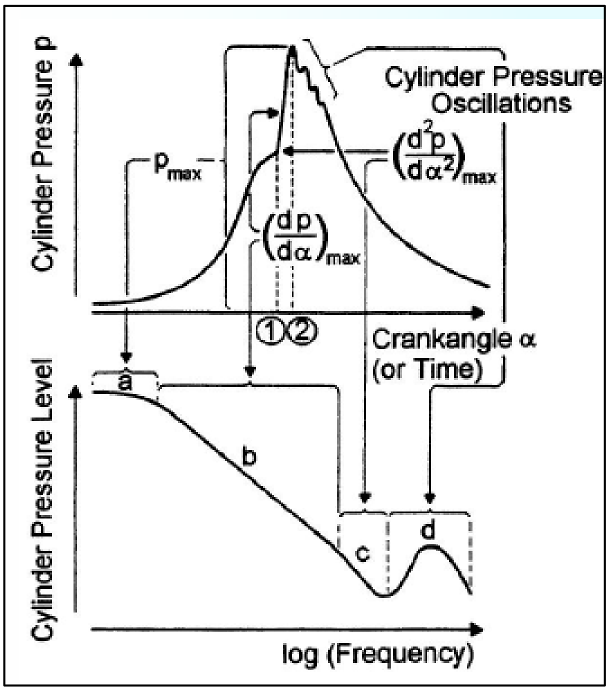


Figure 30 – Pressure behavior "Millo's notes"

The system is the engine structure, which is excited by an input force coming from the combustion process (the pressure in the combustion chamber).
The typical diesel combustion pressure vs. time or crank angle trend has a periodic trend, with period equal to the one of the operating cycle (2 revolutions in 4 strokes engines), the pressure function can be decomposed into harmonics by means of the Fourier transform and the amplitude of each harmonic can be plotted in function of the frequency of the harmonic.

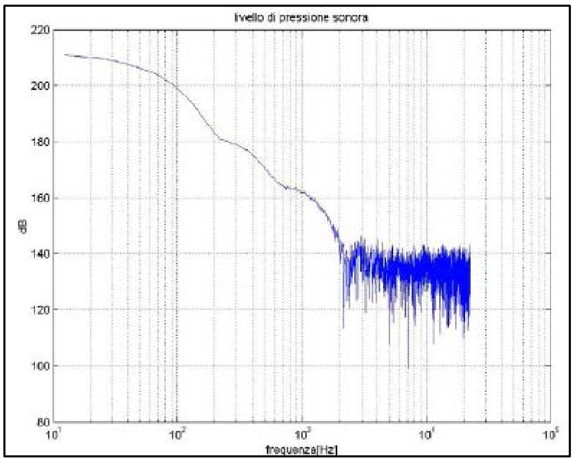


Chart 12 – Sound pressure level

The Chart 12 it's a experimentally diagram that shows frequency spectrum of a diesel combustion pressure at defined operating conditions in terms of speed and load (5bar BMEP – 1500rpm).

The pressure step, highlighted as "b" in the figure n, produces a very wide band of frequency "b" (in figure n2), because a sudden variation corresponds to extremely high frequencies, while the initial and the ending levels (a) correspond with very low frequency.

Then we may have some additional frequency content due to some pressure oscillations

(d) inside the combustion chamber, similarly to what happens in a knocking gasoline engine.

In terms of perceived noise, this region (b) is very important

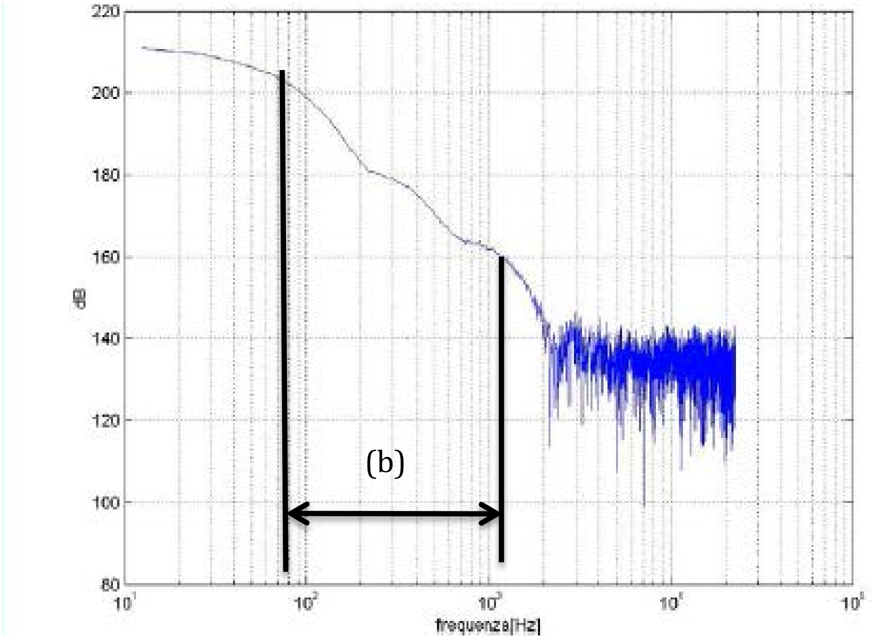


Chart 13 – Pressure Vs Frequency

2.5.2 *Transfer function*

Not only the input spectrum is important, but also how the system reacts to the input. In order to understand it, we should determine the transfer function of the system (engine block).
This is very difficult to analyze the engine block in the design phase because it isn't ready to be build.
For this reason we can use a standard transfer function for all the engines.

Here below we see a filtering transfer function, it represents how the amplitude of the input is going to be transformed depending on the frequency band of the input wave itself. The higher the value, the lower the attenuation, this means:

- In the low frequency range, the high value of attenuation means that the system is able to absorb the input without producing any significant vibration;

- In the central range of frequencies, the system is extremely sensitive and is not able to absorb these kinds of input, that are transmitted to the engine structure without any significant abatement in the high frequencies, there is again a significant abatement of the input.

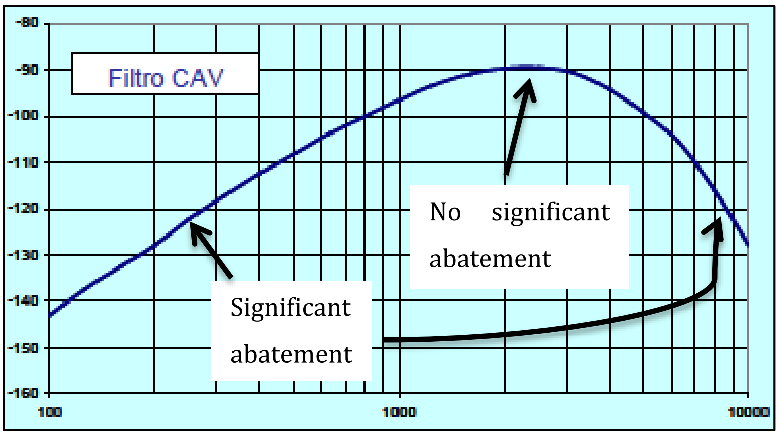


Chart 14 – CAV filter "Millo's notes"

This is a kind of standardized engine transfer function, which is going to be used for every kind of engine. This was set up by AVL in the 70's for diesel tractor engines and it has become a standard in automotive industry, in fact we speak of AVL combustion noise when we use this kind of transfer function: the results we are going to obtain in terms of combustion noise evaluated with this transfer function will not be well correlated with a real combustion noise.

In order to improve the correlation with the actual noise that we are going to perceive, the input function is not only filtered by means of this engine transfer function, but it's also filtered by means of the **A filter**, that corresponds with the human ear filtering; the human ear is sensitive to frequencies between 1 kHz and some kHz and less sensitive to lower frequencies and higher frequencies.

So we will filter the input function through a filter representing the engine, through a second filter representing human ears, and through an additional filter, the low pass filter, used to remove the high frequencies noise mainly due to pressure oscillations.

So after, every OEM has internal procedure to find a better correlation between the standard combustion noise and real combustion noise of won engine.

2.5.3 Parameters that influences the combustion noise

Combustion noise is determined by pressure first derivative. This means that if we are able to measure the pressure in the cylinder, it is sufficient to compute its first derivative to have a first indication of the engine noise, avoiding buy expensive noise meters.

The higher the pressure derivative, the higher will be the combustion noise.

So, if we want to reduce the combustion noise we have to reduce the first premixed combustion phase, in order to reduce the pressure first derivative.

2.5.3.1 EGR rate

The higher the EGR rate, the less reactive the mixture will be, the higher will be the peak of the premixed burn, so the higher the pressure first derivative and the higher the combustion noise.

2.5.3.2 Injection pressure

Increasing the injection pressure, the same amount of fuel can be injected in a shorter period of time, the ignition delay τ more or less will remain the same. For the lower injection pressure, a certain amount of fuel is accumulated in the combustion chamber and burn all together; with higher injection pressures, we will have more fuel accumulated inside the combustion chamber and burning all together, so we have a huge peak of premixed burn due to an intense HRR during the premixed phase, leading to higher pressure first derivative with consequent increase of the combustion noise.

2.5.3.3 Pilot injections

In order to avoid the peak of the premixed phase, we can use the pilot injection. We inject a small amount of fuel in advance before the main injection. This amount of fuel is a few percent (less than 10% usually 5%) of the total amount of fuel and the pilot fuel is going to burn all together producing an increase of pressure first derivative, thus paving the way for the combustion of the main injection. The pilot injection significantly reduces the ignition delay of the main injection, thus suppressing the main peak of the heat release rate HRR; therefore the pilot injection is one of the most efficient ways to suppress the combustion noise.

In the Chart 15 below, for the same engine operating condition, we may have significantly different combustion noise levels depending on the quantity of fuel injected during the pilot injection.

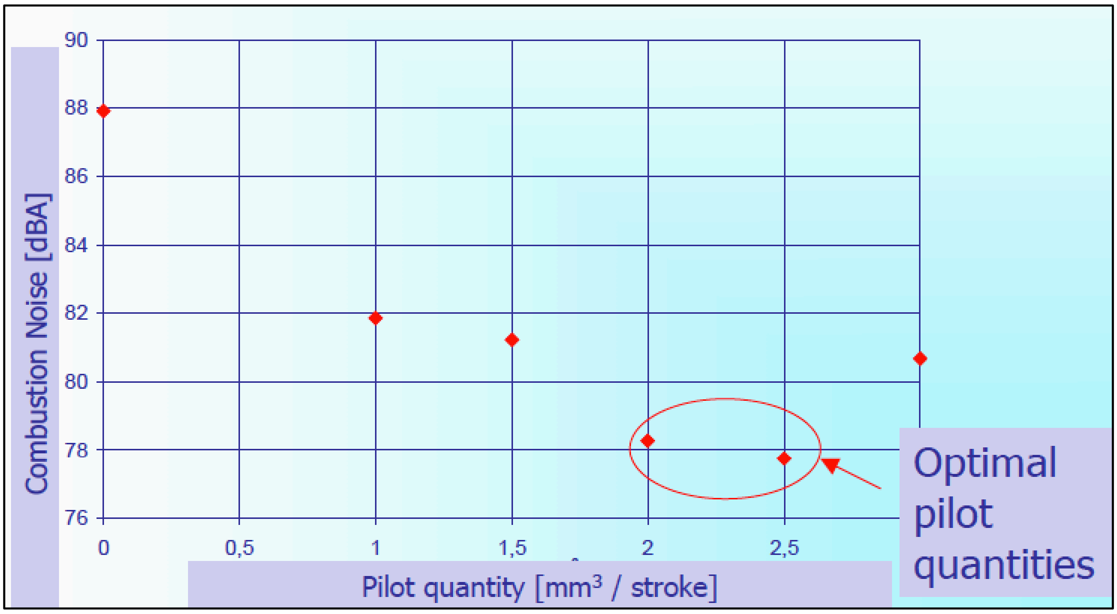


Chart 15 – Combustion noise Vs. Pilot quantity

Adding a pilot injection of 1mm3 we drop the combustion noise from 88 to 82 with a drop of 6 dBA, that is a huge difference in terms of noise reduction. With 2mm3 of pilot fuel quantity, we can drop down of a further 6 dBA, reaching more or less 78 dBA.
A further increase of the pilot quantity leads to a minimum in the noise, but increasing the pilot further, the noise starts to rise again.

We can understand this trend by looking at the pressure trend and at the heat release rate HRR of different combustion processes in the Chart 16 on the following page. Looking at the previous diagram (Chart 15), we notice that the point corresponding to 1,5 mm^3 (blue plot) gives more or less the same noise given by the point in which we have a pilot injection volume of 3 mm^3 (red plot), but the combustion processes occurring in the 2 above-mentioned points are different, despite the same noise level.

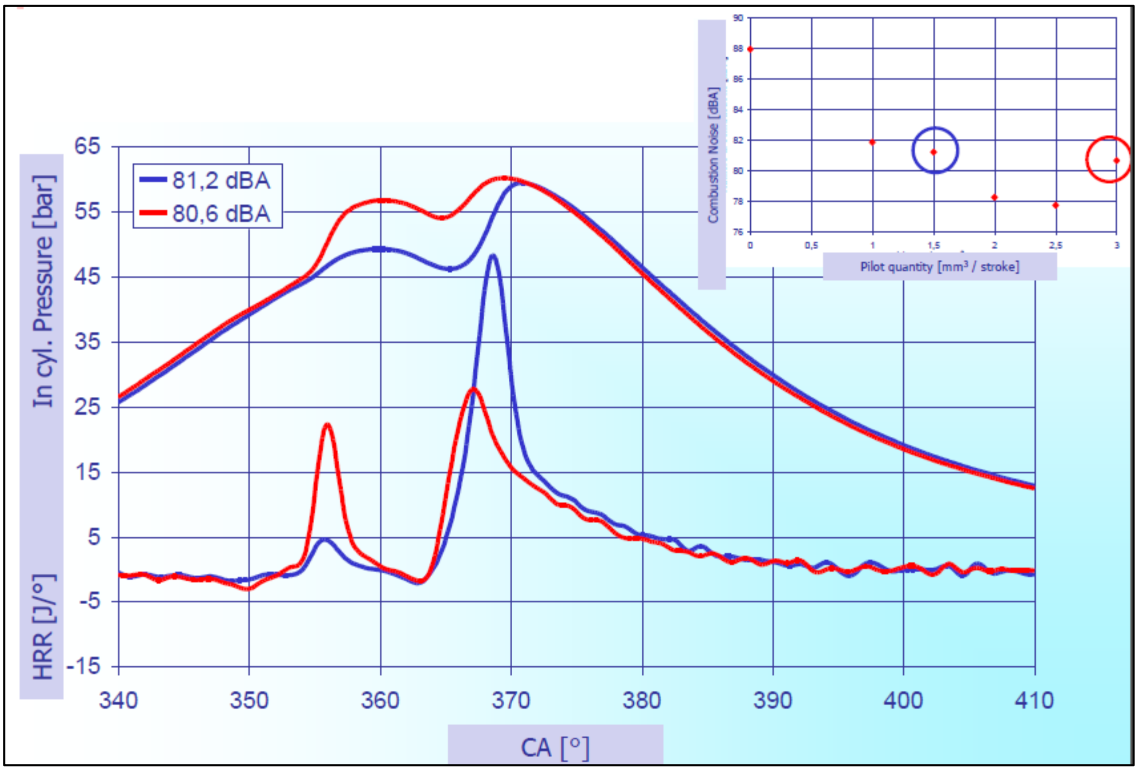


Chart 16 – HRR and In cyl. Pressure Vs. CA

The blue curve shows a moderate effect of the pilot since the HRR of pilot injection has a small peak (see HRR plot between 350°-360°CA).

We can see also the pressure trend: it's clear that the blue curve will give higher combustion noise, because its pressure trend is steeper than the red curve's one, thus the pressure first derivative will be higher and, as a consequence, the noise will be higher too. In the red curve, the pressure trend can be smoothed by increasing the HRR contribution of the pilot injection, thus decreasing the ignition delay and decreasing the intensity of the main combustion process that will show a smaller HRR peak. It is clear that increasing the importance of the pilot injection, as it is the case of the red plot (3 mm^3), also its contribution on noise becomes more important. In the case of the red curve, we observe that the pilot gives a HRR comparable with the HRR of the main injection and the pressure increase may become higher in the pilot than in the main injection.

Therefore, beyond a certain level, an increase of the pilot quantity is not more beneficial for combustion noise reduction, because the pilot injection will start to become noisier than the main injection. There is then an optimal value of the pilot injection, beyond which it is not more convenient to further increase the volume of fuel injected in the pilot. We have to control the pressure derivative or the HRR.

[1] [2]

3 Diesel exhaust after treatment systems

The most critical issues for after treatment systems for diesel engine concern:

- NO_x reduction (engine exhaust environment with high oxygen concentration);
- PM oxidation (high temperature > 600°C are needed for soot oxidation, but this temperature is much higher than the typical exhaust gas temperature level in diesel engine.

The main catalyst technologies that I will discuss for gas pollutant are the following, present in the Table 3:

Catalyst Technology	*Reaction type*	*Reduced Emissions*
Diesel oxidation catalyst "DOC"	*Oxidation*	*CO, HC (incl. PAH), PM (SOF), odor*
SCR	*Selective catalytic reduction by ammonia/urea*	*NOx*
DeNOx (lean NOx) catalyst	*Selective catalytic reduction by hydrocarbons*	*NOx, CO, HC, PM (SOF)*
NOx adsorbers	*Adsorption (trapping) of NOx from lean exhaust, followed by release and catalytic reduction under rich conditions*	*NOx, CO, HC*

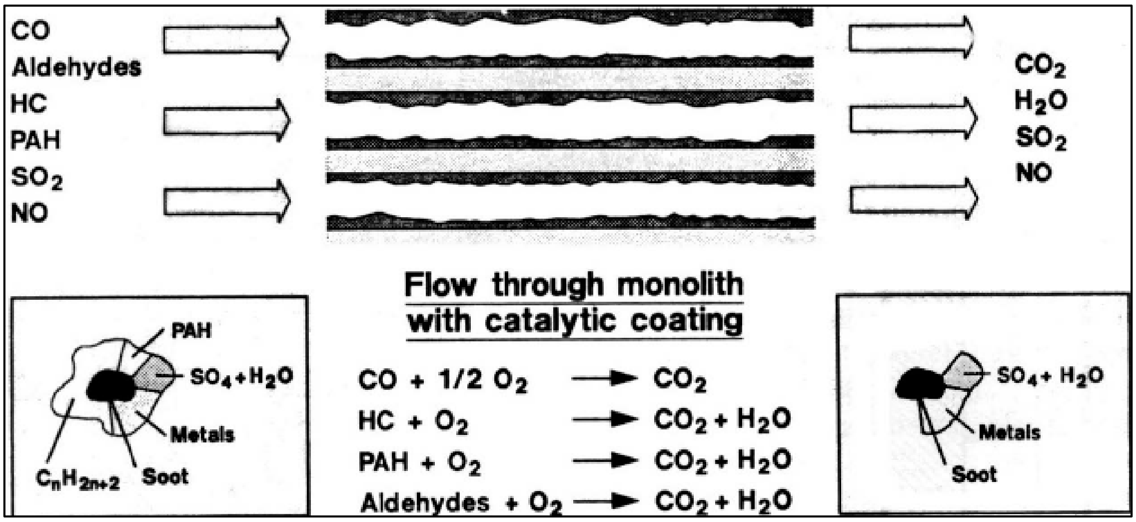
Table 3 – Catalyst technology

I also discuss the DPF, it isn't in the list because it acts as mechanical filter, blocking solid particles in the exhaust.

3.1 Diesel oxidation catalyst "DOC"

3.1.1 Catalyst features

It is pretty similar to TWC[12], it's impossible understand just looking the monolith if it is a DOC or TWC, because there is no difference in structure. It is made with a cordierite monolith, coated by a wash coat where precious metal (typically Pt, Pd) are laid on. The monolith is flow-through type, it means that the exhaust gas is flowing through the monolith channels.



$CO + 1/2\ O_2 \rightarrow CO_2$
$HC + O_2 \rightarrow CO_2 + H_2O$
$PAH + O_2 \rightarrow CO_2 + H_2O$
$Aldehydes + O_2 \rightarrow CO_2 + H_2O$

Figure 31 – DOC catalyst reactions

Ceramic monolith substrates (cordierite) are generally used, the metallic substrates are not common in this case because of the lower temperature of exhaust gas in diesel and in fact they are used only in high performance SI engine with high temperature. The cell density is about 300-500 cpsi.

At sufficient high exhaust temperature, DOC is typically quite effective in oxidation of CO and unburned HC, with efficiency that can be higher than 90%. DOC activity can be extended to PM (reducing the SOF) and on PAH as we can see on the picture above, where there is a PM before and after the catalytic action. Catalyst can act in advance by oxidizing unburned HC when they are in gaseous phase, before they are condensed and absorbed over the soot to form PM.

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[12] TWC=Three Way Catalyst that is a catalyst used in the gasoline engine.

In terms of reduction of PM emitted by the engine, can be quite different depending on the composition of the PM (in particular on the SOF content). We can have a wet or a dry PM, depending on the engine operating condition:

- at high load, we typically have the dry PM, which is mainly made of soot. In this case DOC is almost ineffective;
- at low load, we have a wet PM with an high SOF, PM can be effectively removed from the exhaust.

In terms of reduction of main species that are CO and HC, the catalysts have the following efficiency in function of the exhaust gas temperature:

The conversion efficiency is higher for CO than HC for a given temperature.

The light-off temperature in this case will be 200°C for CO, but it will be more than 250°C for HC (light off temperature = temperature with conversion efficiency equal to 50%).

The high-temperature conversion efficiency for CO or HC or the height of the final plateau depends on the mass transfer, so can be improved by means of higher cell density or reducing the space velocity (using a larger catalyst).
On the contrary the low-temperature conversion efficiency for CO or HC depends on chemical kinetic: it's depending on the catalyst (such as noble metals Pt, Pd) and on their concentrations.

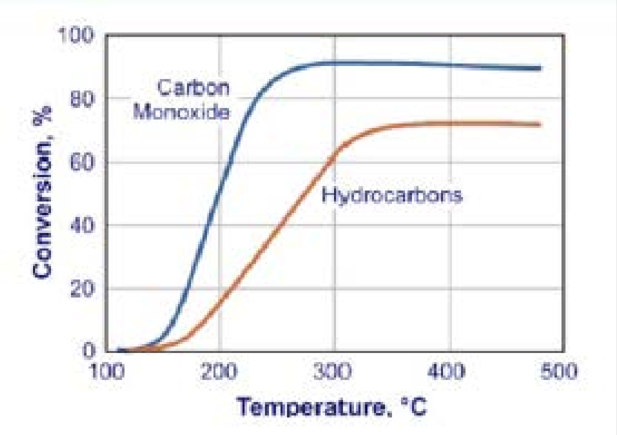

Chart 17 – DOC convertion efficiency
"www.DieselNet.com"

The Chart 17 show the conversion efficiency as a function of inlet gas temperature and gas space velocity. The lower is the space velocity and the higher will be the temperature, the highest will be the efficiency.

However, a DOC also promotes oxidation of other compounds than CO and HC, and may lead to the production of underirable prodcuts, such as:

- PM emissions are increased, because sulfate particulate increases the total PM mass.
  In the past, this problem was typically solved by means of the addition of sulfur oxidation inhibitors in the catalyst formulation (vanadium oxide). Nowadays there are no big issues, because at least in EU and Japan ultralow-sulfur fuels are used, with less than 10ppm of sulfur in the fuel. The formation of sulfur oxide is controlled from the beginning.
- Oxidation of NO to NO_2: NO_2 is extremely toxic and more dangerous than NO, but luckily in the environment, the thermodynamic equilibrium that is reached, is normally limiting the concentration of NO_2 So it's not a big issue. In addition, the higher amount of NO_2 is used for better operation of the SCR and also can promote the oxidation of soot at lower temperature.

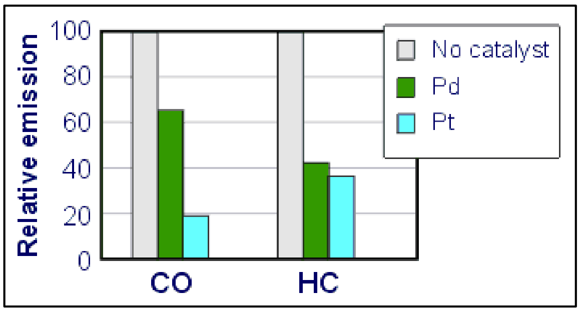

Chart 18 – PGM activity "www.DieselNet.com"

For what concerne the composition of the catalyst, we have to choose in good way the percentage of PGM[13] inside the coat. Pt is most active for the oxidation of CO and HC in diesel exhaust. The lower activi of Pd in diesel application is quite opposite to its activity in gasoline exgaust. Pd is used to improve the light-off performace in gasoline engine. The difference in activity is explained by the different chemical compostion of gasoline and diesel exhaust hydrocarbons because the gasoline on has short carbon chians and contain many unsaturated componunds, hile diesel hydrocarbons are characterized by long carbon cahins and mostly saturated bonds.

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[13] PGM= Platinum Group Metals

3.1.2 Catalyst layout

Until EU4, DOC was typically placed in the engine compartment, close to the engine, in CCC[14] position, and we typically add a second DOC before a DPF in UFP[15] because DOC has not only the extremely important function of oxidize CO and HC during the normal operation of the engine, but also to oxidize HC during DPF regeneration event.

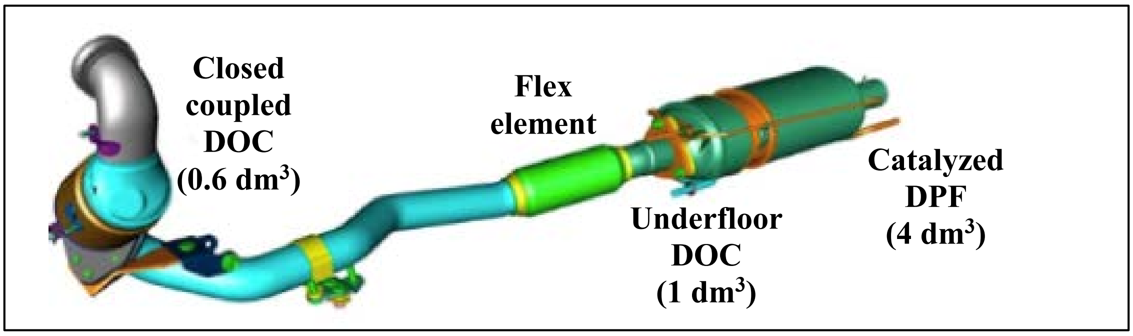

Figure 32 – Catalytic converter layout EUIV

In EU5 and EU6, as we have mentioned speaking about legislation, the more demanding legislation limit concerning PM emission has practically forced all OEM to adopt DPF. DPF is now a must from EU5 and the layout of the exhaust has significantly change from EU4 to EU5: in EU4, DPF was mounted in UFP, starting from EU5 it has been moved in CCC and typically DOC and DPF are integrated in a single canning: DOC before DPF.

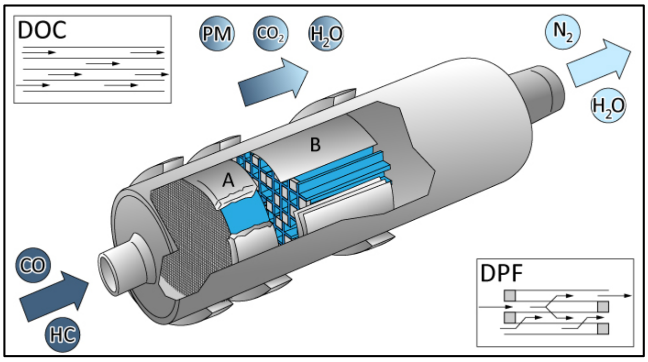

Figure 33 – DOC + DPF

Figure 34 – DOC + DPF mounted on engine

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[14] CCC= Close-Coupled Catalyst'

[15] UFP= UnderFloor Position

The DOC can fulfill the two main requirements:

- Oxidize CO and HC during normal operations;
- Operate as a catalytic burner during DPF regeneration.

Especially for the DPF regeneration, the closer the DPF to the engine, the lower will be the amount of fuel that we need to waste in order to increase the temperature of the exhaust gases (1m of pipe = 100°C temperature drop). In the Figure 33 and Figure 34 is shown the DOC+DPF in a single canning and its layout.

3.2 Selective catalytic reduction with ammonia “SCR”

3.2.1 Catalyst features

It’s a well establish and well known technology coming from large stationary diesel engines and power plants. Then, it has been successfully adapted to control NO_x in mobile applications such as marine engines, trucks and diesel cars.

The environment (exhaust gas) is always rich of oxygen, so we need of using a specific reductant for NO_x: *ammonia* NH_3, which is injected into the exhaust gas upstream the SCR catalyst.

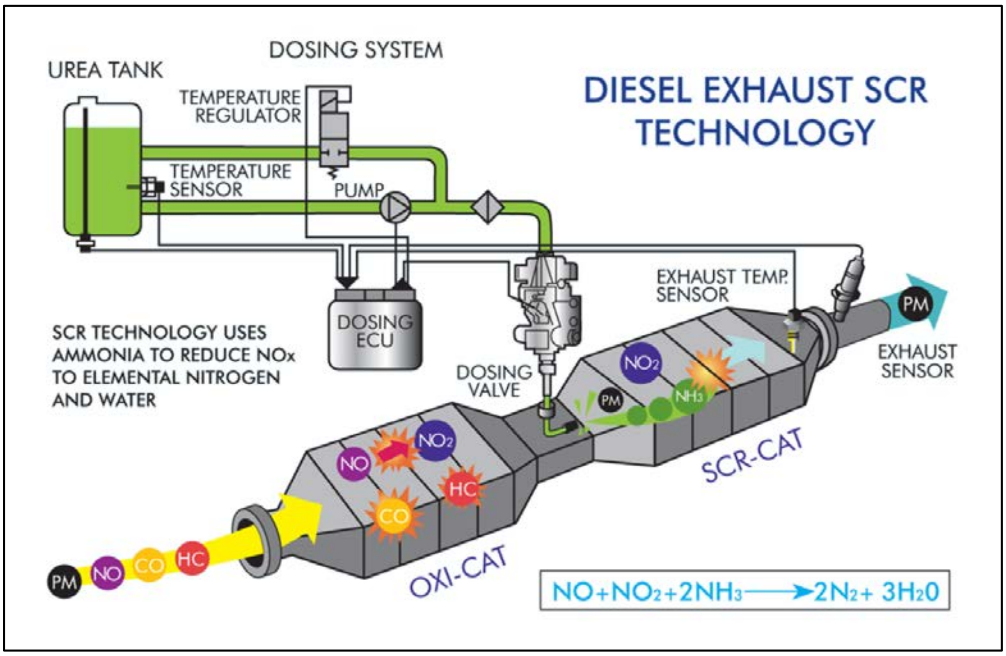

Figure 35 – SCR system “www.bluejaydef.com”

As we can see from the Figure 35, ammonia can react with nitrogen oxide NO and with oxygen to form N and H_2O (vapor). This is just one of the possible reactions that may occur, other ones can take place and can involve NO, NO_2 or both with different reaction rate: it means fast reaction at lower temperature if we are trying to reduce NO_2, and slower reaction or reaction at higher temperature if we are try to reduce NO.

Ammonia is dangerous to carry on our cars or trucks, so typically is produced inside the catalyst starting from something that is less armful and can be more safely and easily stored and carried on board of vehicle: a water solution of urea. Typically is 32.5% urea solution: 1/3 urea + 2/3 water, because it's a composition close to the eutectic composition, the composition with the minimum freezing point.

This solution of urea and water is injected in the exhaust stream by means of a dedicated injector, similar for geometry and characteristics to the PFI injector of gasoline engine without high injector pressure.

So these droplets are injected in the exhaust stream, they mix with hot exhaust, evaporate and they pass through a first section of the catalyst called *hydrolysis catalyst* (this first section is promoting the brake down of the urea molecule into ammonia and CO_2), and then ammonia will react and reduce NO in SCR catalyst.

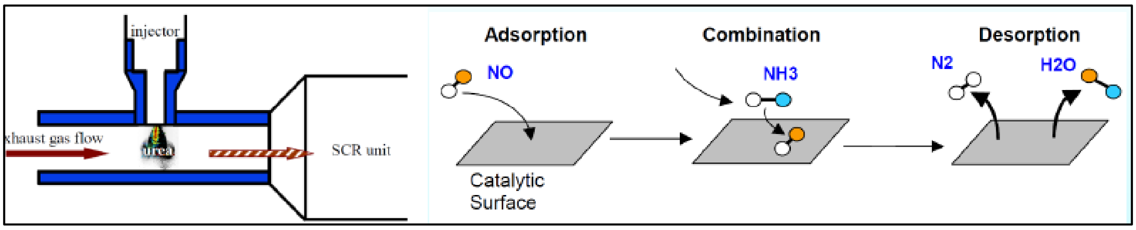

Figure 36 – Catalytic evolution inside SCR catalyst

In the Figure 36 is shown the NO_x conversion in the SCR catalyst:

- Adsorption: NO is typically going to be trapped on the catalytic surface, typically these SCR systems are featuring Zeolites or other components in their structure, that are capable of trapping the molecules;
- Combination: ammonia is going to react with NO;
- Desorption: then the result of the reaction is production of N_2 and H_2O (vapor).

This system require a good dosing of urea or of ammonia, otherwise the risk is that the result will be even worse than the issue that we are trying to solve, because if we inject too much urea, we produce too much ammonia, *ammonia slip* occurs (emission of ammonia) that is worse than the emission of NO_x itself. For this reason, there is a legislation limiting the maximum concentration of ammonia in the exhaust <15 ppm.

The urea consumption, especially in the HD application, is very important because can affect the cost of the transport, so we have to find the right calibration of the ATS[16] in order to find the best trade-off between the BSFC and the urea consumption.

The efficiency of this catalyst is upper than 90% in NO_x reduction and the ratio NO_2/NO_x play an important role in the efficiency.

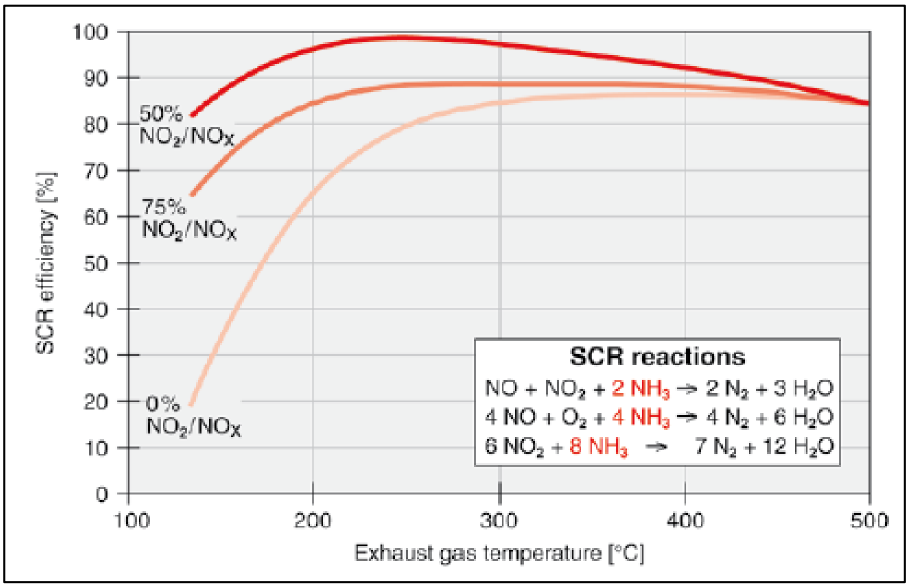

Chart 19 – SCR efficiency Vs. Exhaust gas temperature

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[16] ATS= AfterTreatment systems

3.2.2 Catalyst layout

This is the typical layout of the EU6 application with SCR catalyst in UFP and a DOC+DPF in a single canning in the CCC.

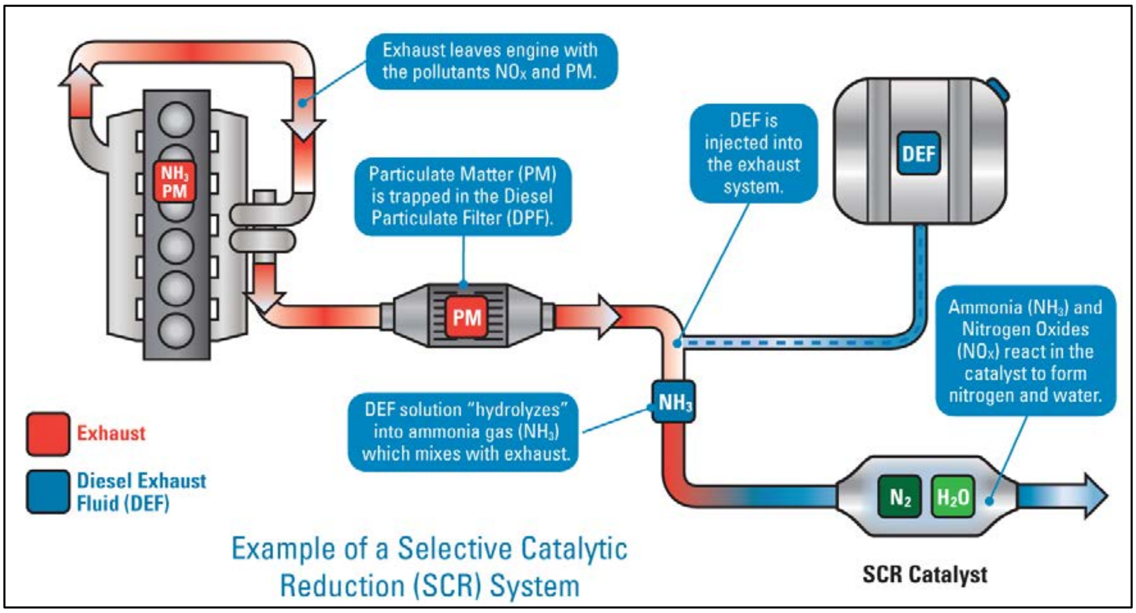

Figure 37 – SCR layout system "www.ThePinsta.com"

Quite often a *mixer* is used in front of the catalyst in order to enhance turbulence mixing between the urea and the exhaust flow, but it increases cost, complexity and backpressure seen from the engine., as is shown in the figure

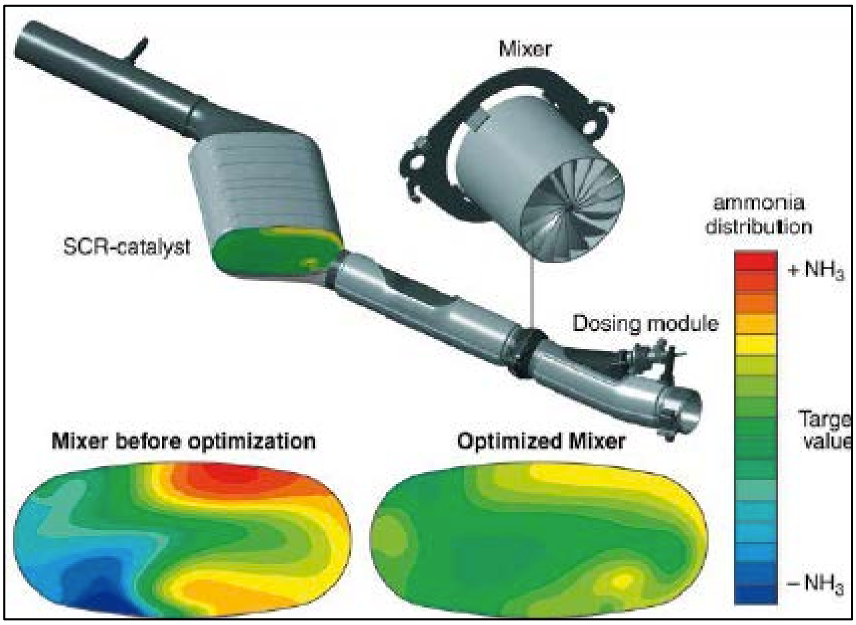

Figure 38 – Mixer for SCR system

We need also a NO_x sensor in order to have a closed loop ammonia dosing strategy, because we need to know how much ammonia we have to inject in order to match our target efficiency and in the same time, to avoid overdose, that it will produce ammonia slip.

3.2.3 *Practical application of SCR*

If we can see the 2 trade-off PM/NO_x and BSFC/NO_x, if we use the SCR we search to stay with the calibration in the area of very low PM and BSFC (orange circles in the figures) so with high level of NO_x thanks to the efficiency in NO_x reduction.

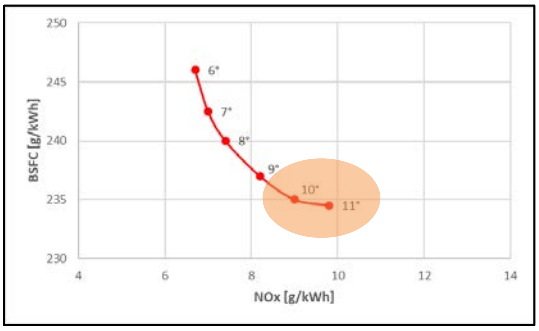

Chart 20 – Nox Vs. BSFC

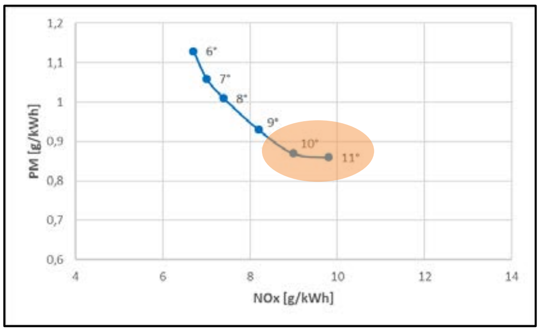

Chart 21 – Nox Vs. PM

In some HD application, we can eliminate the EGR systems for having the lowest as possible soot emission and break down the level of NO_x only with the SCR systems.

3.3 Selective Catalytic Reduction with hydrocarbons “deNO$_x$ or lean NO$_x$ catalyst”

It has been proposed for diesel engine: the idea is to use unburned HCs to reduce NO$_x$, instead of ammonia.

3.3.1 Different DeNOx catalyst types

This can be basically achieved in two ways:

- Passive system: it just uses the unburned HCs normally produced by the engine. Typically this system has a quite poor conversion efficiency because in a diesel engine the amount of unburned HC are not so much, some tens of ppm that are typically not enough to achieve good conversion efficiency. We need a ratio between unburned HC and NO$_x$ of 10:1 (10 unburned HC and 1 NO$_x$ molecules) to have a good efficiency. Instead we have a ratio below 1. We are emitting less unburned HC than NO$_x$. Typically this strategy can’t be applied.

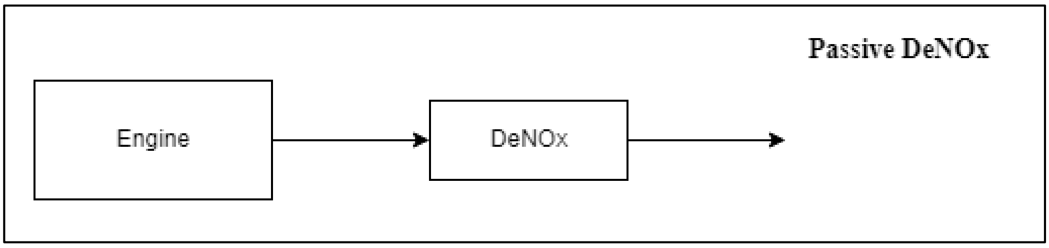

Figure 39 – Passive DeNOx

- Active system: we can achieve better conversion efficiency if we can inject some additional HC in the exhaust streams. The additional HC injection can be done in two ways:
  - By means of an additional injector placed in the exhaust manifold that inject some additional fuel into the exhaust;

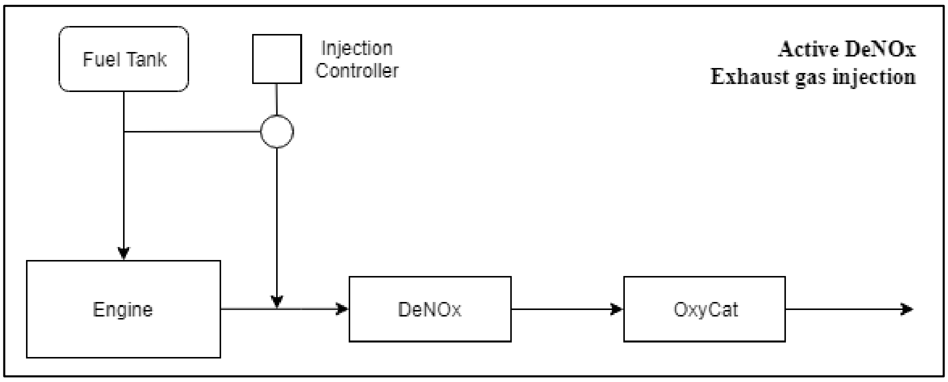

Figure 41 – Active DeNOx (Exhaust gas injection)

- By means of the injection of the common rail system performing a post injection (injection during the exhaust stroke close to BDC). In this case we do not need any additional injector. We can manage it by means of the normal injection system. The problem of this system is wall impingement because the fuel injection happens at BDC when the air density inside the chamber is very low so the fuel injected at high pressure impacts the liner and produce lubricant oil dilution.

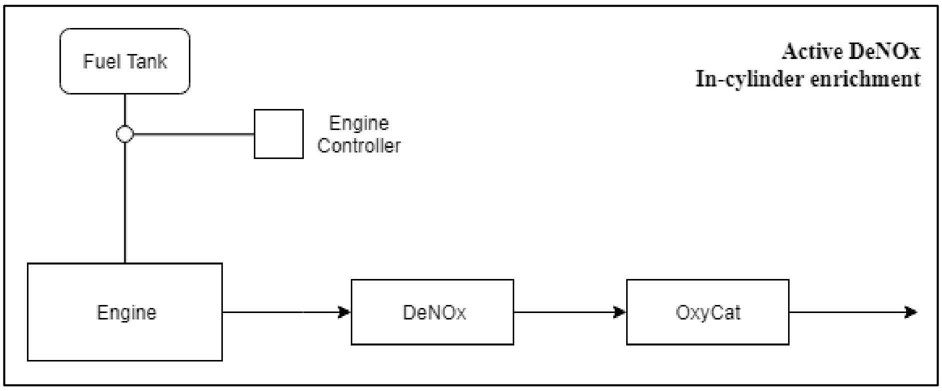

Figure 40 – Active DeNOx (In-cylinder enrichment)

3.3.2 Catalyst features

Two different classes of DeNOx catalyst have been developed:

- Platinum-based catalysts “Pt/Al_2O_3”;
- Base metal catalyst “Cu/ZSM5”.

Each of these catalysts has its specific drawbacks, related primarily to narrow temperature windows, insufficient thermal durability and/or sulfur tolerance.

More importantly, peak of NO_x conversions are in the range of 40-60%.

Combined with relatively narrow temperature activity windows, these numbers translate to only about 10-20% NO_x conversions of passive systems on regulatory test cycles, such as the ECE/EUDC.

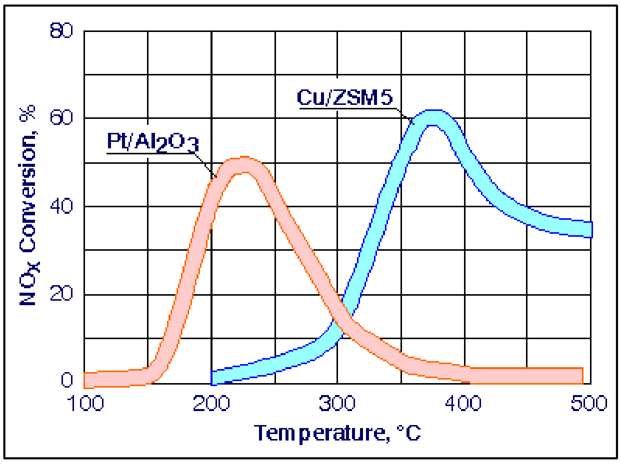

Chart 22 – Temperature Vs NOx base on the PGM

Therefore, DeNOx catalysts are not currently seen as a technology capable of coping with the stringent future emission targets.

3.4 NO$_x$ absorber catalyst systems or Lean NO$_x$ traps "LNT"

3.4.1 *Catalyst features*

Lean NO$_x$ trap is a chemical trap: traps NO$_x$ into the catalyst in a chemical way. It is similar to TWC and DOC, the difference is in the chemical composition of the wash coat, because the wash coat of lean NO$_x$ trap contains barium oxide that can trap nitrogen in the form of NO$_2$ formed by the catalytic oxidation during lean operating conditions.

The working phases of this catalyst are the following:

- *Loading phase*: during the engine normal operation mode, LNT traps NO$_x$ and it oxidizes HC and CO. Typical capacity of LNT in the fresh state is on the order of 2 g of NO$_x$ per liter of catalyst volume.
  After some hundreds of seconds, the barium oxide sites will be completely saturated and traps have to be regenerated.

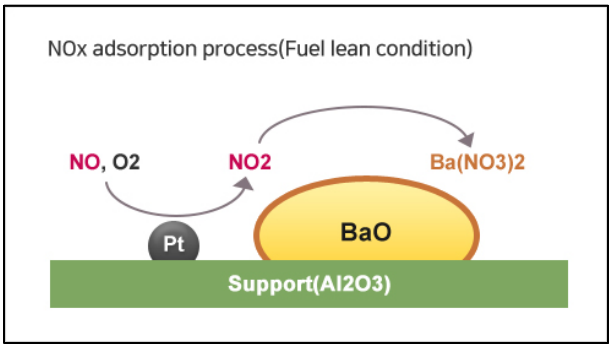

Figure 42 – Nox adsorption process "www.hscatalysts.com"

- *DeNO$_x$ Regeneration phase*: short duration phase (5-8 sec) performing by feeding the catalyst with some reducing species such as unburned HC, CO and H$_2$ rich mixture, where LNT is cleaned up (NO$_x$ is removed) and its storage capacity is recovered. In diesel engine it's impossible to run the combustion with λ_{stoich} or $\lambda<1$, because otherwise it will exponentially increase the soot emission. For this reason, the reduction phase is obtained by means of an additional fuel injection into the exhaust, that can be performed with an additional injector placed in the exhaust or with the injectors within the

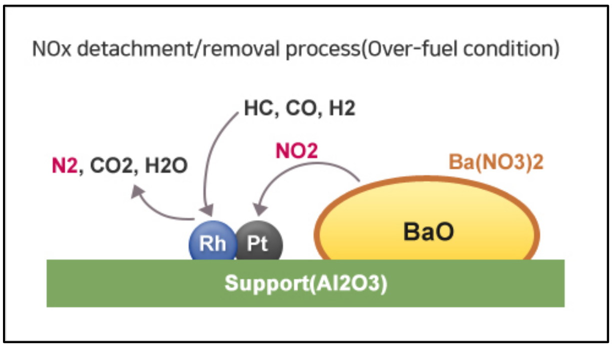

Figure 43 – Nox detachment/removal process "www.hscatalysts.com"

engine by performing a post injection (an additional injection at the end of the expansion stroke).

Depending on the engine emissions, catalyst size and condition, desired NO_x reduction, the regeneration must be performed approximately every 30-120 seconds. During NEDC, about 10 regeneration events occur, one every 2 minutes.

- *DeSO$_x$ Regeneration phase:* needed to recover LNT from sulphur deposition, by means of a dedicated combustion mode, rich mixture and higher temperature. It's performed with lower frequency (one event every 1000km) with respect to DeNO$_x$ Regeneration phase.

LNT control needs also an additional sensor with respect to EU5 system: a lambda sensor downstream LNT catalyst.

The activity of lean NO_x trap covers a wide catalyst temperature window, extending from about 200°C (T1) to 450-500°C (T2). At low temperature there is a negligible NOx conversion efficiency because temperature is too low to be able to oxidize NO into NO_2 and to combine NO_2 with barium oxide to form barium nitrates.

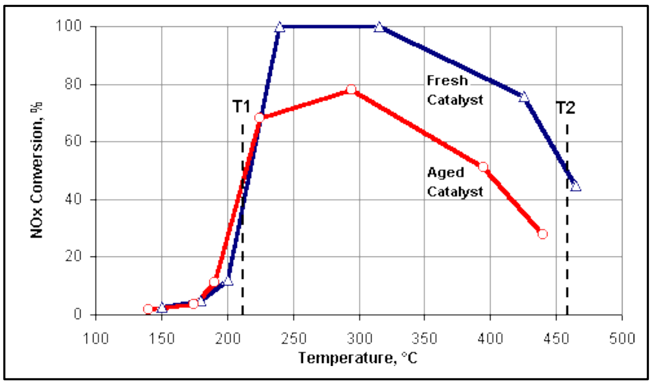

Chart 23 – Temperature Vs. NOx convertion b/w fresh and aged catalyst "www.DieselNet.com"

At higher temperature the NO_x conversion efficiency will drop because the nitrates became unstable and therefore tend to be released.

3.4.2 Catalyst layouts

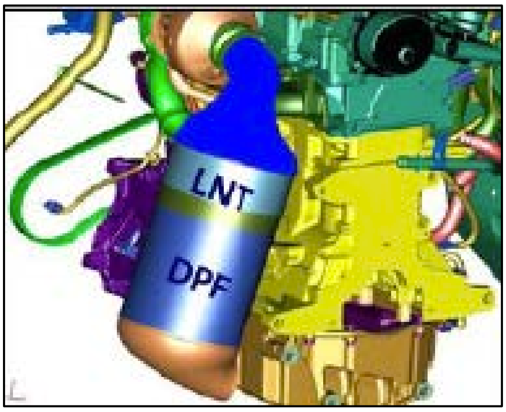

Figure 44 – LNT + DPF "Millo's notes"

LNT has the same structure of the Oxidation catalyst DOC and will replace it in close-coupled position taking care on NO_x reduction and HC and CO oxidation in the exhaust gas.

Due to the sulphur poisoning, is possible to have a Sulphur trap in close-couple position and a NO_x Adsosorber in underfloor position.

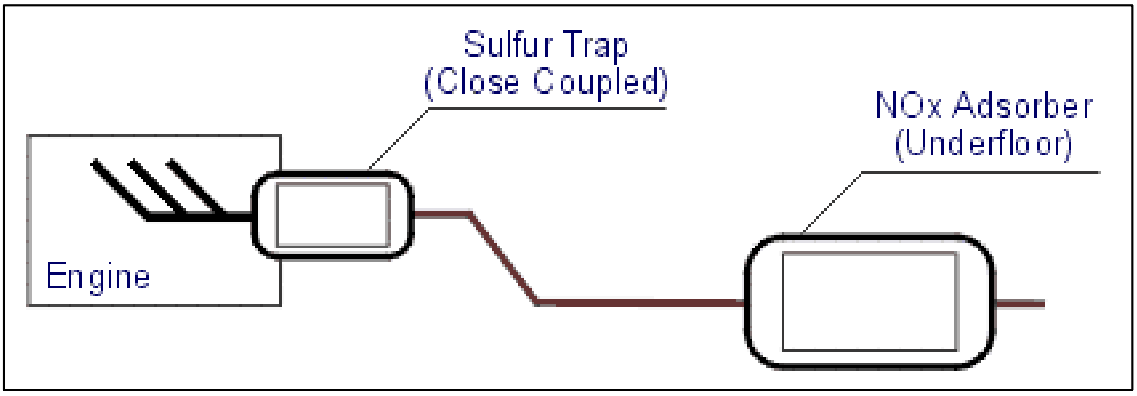

Figure 45 – Sulfur trap + NOx adsorber "www.Dieselgate.com"

3.5 Comparison between SCR and LNT

3.5.1 *Performance differences*

This is a comparison of the operating window of different diesel catalyst technologies.

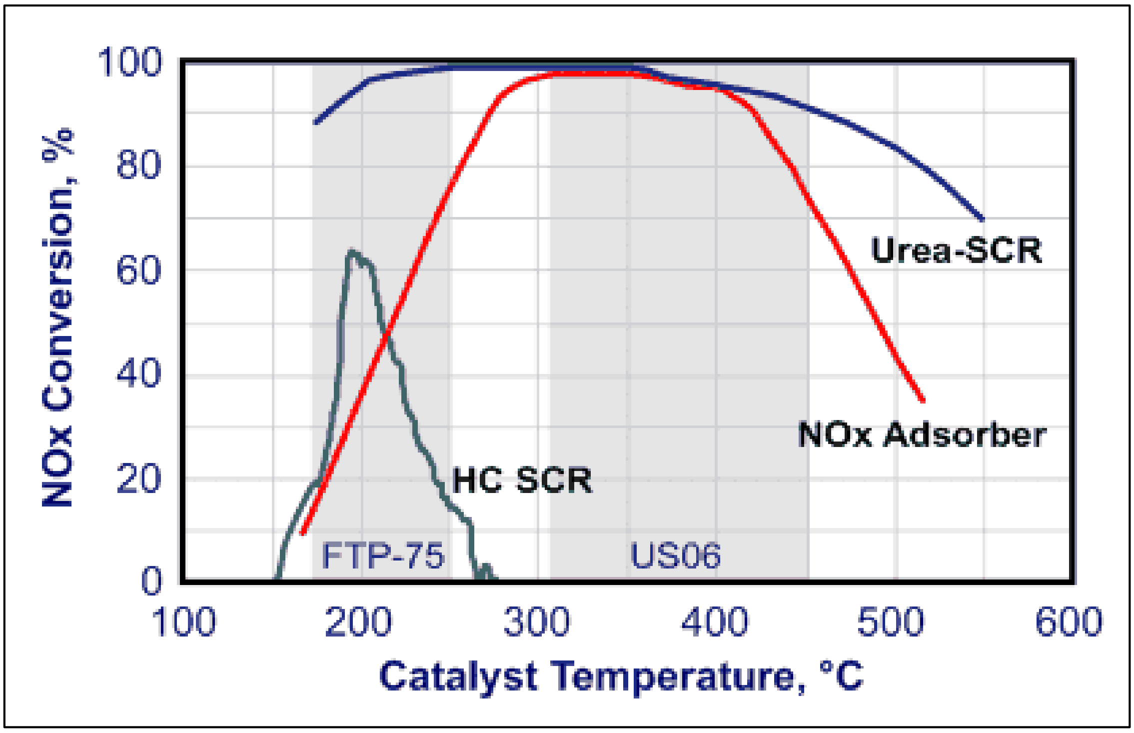

Chart 24 – Catalyst' conversion efficiency b/w HC SCR, NOx adsorber, Urea-SCR "www.DieselNet.com"

The SCR based on HC has the narrowest operating window and can match only just few applications such as FTP-75 for passenger cars, but if we just consider the US06, which is more aggressive with higher temperature, the temperature will typically drop in this temperature range and here there is no way to use this technology.

3.5.2 *Cost differences*

The cost of the LNT is typically lower than the SCR. The chart shows the items cost for LNT, instead the diagram compares the cost of LNT and SCR in function of engine displacement.

The cost of 2L engine displacement is above 400$, 10% lower than SCR for the same engine size, but the cost items are essentially the precious metal and in particular Rh to

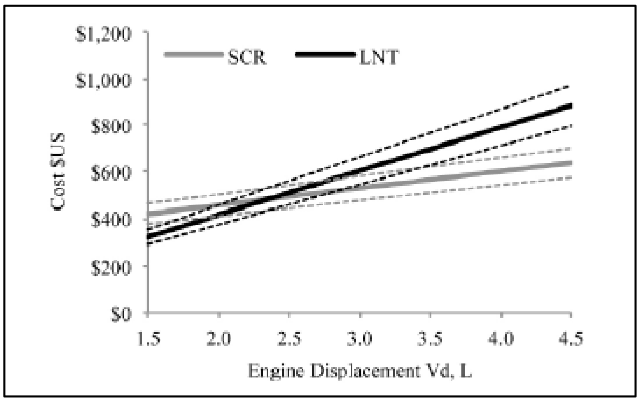

Chart 25 – Engine displacement Vs. SCR and LNT cost "Millo's notes"

enhance the reduction reaction. Rh is extremely expensive: it costs more than 3 times than the Pt. If we sum up the cost of PGM and the cost of wash coat, this is almost the overall cost of the system. Of course we need an additional oxygen sensor to control the system, but the cost is mainly related to the PGM.
Completely different from SCR, where cost is mainly due to the accessories, here the cost depends on the cost of the LNT catalyst.

The control of LNT is quite complex as for SCR, but we have some advantages: diesel fuel is used as reductant, so we don't need control on the level or on the quality of the reductant, because the reducing agent is the fuel itself; also dosing is simplified because can use the same injector.
All those aspects simplify a lot the system.
We need a couple of lambda sensors: upstream and downstream the LNT to manage the rich and lean operations. We need also two temperature sensors to control the operation of the LNT upstream and downstream the LNT.
It's extremely important control the temperature during the desulphation, avoid that we reach too high temperature.
The temperatures have to be kept within a window at a level high enough to remove sulphur and low enough to avoid danger of aging or damaging the catalyst.

3.5.3 Catalyst layouts: integration of all NO_x reduction catalyst

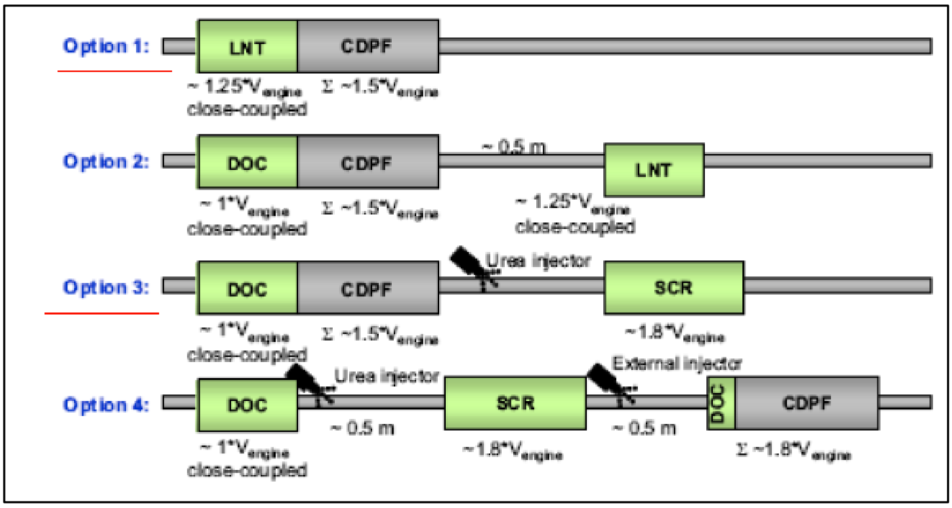

Figure 46 – Different catalyst layouts "[13]"

There are four possible layouts:

1) LNT+DPF mounted in CCC. It's basically the same architecture used in EU5 (DOC+DPF) but in this case an LNT is used instead of a DOC.
2) DOC+DPF in CCC +LNT in UFP: it isn't a good layout because the LNT in underfloor is out of the working temperature.
3) DOC+DPF in CCC + SCR in UFP: with this solution we can exploit the larger operating temperature of the SCR with urea and use the DOC to regenerate the DPF.
4) SCR in CCC+DOC+DPF in UFP. Some OEMs, such as PSA group, is currently using it. The reason is pretty simple, because they are the only one using fuel borne catalyst FBC: it's an additive to lower the oxidation temperature of the soot trapped in the DPF, so they are the only one that can more easily afford the usage of an underfloor DPF. Normally placing the DPF far from the engine, require more extra fuel to reach the temperature at which soot can be oxidized. In this way they can benefit of having the SCR closer to the engine, therefore it can be more rapidly activated over the driving cycle, while normally cars can't inject urea in the SCR system until they reach the extra urban section EUDC of the driving cycle.

3.6 Diesel Particulate Filter "DPF"

3.6.1 Catalyst features

It's a mechanical filter for the particular matter PM, it's mechanically blocking/trapping particles in the exhaust stream.

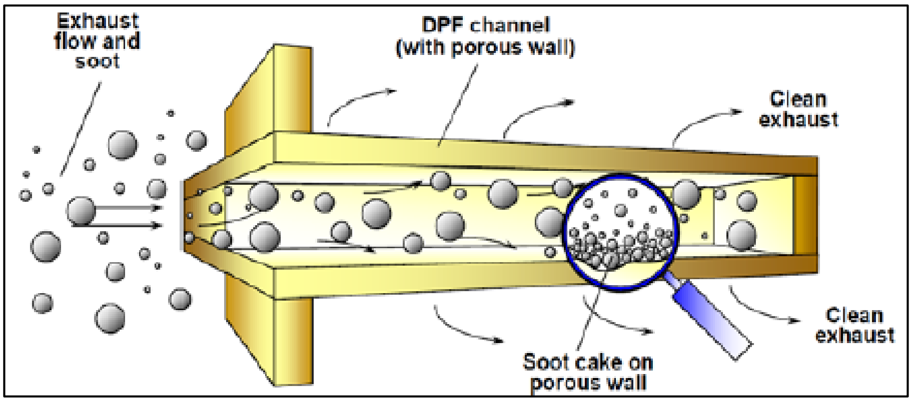

Figure 47 – DPF filter explanation "Millo's notes)

It has a wall-through monolith: the cells are open at the inlet, closed at the outlet and they are surrounded with neighbor channels, which are closed at the inlet and opened at the outlet.

The exhaust gasses enter in the channel, the opposite end is closed, so the exhaust gasses can get out from the channel only passing through the channel walls. The walls are porous, while gas can pass through them, solid particles will be trapped on the wall. The following diagram shows how the DPF looks like in comparison with the flow-through catalyst

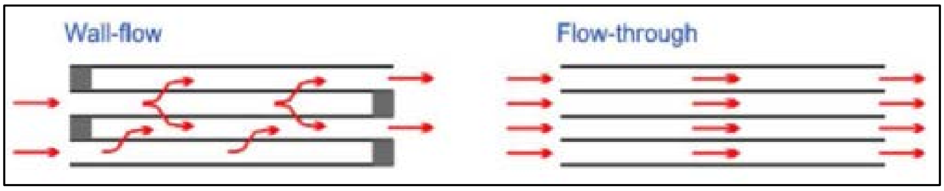

Figure 48 – Type of catalyst channels "Millo's notes"

The typical filtration process occurs in 2 steps in time sequence:

- _Deep bed or depth filtration_: particles penetrate inside the porous structure (channel walls) and fill in all the pores;
- _Soot cake_: After the pores has been filled in and saturated, the layer of soot start accumulating on the wall surface. This soot layer progressively grows up and it contributes significantly to the filtration mechanism, through the so-called soot cake.

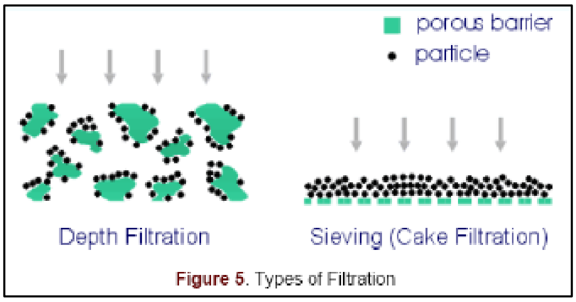

Figure 49 – Mechanism of DPF filtration

If we keep the engine at constant operating condition (constant load and speed) and we just measure the pressure drop across the DPF, we see:

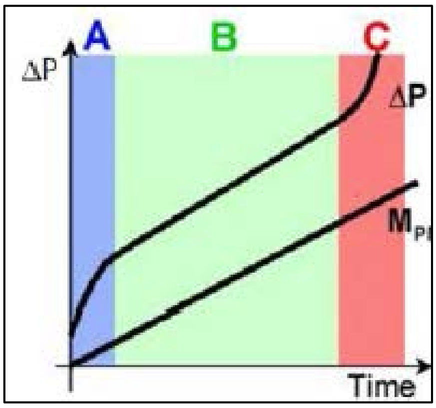

Chart 26 – Pressure drop inside the DPF “Millo’s notes”

A. Initial fast rise of pressure drop during the filling of the porous structure [A] (deep bed filtration);
B. Almost linear increase of the pressure drop across the filter [B], because, if we run the engine at constant operating condition, we have constant soot production and soot cake will be more and more thick;
C. Until we reach a region [C] where we need to regenerate the trap because the pressure drop is too high.

The soot distribution inside the filter changes with the working operation point, because also the soot composition changes over the engine working conditions. So the accumulation of the soot on the filter is different if we are in steady-state or in transient operation.

The properties of the soot cake are depending on the Péclet number; or in other words, the higher the velocity of the flow through the channel, the thinner will be, so the less permeable and denser will be. On the contrary, the lower the speed flow, the softer but ticker and more permeable the layer of soot will be.

3.6.2 Catalyst control strategy

3.6.2.1 Accumulation phase

We need to know how much soot we have on the trap before starting the regeneration events, because it's not just a matter of regenerating the trap when we have reach a certain pressure drop, but it's a matter of also protecting the trap from being damaged. Because if we accumulate too much soot on the trap before regenerating, the risk is that during the regeneration we may reach extremely high temperature and damage the trap.

The problem is that the ECU can't detect the amount of soot accumulated on the trap on the basis of the pressure drop, because, the characteristic of the soot is different depending on the engine operating condition during soot accumulation: it means different behavior of the soot layer and different pressure drops.

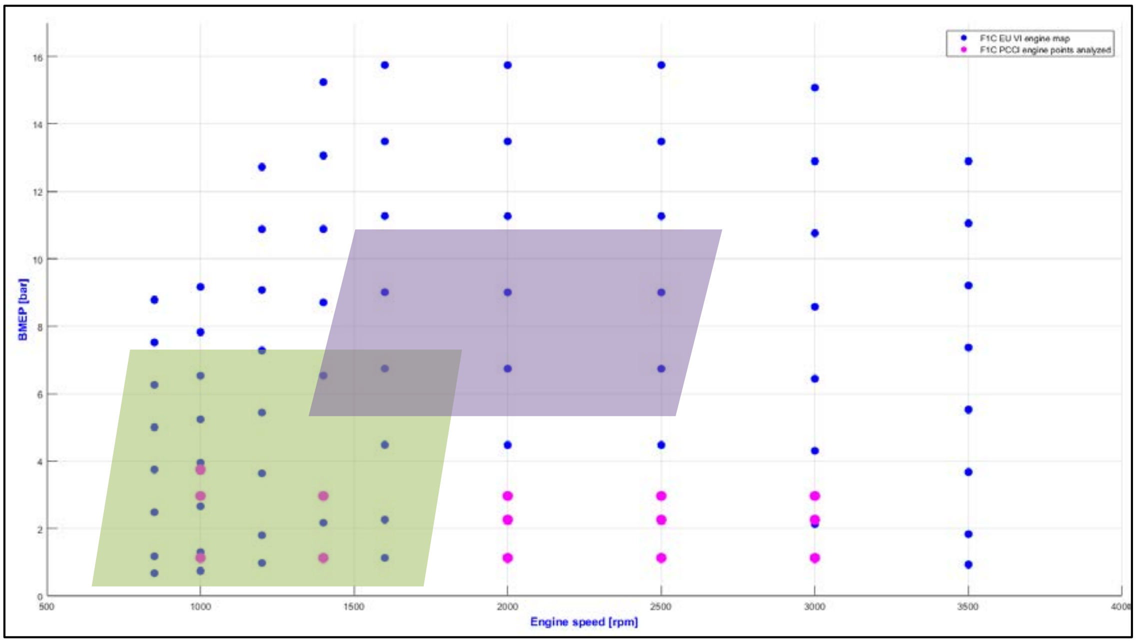

Chart 27 – Engine map with the DPF working areas

So the ECU will typically use a statistical approach by dividing the engine operating map in 2 zones (green and purple areas of the Chart 27) and for each zone the information of the soot production is stored in the ECU, that by simply integrating the information in the time domain, it can evaluate how much soot is going to be produced and therefore how much soot is going to be accumulated into the trap.

When a certain soot level has been reached, ECU starts to identify the right time for regeneration opportunity: when the maximum level will be reached, a regeneration starts, regardless if we are in a favourable condition or not.
The most favourable conditions is the highway at constant velocity, this situation allows the low as possible increase of fuel consumption for the regeneration phase.

3.6.2.2 Regeneration mode

The regeneration mode is the operation allowing the clean-up of the DPF.

The ECU has a second map concerning the trap regeneration efficiency: it calculates also the amount of soot that has been removed during the regeneration event, so that the total mass of soot inside the DPF is known by the ECU at each time, because we may start the regeneration event and then, for several reasons, we switch off the engine.

For this strategy, there is the division of the engine load map in two regions, used for ECE and EUDC parts of the NEDC.

The regeneration process is generally occurring in two following steps:

- *First step*: the combustion process is shifted towards the retarded direction, so all the injection pulses are postponed as much as possible in order to increase the temperature of the exhaust.
  Of course this have to be done without any torque variation perceived by the driver (torque neutral).
  The postponed combustion process will lead to reduced combustion efficiency, leading, for the same amount of burnt fuel, to a lower torque, so the ECU needs to increase the quantity of the main injection to compensate for the reduced efficiency and to keep the torque delivered by the engine at a constant level. Additional measures can be taken playing for instance on the EGR quantity, on

the air flow rate: throttling may be needed in order to increase the temperature of the exhaust.
This operation with a retarded injection increases the temperature of the exhaust, is maintained until the temperature sensor, which is placed on the DOC, confirms that the DOC has reached the light-off temperature for unburned HC (at least 250°C).
Only when we have a confirmation, we start the second phase of the regeneration process.

- *Second step*: a post injection is added, typically performed close to BDC, at the end of the expansion stroke, when the exhaust valve is already open. This is done to produce HC that are going to be burned in the DOC, which, in this phase, is operating as a kind of catalytic burner. So the purpose of the DOC in this phase is to oxidize this unburned fuel to increase the temperature upstream of the DPF, rising the temperature at the inlet of the DPF at level high enough to produce soot combustion, so above 600-650°C.

The ECU search to do the regeneration phase during the minimum of the resulting fuel penalty, that is the sum of the fuel penalty due to the pressure loss and the fuel penalty due to the temperature increase.

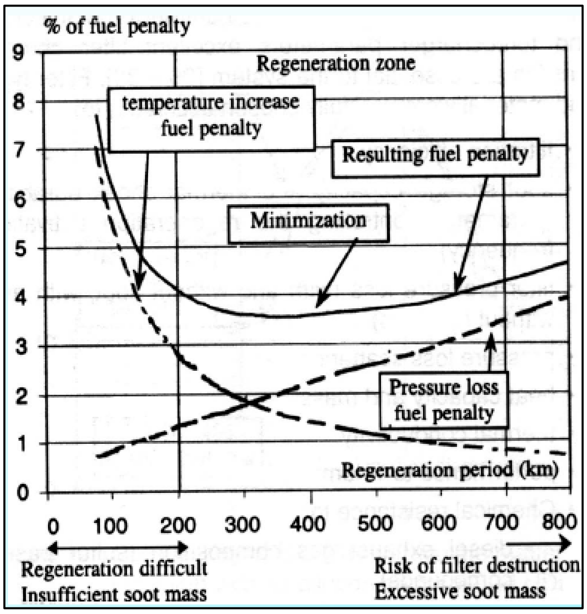

Chart 28 – Resulting fuel penalty during regeneration phase of DPF "Millo's notes"

This phase can create some problems:

- *Increase of fuel consumption*: fuel is wasted just to increase the temperature of the exhaust. Typically, this fuel consumption increment is about 2-3%. Different from other regeneration phase, this occurs with a very low frequency, but requires a significant amount of time.
- *Increase of lubricant oil dilution*: by using post injections, oil dilution will increase because the fuel injected close to BDC will impact on the liner and will dilute the lubricant.
- *Risk of DPF damages in case of uncontrolled regeneration event*: if happen the drop to idle, the most dangerous event of uncontrolled regeneration, we can have an increasing of temperature that can damage the DPF. This is the reason why the DPF has a *soot limit*, a safety threshold that represents the maximum amount of soot that can be accumulated over the trap before the need to regenerate and it depends on the material used for DPF.
- *Risk of accelerated aging or damages to other after treatment components placed downstream of the DPF*: during the regeneration of the DPF, there will be very high temperature in the exhaust that might produce an accelerated aging or even a permanent damage of other catalytic devices (e.g. SCR).
- *risk of engine malfunctioning due to DPF clogging in case of failed regeneration event:* the ECU might not be able to achieve regeneration for some reasons: because there is a damage to a temperature sensor and the measurement of the DOC temperature is compromised; it might happen that the regeneration would never start because the correct value of temperature is not read and this increases the amount of soot being trapped over the DPF without regeneration event, this leads to a not-well-controlled regeneration and to possible engine malfunction for the increase backpressure level at the exhaust.

3.6.3 Catalyst layout

The DPF layout for EU5 and EU6 applications is in general close coupled position CCC into the engine compartment, mounted in the same canning with the DOC, typically mounted directly at the turbine outlet.

This allows also the adoption of long route EGR, because we can take the exhaust gases being recirculated from the output of the canning, which is typically quite close to the inlet of the compressor, so the long route becomes reasonably short, or short enough to be feasible.

This architecture has the advantage of:

- *Limited heat losses*: remember that 1m of pipe means 100°C drop in temperature, so if you put the after treatment device as close as possible to the exhaust, heat losses will be minimized, and therefore, the amount of fuel wasted during regeneration process by post-injection can be minimized too. As a consequence, the oil dilution issues and the fuel penalty can be both minimized.
- *Faster response time*: the time needed to warm up the catalyst will be shorter.

The drawbacks, related to the constraints that such installation creates, are:

- *Limited DPF volume*: that means limited possibility to accumulate soot and this means that the trap needs to be regenerated more frequently (higher regeneration frequency).
- *Limited DOC volume*: that means higher risk of HC slip, because the space velocity inside the DOC is high (as consequence of the limited volume), so the DOC may not be able to fully oxidize unburned HC, due to the limited residence time of the gases in the DOC. The lower space velocity is compensated by higher temperature during soot combustion.

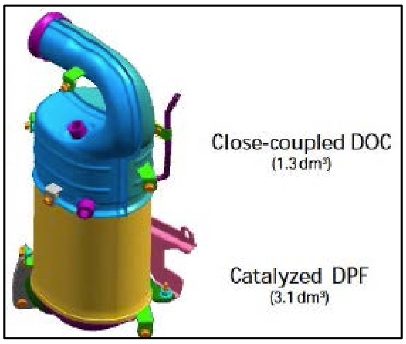

Figure 50 – DPF layout "Millo's notes"

4 Advanced diesel combustion strategy

Nowadays the legislation limits are always more stringent and we know that all the actions that we take to lower NO_x lead to a worsening of soot emission and vice versa: this is what we can see in Kamimoto diagram.
The diagram is just a different representation of it: instead of using the typical temperature vs. equivalence ratio representation, it shows temperature vs. local air excess ratio λ (that is the opposite of the equivalence ratio).
For this reason the soot peninsula is not on the upper part of the diagram, but in the bottom part, and NO_x region appears in the right upper corner.

We said that techniques able to lower the NO_x usually lead to higher soot: an example may be the EGR: if the EGR rate is increased, the temperature drops down and we move away from NO_x region but, another consequence is the reduction of oxygen availability, so that we may end up in the soot region ($\lambda<1$).
Similar remarks may be done for other actions, such as the delayed combustion: by retarding the combustion, temperature will decrease and soot emissions will increase (no time for soot burnout).

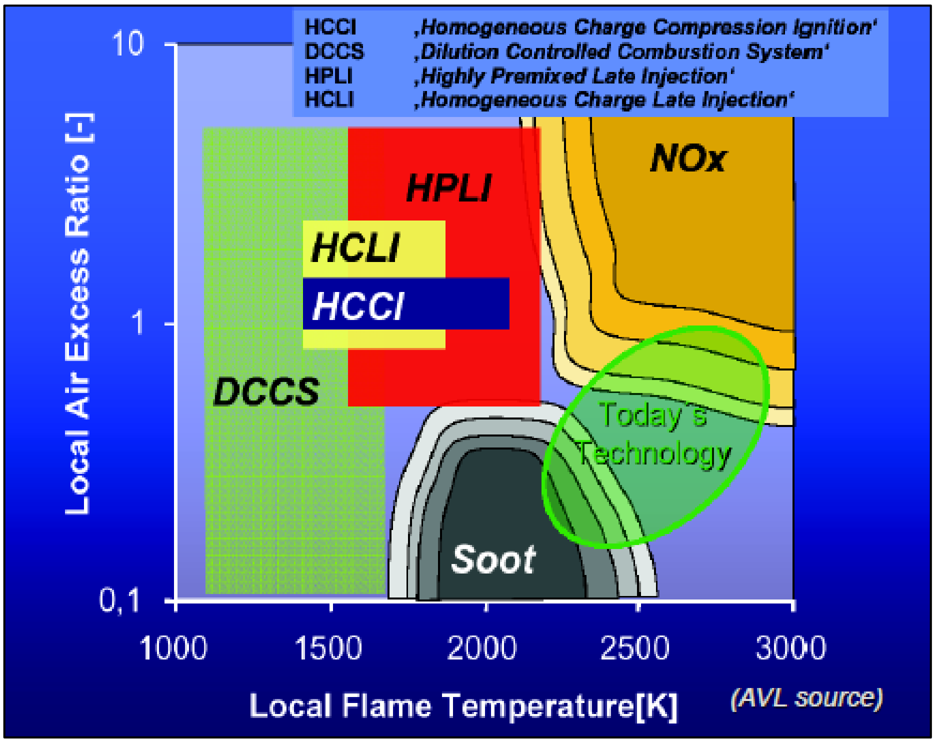

Figure 51 – Different combustion modes on the Kamimoto diagram "AVL"

With the current operating regions (today's technology), we have to define a trade-off between soot and NO_x emissions, but as soon as we move away from this region, soot or NO_x will increase.

The idea may be to operate the engine in these other areas with the advanced combustion strategies:

- HPLI "Highly Premixed Late Injection";
- HCLI "homogenous Charge Late Injection";
- HCCI "Homogeneous Charge Compression Ignition";
- DCCS "Dilution Controlled Combustion System".

4.1 HCCI "Highly Premixed Late Injection"

The basic idea of HCCI is to fill the cylinder with a highly diluted, extremely lean and homogeneous mixture, that can auto ignite at TDC.
If the air, fuel and dilutant are properly dosed and mixed together, the mixture has low chemical reactivity. It means that it can be compressed without spontaneous ignition during the compression stroke and it will spontaneously auto ignite at TDC with a standard CR, leading to a low temperature combustion process (smokeless and NO_x-free).

In this case we would be able to perform almost constant volume combustion

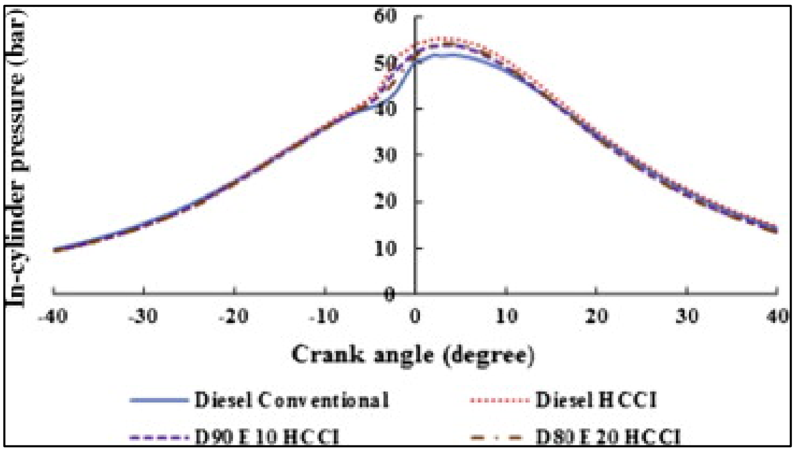

Chart 29 – CA vs In-cylinder pressure "Millo's notes"

This is the best efficiency combustion process that may be achieved.

As regards emissions, since an extremely lean mixture is used ($\lambda = 2\text{-}4$), soot is not going to form with respect to conventional diesel combustion.

At the same time, since the mixture is so lean in HCCI and it is also diluted with burned gases, the peak temperature of combustion will be pretty low because, even if the combustion occurs almost at constant volume, being extremely lean and diluted the mixture, the energy content per unit volume is low and the peak temperature is low. But in this way we can stay below the threshold temperature for NO_x formation and NO_x is not formed.

This type of combustion can theoretically works at low load low rpm, but if the load is increased, the amount of fuel injected increase too so we aren't able to have a good mixing with a right λ for avoiding the soot formation.

Instead, if the engine speed is increased and we have calibrated the mixture for having the ignition at TDC with lower rpm, the compression stroke will be shorter so, at the end, we have an auto ignition delayed respect to TDC.

4.2 PCCI “Premixed Charge Compression Ignition”

PCCI is achieved by injecting fuel with a huge advance before TDC, so that the injected fuel is going to disperse inside the combustion chamber and, even if it will not perfectly homogeneously mixed with air, it will mix pretty well with air giving us premixed combustion mode because, when the combustion will start, there will be fuel and air already premixed together without an overlap between combustion and mixing, as it happens in a conventional diesel.

In a conventional diesel, the injection occurs close to TDC and the injector matches the combustion chamber shape so that the injected fuel remains in the bowl, but in PCCI, injection takes place well in advance with respect to the TDC, when the piston is pretty far from the TDC and pressure and density in the chamber are definitely lower and the fuel jet may impact the liner (the lower the air density at the injection time, the lower the air resistance on spray, so the higher will be the spray penetration). To avoid this, it is necessary to use a narrower angle of injection cone than the conventional design, but a narrower angle gives a worse mixing when the engine is operating as a conventional diesel.

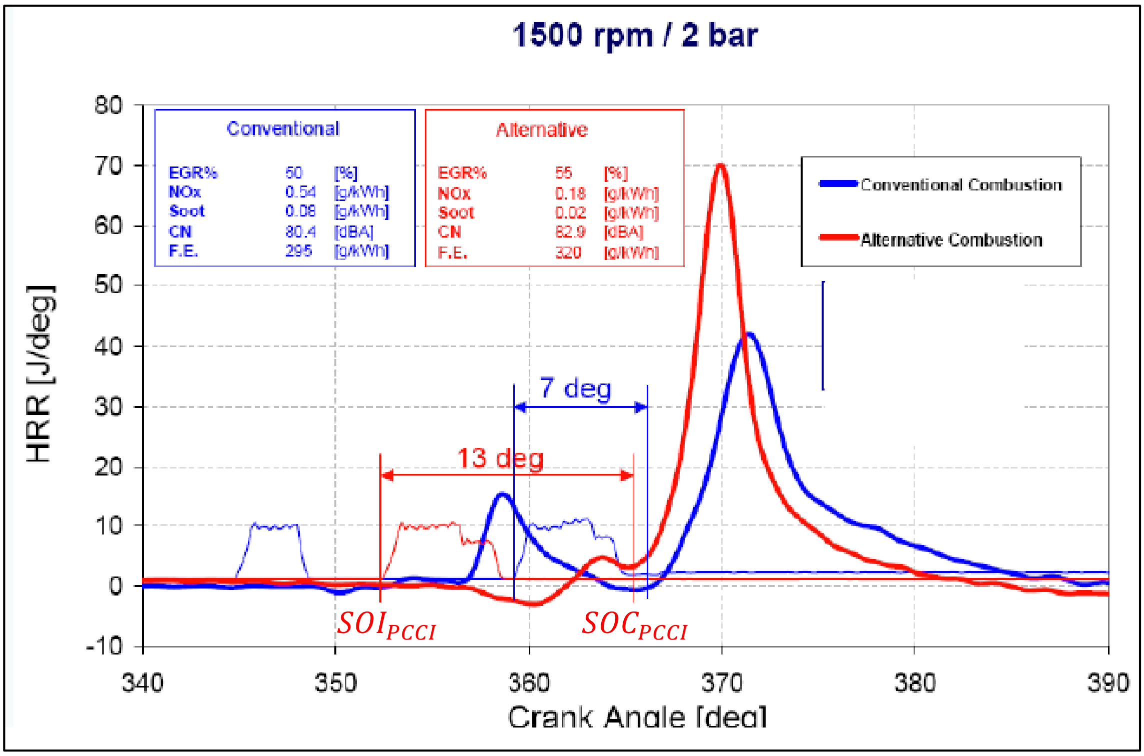

Chart 30 – HRR different behavior b/w conventional and alternative combustion

In the Chart 30 above, we can see a premixed combustion compared with a conventional combustion for a low load – low speed operating point (1500x2).

In order to get a highly premixed combustion, we should remove the pilot because, while in a conventional combustion we use the pilot to reduce the ignition delay, now our target is exactly the opposite, we want an ignition delay as long as possible, and to achieve that we increase the EGR rate in order to have less reactive mixture.

4.2.1 Advantages

Looking at the figure n, it is possible to see the pressure behaviour of the 2 combustion modes, the PCCI has all the mixture that burns together with higher peak pressure and also higher pressure first derivative but the temperature will be lower because the high residuals present inside the chamber act as inert gases so absorb heat. This is good for the NO_x emission that, how we can show from the table, is lower than 1/3.
Also the soot emission is lower thanks to the good mixing so no presence of rich pockets like in the conventional combustion.

Thanks to these pollutant emissions it is possible to use diesel engine with only a DOC as a catalyst because the NO_x and soot are under the limit legislation.

4.2.2 Drawbacks

The first drawbacks that we can aspect is the noise due to the very high pressure first derivative and there no way to reduce it if we want to apply a PCCI combustion mode, because in the conventional combustion mode we use a pilot injection, that in this type of combustion is avoided for having longer ignition delay.

The BSFC is worse than the conventional diesel combustion, because the conventional combustion has a better phasing: there is a pilot injection that releases a significant amount of energy before the main combustion, shifting the “centre of weight” of the combustion in advance. Now we are burning all together, but too late.

This combustion process can’t be use at higher load because the amount of fuel will be higher and so also the reactivity of the mixture.

To keep this mixture quiet until we reach TDC, we need to use more and more EGR rate to keep the mixture sufficiently lean, but, in a conventional system with short route, it's difficult to achieve more EGR and have a good level of boost, because the boost is slowing down.

So it is very difficult to keep the ignition delay longer than the injection duration, and if we want to switch to the conventional combustion mode we have different hardware that isn't optimize for doing both combustion mode, like the injector that has a narrow spray angle for the PCCI or the shape of the piston bowl.

In addition to that, we in general need a closed loop combustion control of the combustion process because if, for instance, we have a different EGR rate in different cylinders, we may have a more reactive mixture in one cylinder and a less reactive one in another cylinder and we may need to compensate in some way to the different reactivity.

We will also need a kind of transition control strategy (or exit strategy) that allows switching to a different combustion mode when the higher loads are approached in a torque neutral way (it means that the driver can't perceive any difference).

It is also difficult to obtain a cold EGR, because the more we recirculate (higher EGR mass flow rate), the lower will be the efficiency of the cooler EGR.

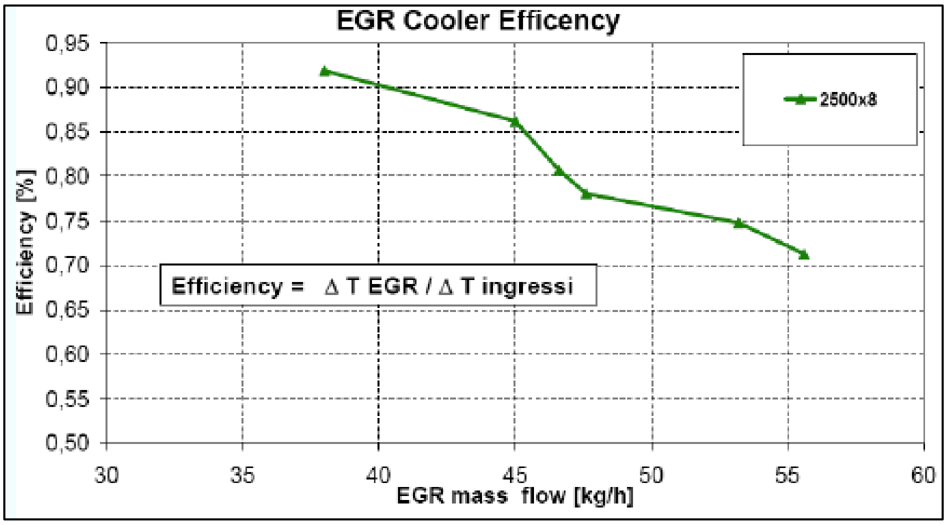

Chart 31 – EGR mass flow vs. Efficiency "Millo's notes"

4.3 HPLI "Highly Premixed Late Injection"

At higher load we can exploit a different combustion regime: the HPLI (highly premixed late injection).

Here the idea is try to mix all the fuel injected before the combustion and using an extremely delayed combustion to avoid having a too huge increase of pressure at TDC, that leads to high temperature producing NO_x, stresses and noise. As we can see in the diagram above, the heat release occurs at 380°–390° CA, so 20-30° CA after TDC, when the pressure is already dropping down due to the expansion.

This non-conventional combustion process has very good results in terms of NO_x reduction with respect to a conventional combustion (-50%). We have almost zero soot, but this is going to be a nightmare in terms of fuel economy (BSFC), that goes from 241 to 279 g/kWh, and again, it will be necessary a closed combustion control.

Also in this case, we need again a closed loop combustion control and a transition control strategy.

4.3.1 Combination of early and late injection

Now we see these 2 examples of HPLI premixed combustion with early injection (left) and with late injections (right) on the Chart 32:

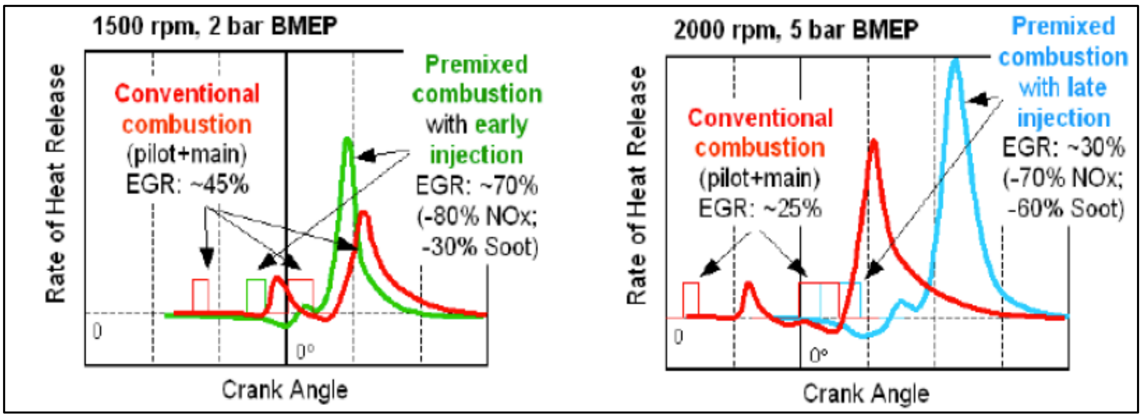

Chart 32 – Rate of Heat release difference b/w Conventional combustion and premixed combustion with early or late injection "Millo's notes"

Here, in Chart 33 and Chart 34 we can see the results in terms of instantaneous NO_x emissions during the first portion of the NEDC (ECE):

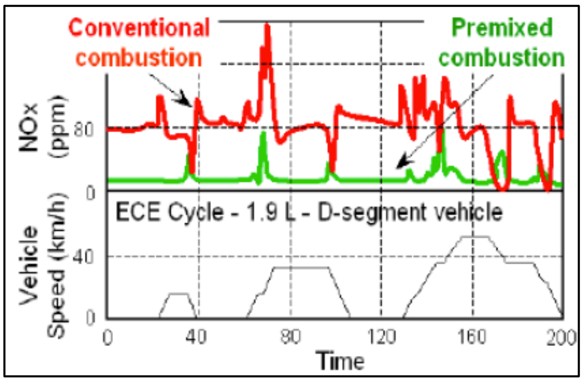

Chart 34 – Instantaneous NOx emissions "Millo's notes"

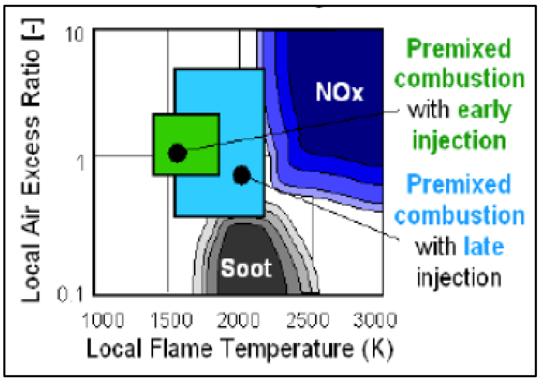

Chart 33 – Areas of 2 HPLI combustion mode on the Kamimoto diagram "Millo's notes"

This is a portion where the load of the engine is so low that you can use a highly premixed combustion and the emission of PCCI are lower than the conventional combustion emission by a factor of 2-4 times.

These are the regions on the Kamimoto diagram on which we work: lean mixture low temperature combustion, but we need, as said before, a transient control strategy.

Using the 3 combustion mode, early injection, late injection and conventional combustion can be difficult because the transition between each combustion mode has to be without any torque variation, so this is very difficult to be done.

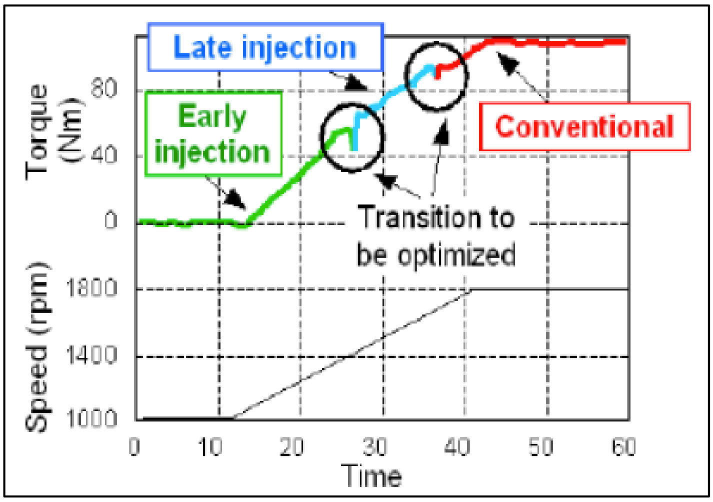

Chart 35 – Time vs Torque 3 different combustion modes linked "Millo's notes"

4.4 Smokeless Rich Combustion

This type of combustion ca be achieved by using a large amount of cooled EGR under a near stoichiometric and even in a rich operating condition.

The soot can be suppressed, because soot formation doesn't need only rich conditions, but also a certain temperature, below or above a certain temperature value, soot is not going to form because, the dehydrogenation reactions inside the hydrocarbon molecules don't take place.

Thanks to the rich condition the NO_x hasn't a good environment to be form so we have no NO_x pollutant emissions

4.4.1 Comparison between High smoke and Smokeless combustion

These Kamimoto diagrams show what is going to happen in different moments of the combustion process.
These points are the temperature and the equivalence ratio local conditions that you have inside the combustion chamber at a certain moment during your combustion process.

We can obtain this by a CFD analysis subdividing the volume of the combustion chamber in sub volumes and in each one we calculate the local air fuel ratio and the local temperature

In the High smoke combustion, the region "I" is rich but at low temperature. We see that as the combustion process progress, the temperature is increasing and we are progressively ending up in the soot peninsula.

Combustion process

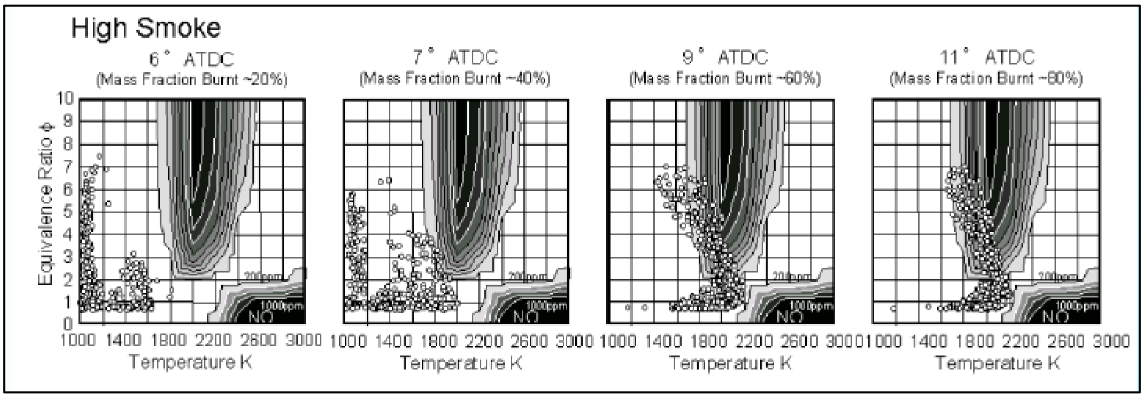

Figure 52 – "High smoke" combustion process "Spessa's notes"

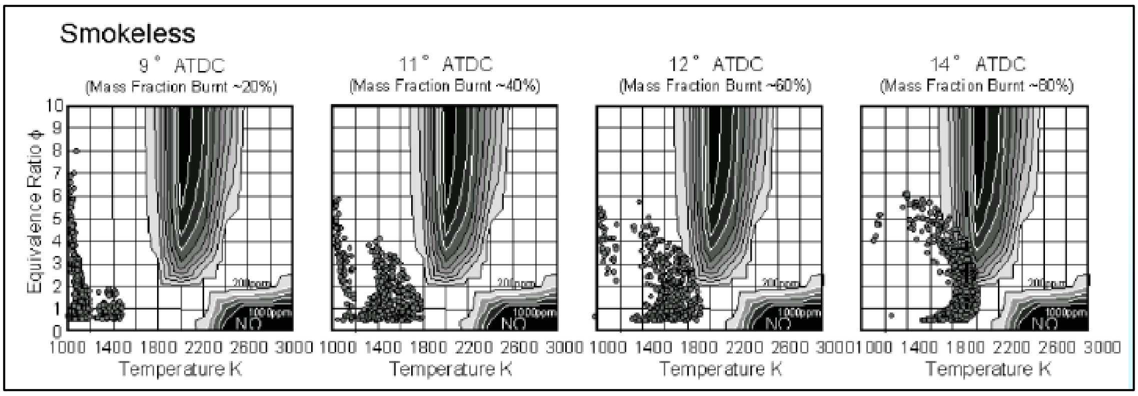

Figure 53 – "Smokeless" combustion process "Spessa's notes"

This is what happens if you manage to keep the temperature low. We are still rich, because we are still in this region "I" but, if we keep the temperature low, we will just attain the soot peninsula and we will not enter the soot peninsula.

5 Experimental set-up

The whole experimental activity is done in the DENERG department, at PolitTo[17], in which there is the *ICE Advanced Laboratory*.
In this laboratory is present the all instrumentation for developing an R&D project on the engine.

In particular, my activity is based on the engine *FPT F1C* modified for working in PCCI combustion mode, provided by FPT Industrial of Arbon (CH).

All the instrument involved in the project are made by AVL and I will discuss better in the following chapters.

In addition, the engine is instrumented by additional sensor for controlling better all the value of temperature and pressure during the combustion process.

5.1 The engine

Our engine is based on the FPT F1C EUVI 4 cylinder in line, diesel engine that has 3.0L, 129 kW @ 3500 rpm with torque equal to 430 Nm @ 1600 rpm.

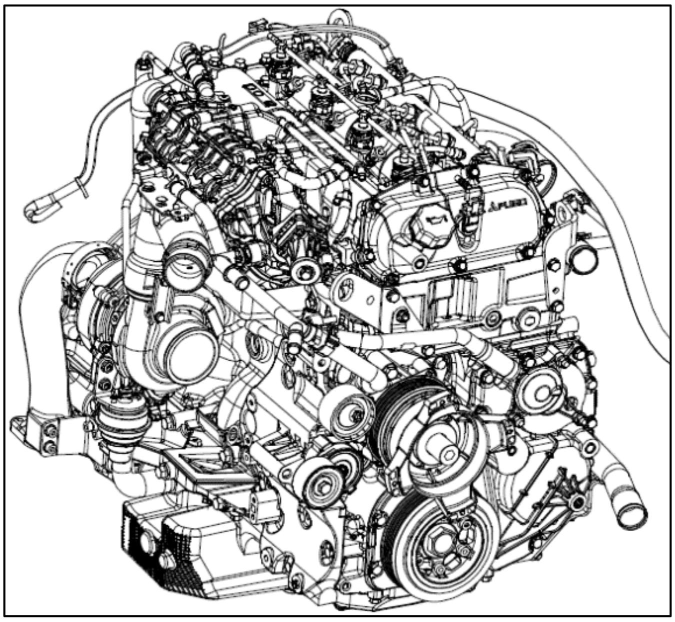

Figure 54 – Engine F1C

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[17] PoliTo= Politecnico di Torino

This engine is equipped with many sensors described in the following chapter plus other sensors not present when the engine is mounted on the vehicle but they are used only for the experimental project because they are so expensive.

For having a complete view of the sensor mounted and also for knowing the right position, below there is a complete scheme of our instrumentation.

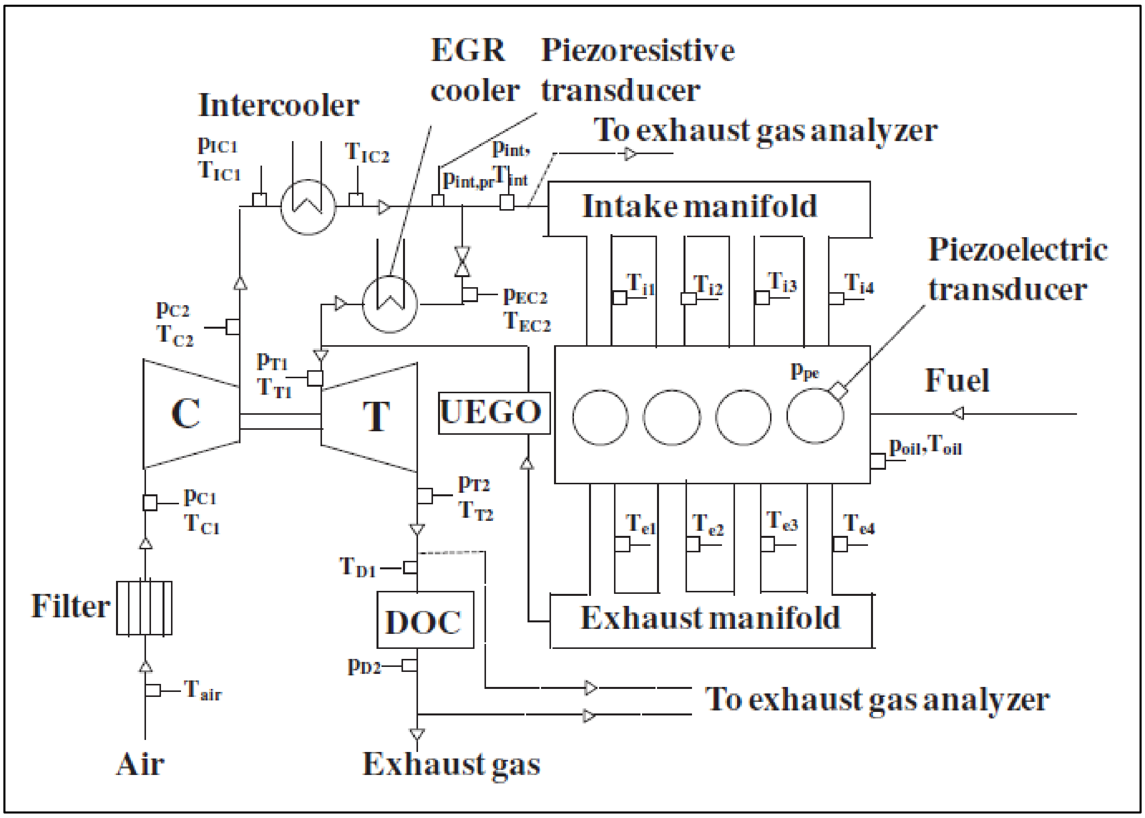

Figure 55 – Scheme of engine instrumentation "[23]"

5.1.1 Modification of the engine

For working better in the PCCI combustion mode, the engine is modified by FPT Industrial, with these following hardware:

- *Nozzle spray angle*: in this way there is the mitigation of the spray impingement due to the lower density of the air inside the chamber, when the injection happens;
- *Bowl shape*: the combustion chamber is modified to enhance the mixing of the fuel with air and residual in order to avoid rich pocket of fuel that generates soot;
- *VGT*: the TC system is replaced with a little one for reducing the turbolag and for having better performance @low/medium low, where the PCCI is used;
- *EGR cooler*: it is replaced by a bigger one, for increasing the heat dissipated from the EGR gas. In this way the intake air temperature can be lower and allows to use the PCCI combustion also at higher load;
- *Reduction of rc*: we exploit the reduction of compression ratio, so in this way, the peak temperature is reduced and we can also achieve greater specific power at given peak firing pressure, or better having the same specific power with lower PFP, that is positive for the noise production and for the durability of the engine.
- *Add throttle vale after compressor*: in this way we can decide to use it instead of the throttle valve at the turbine out for increasing the backpressure and so increasing the EGR rate, that is essential for having a PCCI combustion.

The other components are the same of the F1C EUVI engine.

5.1.2 Engine pressure sensors

5.1.2.1 Combustion chamber pressure sensors

The F1C cylinders are equipped with 4 Kistler 6058A pressure sensors, which sampling frequency allows them to measure the pressure traces of combustion cycles. They exploit a piezoelectric crystal and are characterized by high thermal stability and high sensitivity. Since the engine layout does not allow enough space to drill the engine head, it though to place them by displacing existing glow plugs. Technical data of in-cylinder pressure sensors are reported in table

High frequency oscillations implies the need of the proper tightening torque applied on the sensor. Since this sensor is piezo-resistive, it needs to an electronic amplifier. For this purpose, an AVL microIFEM amplifier is exploited by the test bench, which also allows a filtering of the acquired pressure signal from white noises and interferences.

Technical Data	
Measure ranges [bar]	*0 – 50; 0 – 100; 0 – 150; 0 - 250*
Overload [bar]	*300*
Shock resistance [g]	*2000*
Temperatures (min/max) [°C]	*-20/350; -50/400*
Tightening torque [Nm]	*1.2*

5.1.2.2 Intake manifold pressure sensor

A high frequency pressure sensor is also placed on intake manifold. It is a Kistler 4007C, characterized by an operation capability up to 200°C temperatures. Its work principle is based on the presence of a Wheatstone bridge, which generates an electric signal proportional to the detected pressure. Technical data for this sensor are summarized in Table

Technical Data	
Measure ranges [bar]	*0 – 250*
Overload [bar]	*15; 30; 40; 100; 200; 400*
Temperatures (min/max) [°C]	*-40/200*
Tightening torque [Nm]	*1.5 – 2.5*
Reference Temperature [°C]	*25*

5.1.2.3 Exhaust manifold pressure sensor

Also for what exhaust manifold concerns, a high frequency pressure sensor is provided. It is a piezo-resistive Kistler 4049B, equipped with an autonomous cooling system needed because of high temperatures due to exhaust gases the sensor is subjected on. The cooling of the sensor is managed by a PID control system. As a reference, Table 6 reports main technical data of this sensor.

Technical Data

Measure ranges [bar]	*0 – 5; 0 – 10*
Overload [bar]	*15 – 25*
Temperatures (min/max) [°C]	*0/120*
Tightening torque [Nm]	*20*
Reference Temperature [°C]	*60*

5.1.3 Engine temperature sensors

5.1.3.1 Thermocouples

The thermocouple is a temperature sensor which working principle is based on a temperature transducer. Here, a voltage difference takes place between two difference material conductors when a temperature gradient is present. They are versatile and low cost sensors because of its simplicity, while allowing to measure a wide range of temperatures (-200 ÷ 1350 °C). For the purpose of this project, K and T kind of thermocouples has been exploited in those components of the engine where fluids temperature should be monitored.

5.1.3.2 Thermistor

Thermistors working principle is based on the different resistance that they experience if different temperature conditions are present. The variation of the resistance implies a voltage drop across the thermistor that is related to the temperature itself. Once calibrated, for a certain voltage difference a certain value of temperature corresponds.

Actually, PT100 thermistors are installed in the engine with the purpose of monitor the cooling fluid temperatures of EGR, intercooler, engine itself and to monitor temperatures of test bench systems.

5.1.4 Engine additional sensors

5.1.4.1 Lambda sensor

The Lambda sensor is a device devoted to the measure of the oxygen quantity of exhaust gases, in order to estimate the right quantity of fuel to inject inside the combustion chamber.

In our case we use the *ETAS S636 lambda module* that allows us a good evaluation of the lambda value in the exhaust gases.

This is the link between the PC, the ETAS module and the lambda sensor, how you can see, the lambda value acquired by ETAS module can be spread to the PC and also to the ECU, that in this type of tests is an open ECU for developing.

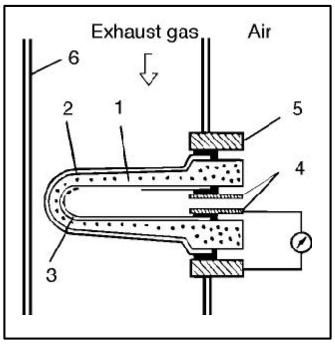

Figure 56 – Lambda sensors " "D'Ambrosio's notes"

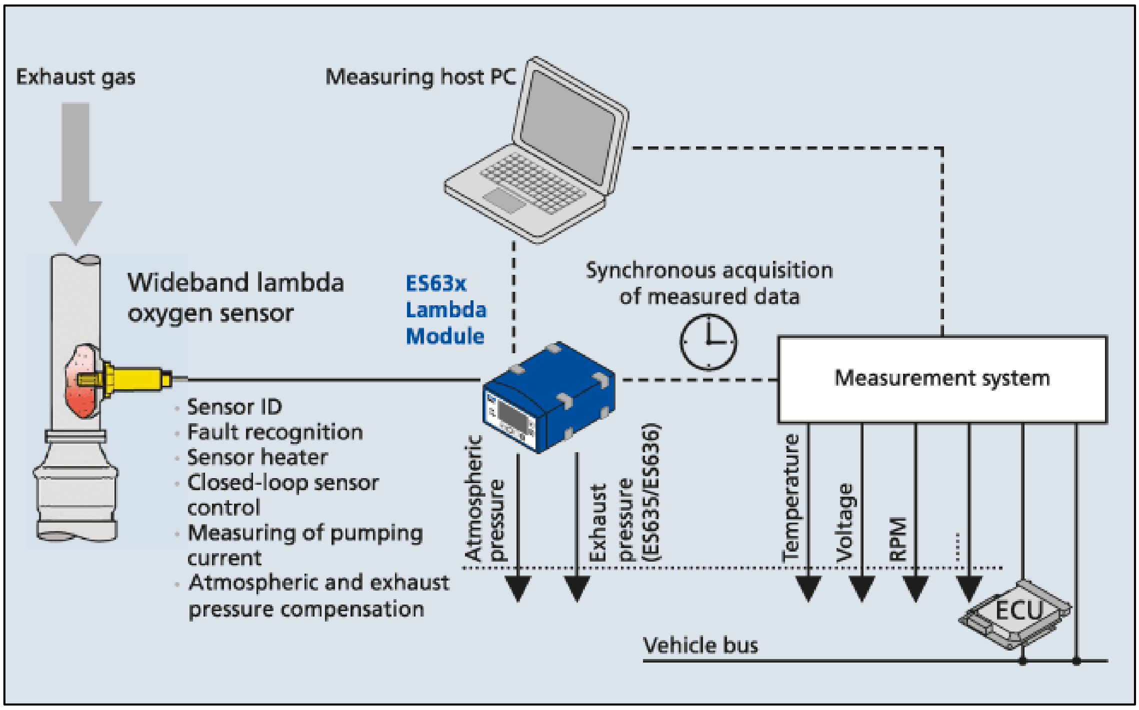

Figure 57 – Scheme of the interconnection b/w the measuring host PC, ECU and lambda sensor "www.AVL.com"

5.1.4.2 Blow-By gases flow measuring system

The blow-by refers to those quantities of gases that during the expansion phase escape from the combustion chamber, by crossing piston rings and reaching the crankcase. This phenomenon causes a loss in engine output power since the piston is not able to collect all the provided work coming from the combustion pressure increase. Furthermore, this implies oil dilution and pollutant emission increase.

In order to take under control this phenomenon, the engine is provided of a circuit that extracts gases from the crankcase and introduces them upstream the compressor. The test bench is equipped with a sensor – the AVL blow-by meter – devoted to the blow-by gas flow-rate measurement.

5.1.4.3 Turbine rotational speed sensor

A *Pico-Turn* rotational speed sensor is mounted on the engine turbocharger. It allows to measure the turbine rotational speed, which is related to the boost variation on the engine. The monitoring of it becomes important when the engine reaches maximum speed conditions, near to the tolerability limits of turbine shaft bearings. For this purpose, a devoted alarm on AVL Puma is set to a maximum allowable speed.

For the sake of completeness, the engine scheme with sensors location and list is attached on Outlooks section of this project.

5.2 The test bench

The test bench used for the tests in engine department is an AVL.

All the parameters coming from different sensors are acquired through the *F_FEM “Firewire Front End Module”*.

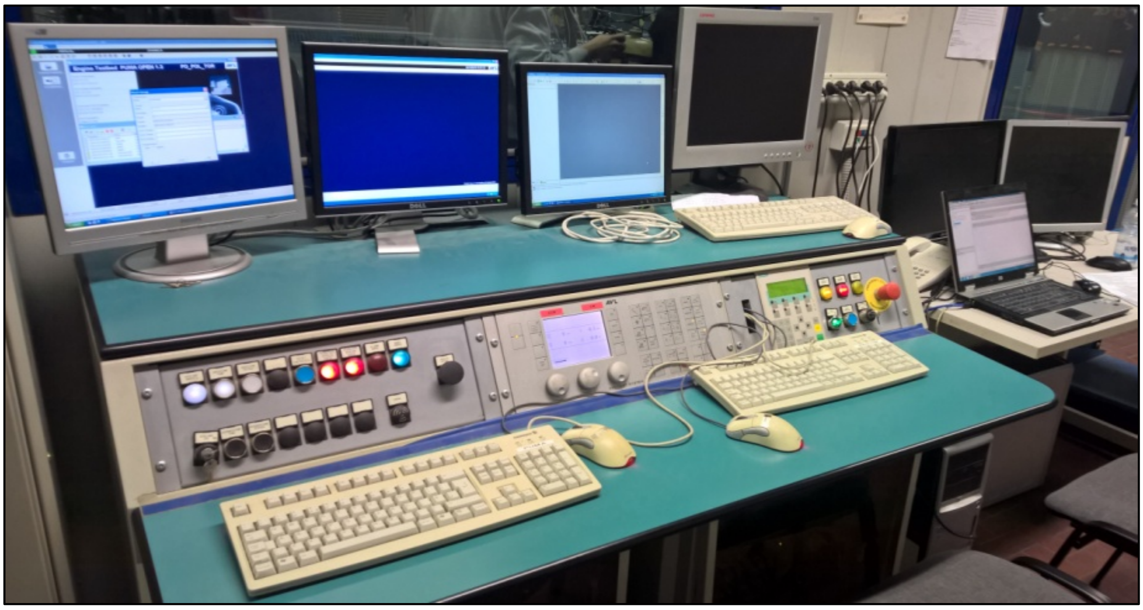

Figure 58 – Engine test bed

5.2.1 Software

All the activities are done by using 3 different software:

- AVL Puma;
- AVL Indicom;
- ETAS INCA.

So now I’m going to discuss more in details the functions of each software

5.2.1.1 AVL Puma open 1.3.2.

This software is used to control the engine test bench, so it is the master of our system at which are linked all the devices for pollutant emissions measurements and receive all the electrical sensors from the sensors.

These electrical signals come to Puma through the F_FEM.
To Puma comes also the variables from the ECU through a connection ASAM with ETAS INCA.

It’s possible to set the load variable in different combinations:

- Rpm x Torque;
- Rpm x Throttle;
- Rpm x Power;
- Throttle x Torque.

It’s also possible so have different approach to reach the desired working point like ramp, step and so on.

It manages also all the conversion from the electrical signal to a measurable variable that is shown on the monitor.

At the end, is possible to acquire all these variable that Puma reads and store in a txt file that will be processed during the Post-processing phase, with matlab and excel.

5.2.1.2 AVL Indicom

We use this software for threating the high frequency pressure signal coming from each combustion chamber.

With this signal is possible to evaluate in real time the noise, the HRR and the MFB50 for each signal.

The last one value is very important for seeing the centre of combustion and how it changes changing the engine parameters like the SOI.

5.2.1.3 ETAS INCA

The control of engine parameters is performed through a control of the ECU by means of the software ETAS Inca, which allows the engine variables and maps control.
For this purpose, FPT Industrial has unlocked some ECU variables, in order to investigate effects of input parameters on the combustion or emissions.

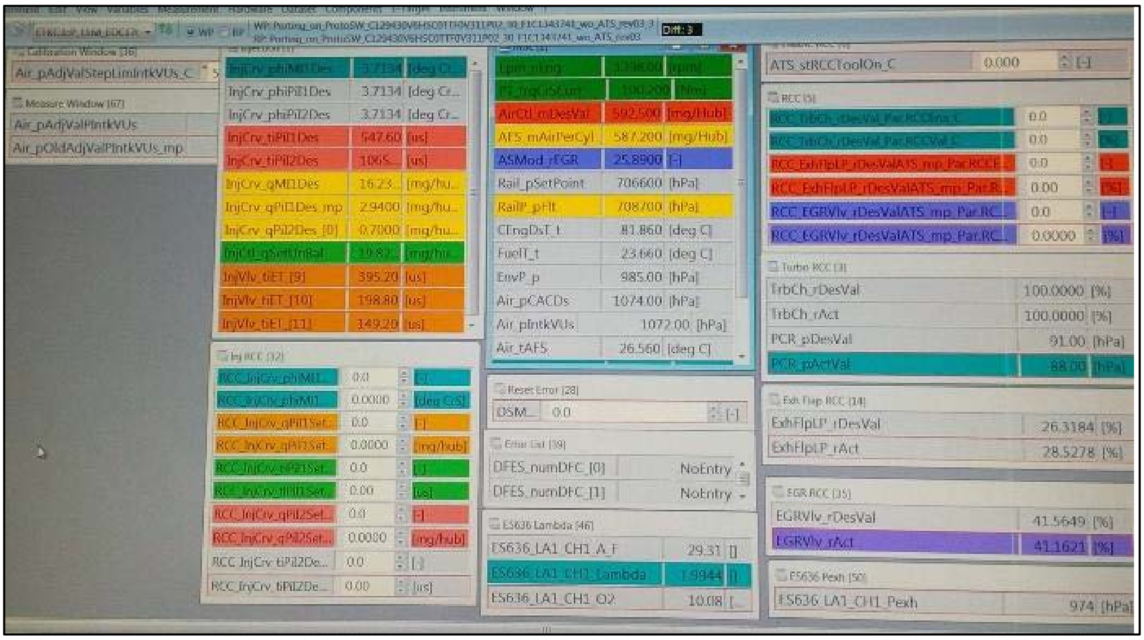

Figure 59 – Experiment work page

By means of Inca, it is possible to download on an ETK module the desired calibration. In our case we use the ETK S910 that allows us, not only to put in the right calibration but also to insert a model working in real time.

5.2.2 *Test bench equipment*

In this chapter I'm focusing on the equipment used in the engine test bench for analysing the pollutant emissions and for running the engine.

5.2.2.1 *Dyno system*

The test bench is equipped with an AVL APA 100 dynamometer. It corresponds to 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, which braking effect is took out from induced eddy currents.

It can be operated also as a motor: this operation could allow investigating on engine organic losses. This kind of dynamometer presents two output shaft which are connected: this permits to test another engine, also if no contemporary, without dismount the equipment of the other one.
When a Team need to test one engine, the joint connection of the other must be dismounted.

Some technical feature of the dynamometer used for this project are reported on

Technical Data

Maximum power [kW]	*200*
Maximum torque [Nm]	*525*
Maximum rotational speed [rpm]	*12000*
Inertia moment [Kgm^2]	*0.32*

5.2.2.2 Emission analyser

Since the target of this project is the optimization of some engine working points in terms of pollutant emissions reduction, a proper emission analyser system must be provided. The emission control type is strictly related to the state of the collected quantity: in this case, emissions are raw, i.e. they are not diluted as well as homologation procedure concerns.

Collected quantities must not react chemically in order to avoid their change in composition. Solid pollutant particles must be eliminated from stream lines by means of devoted filters.

Furthermore, steam water in stream lines must be avoided. For this reason, some procedures of purging are needed, accompanied by frequent calibration procedures which exploit gas quantities of a precise and known chemical composition.

The emission analyser apparatus used for this purpose is the AVL AMA i60 (Figure 60) with three stream acquisition lines.

This system has 2 lines, the line 1 has the pick-up point after the DOC and the line 2 has the pick-up point before the DOC.

These 2 lines are able to perform the analyzation of all the pollutants and for having a precise measurement we use sample gas for calibrating each analyser.

In addition, it is present a 3rd line used only to measure the CO_2 presents in the intake, in this way we can evaluate the percentage of EGR.

5.2.2.3 Fuel consumption measuring system

We use a device for a continuing measuring of the fuel injected that is AVL KMA4000 (Figure 61).

This allows us to measuring the BSFC in each working point, and the signal is sent to Puma that collects all the values.

These are the technical data of the device:

Technical Data

Flow rate measure range [kg/h]	*0.02 – 380*
Density measure range [g/cm³]	*0.5 – 2.0*
Volume flow rate error	*± 0.1%*
Density measure error	*± 0.03%*
Mass flow rate mearure error	*± 0.1%*

Figure 61 – AVL pollutant emissions analyzer AMA i60

Figure 60 – AVL fuel measurement system KMA 4000

5.2.2.4 *Smoke meter*

The smoke meter is used to measure the soot emission, and as we already know, the soot is defined by the sampling method.

The measure is done in FSN "Filter smoke number" that indicate how much dark is the filter exposed by the exhausted gas.

This device can be used only in steady-state mode, for having a continuous acquisition of the soot emissions we need another instrument called opacimeter that works with a laser and measure how much light is absorbed by the gas.

Figure 62 – AVL Smoke meter

5.2.2.5 *Engine cooling fluids system*

In the engine test bench is present the cooling circuit that replace the real one installed on the engine.
This external circuit exchanging heat with the engine cooling circuit by a WCAC[18].

In this system is installed also a heater, for fastening the heating of the engine at cold start and also a throttle valve to reduce the flow rate inside the circuit.
These 2 devices are controlled by PIDs present on Puma.

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[18] WCAC = "Water Charge Air Cooler"

On the engine side, it’s present an addition pump to create more load on the circuit to restore the right value of coolant flow rate.

Figure 63 – Engine cooling fluid system

The external circuit has also the aim to cool-down the EGR gas because we replace the older EGR cooler with one bigger and it is alimented by the external circuit, different from the first one that was alimented by the engine coolant.

6 How Hot EGR affects the engine performances

This is the main chapter of my thesis, I'm going to discuss a research done in my activity.

This involves the evaluation of performance changes using a hot EGR instead of cold EGR in working points shown in:

1000x1.13	*1400x1.13*	*2000x1.13*	*2500x1.13*	*3000x1.13*
-	-	*2000x2.27*	*2500x2.27*	*3000x2.26*
1000x2.97	*1400x2.97*	*2000x2.97*	*2500x2.97*	*3000x2.97*
1000x3.93	-	-	-	-

Table 4 – Hot EGR working points

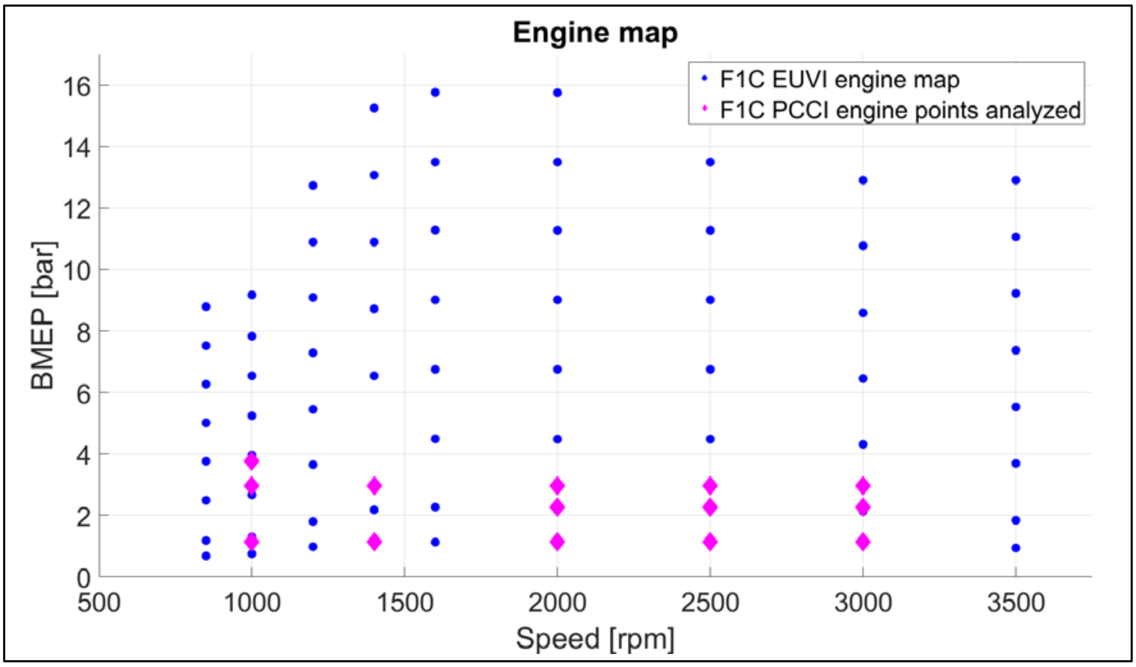


Chart 36 – Engine map

All the results will express as a percentage difference for not showing the numerical value of our tests, as the following expression.

$$\frac{Value_{indicated} - Value_{ref}}{Value_{ref}} * 100$$

So in the graph will be always a point in which the value will be zero.

6.1 Cold EGR layout

The calibration of previously points is found by experimental procedure, changing one engine parameter per time, searching to mitigate the drawbacks of PCCI combustion and exploit also all the benefits as the NO_x and PM emissions.

The graphs of trade-off between NO_x and soot, in this type of combustion is useless, so it's better to have a comparison between the engine pollutants emissions with BSFC and noise.

For these points, the layout has a bigger EGR cooler.
The heat is rejected by a coolant that is linked to the engine test bed.

6.1.1 Problems related to this configuration

Due to this configuration, with the replaced EGR cooler we have these following problems that I would to explain, and focus on the solution used to avoid them:

- Clogging of EGR cooler;
- Too low temperature at engine war up;
- Additional heat load on the cooling systems.

6.1.1.1 Clogging of EGR cooler

This problem is due by the condensation of the HC inside the EGR cooler.
The out temperature of the EGR cooler is set at 85 °C and it is controlled by a PID controller.
This temperature brings the condensation of HC that clogging the EGR cooler, so after a few hours of working we have to dismount it and clean to have a reliability results in the tests.

This procedure can't be done in the normal life of the vehicle because the driver would to use the car without make many repairs during its life, so we have to find a solution for mitigating or eliminating the problem.

These are the solution that can be adopted:

- Use a DOC before the EGR cooler;
- Active system cleaning the EGR cooler;
- Increase the output temperature of EGR cooler;

6.1.1.1.1 DOC at EGR cooler inlet

Installing the DOC at EGR cooler inlet can be a good solution for oxidation of the HC that will be condensate inside the cooler.

We have schedule these tests with the DOC and we will try different DOC dimensioning but this activity will be done in the following month in which I will not be present in the laboratory.

There are 2 problems relating to this solutions:

- Increase the inlet temperature of EGR gas that increase the heat load at the cooling system.
  In our case isn't a problem because we use an external circuit for the EGR cooler, but in real application we have to consider that;
- Not reach of light-off temperature can be another problem that make this solution useless.
  This has to be evaluated by tests on the engine test bench.

6.1.1.1.2 Active system cleaning the EGR cooler "SECFR"

A steam EGR cooler fouling remover (SECFR) system was designed using evaporated washer fluid to remove fouling on the cooler surface.

However, the injection pressure is only 5 bar, which is not sufficient to atomize the washer fluid.
Therefore, an impinging plate was used to promote the atomization of sprayed washer fluid, and the droplet area ratio (DAR) and droplet uniformity index (DUI) were measured through quantitative analysis of high-speed images.

To remove fouling on the EGR cooler, the washer fluid must be vaporized as heated steam. To ensure vaporization, it is necessary to select engine operating conditions to achieve a high temperature up to 200 °C and a relatively large amount of EGR gas.

As washer fluid is injected at the inlet of the EGR cooler, the outlet temperature of the EGR gas decreases dramatically due to the evaporative latent heat of the washer fluid. To compensate for this, a temperature sensor is installed to automatically shut off the injector when the outlet temperature of the EGR cooler falls below 100 °C.

So this system is able to removes the fouling on EGR cooler and also decrease the temperature inside the EGR cooler that is used for evaporating process, so increase the volumetric efficiency.

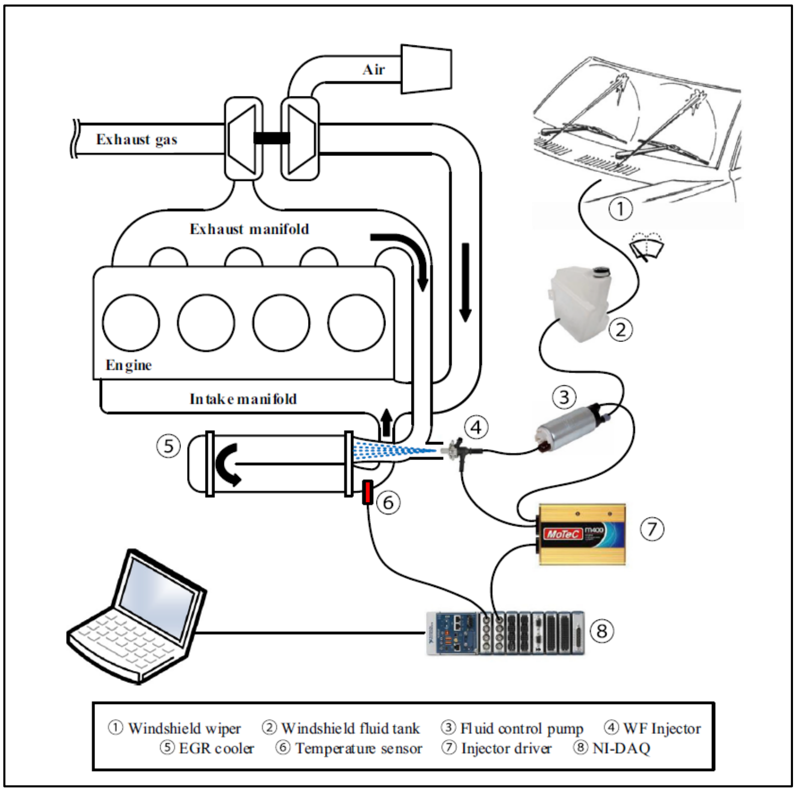


Figure 64 – SECRF system layout "[25]"

6.1.1.1.3 Increase the output temperature of EGR cooler

This is the simplest solution to be actuated because we simply change the target temperature of the PID controller.

Changing the output temperature, high enough to avoid the condensation of HC, is a good solution in terms of EGR cooler clogging but can be detrimental for the combustion.

In fact, if we increase the output temperature of the cooler, this means increase the temperature at the beginning of combustion, that leads a more active mixture. This can't allow us to exploit the PCCI combustion at very high load, where the injected fuel is higher.

So, we need to accept a reduction of the maximum load reaching with the PCCI combustion mode.

6.2 Hot EGR layout

To simulate the by-pass of EGR cooler, we replace it with a tube that doesn't exchange heat with the coolant, so the gas recirculated inside the chamber is hot and the temperature depends on the load and engine speed.

6.2.1 Choice of the best calibration for each working point

For finding the best point in each conditions, I comparing all results in terms of BSFC, NO_x and soot emissions and finding the minimum of BSFC without huge penalty in the other 2 terms.

I give you an example without insert the real values about the BSFC, NO_x and soot for copyright reasons, but they are related to the minimum.

This is evaluated for the point 2000x2.27.

I choose the test number 27, where the BSFC is very close to the minimum and the penalty in the NO_x and soot is little enough to stay below the limit, as is showed in the 3 following graphs.

For the NO_x graph I choice to set this scale for looking the real behaviour of the behaviour of the NO_x concentration. In fact, it's true that point chosen is around 70% higher than the minim but the concentration in ppm is very low in both case.

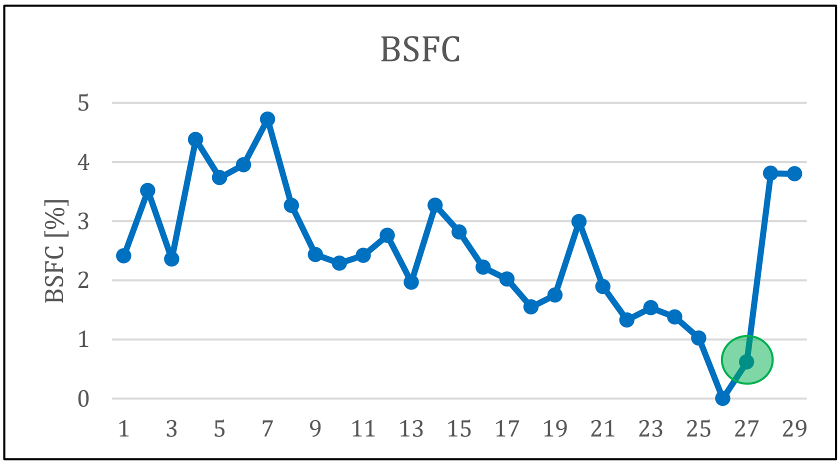


Chart 39 – Best calibration point choice "BSFC"

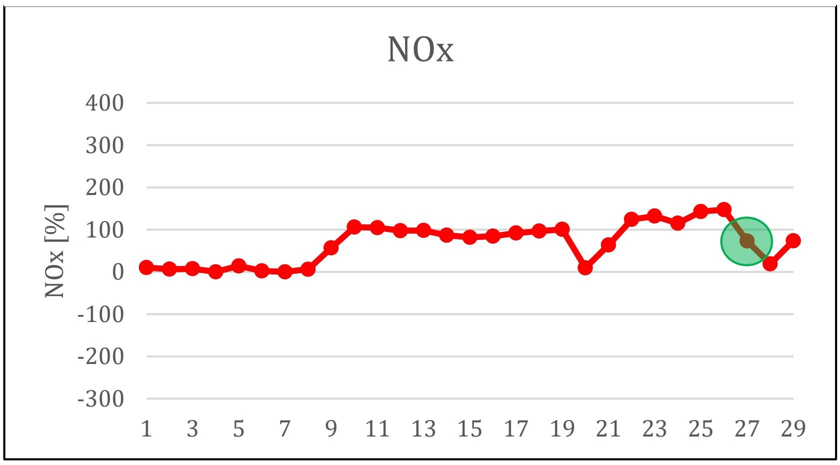


Chart 37 – Best calibration point choice "NOx"

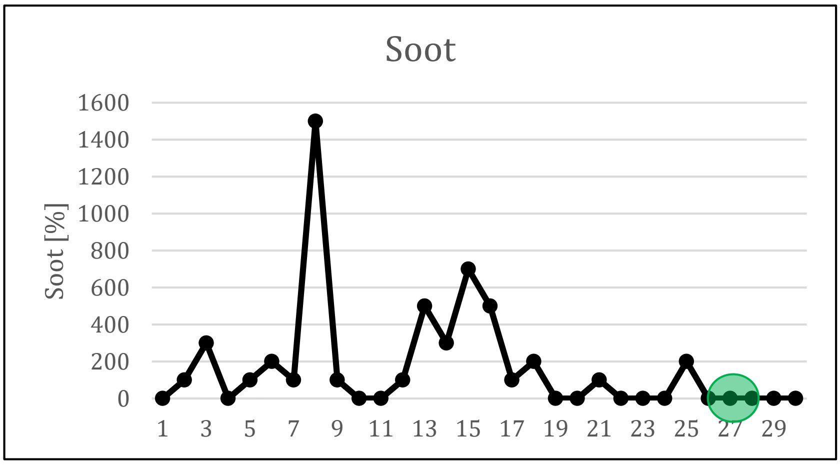


Chart 38 – Best calibration point choice "Soot"

6.2.2 Advantages of this layout

This layout can increase the temperature at the engine inlet because now there isn't the EGR cooler.

In addition, the temperature varies with engine load and speed because there is the control on the temperature of residual gas before to enter in the intake manifold.

This can increase the catalyst and the engine warm-up but increasing the load creates instability in the combustion.
After, in the results chapter I will show the limit due to the inlet temperature.

The high temperature creates a higher activity of the mixture so this solution has a limit for the PCCI combustion above certain load.

If we want to minimized the fouling effect inside the EGR cooler, we can by-pass always the cooler when the speed of the exhaust gas is below certain level, because we know increasing the exhaust gas velocity we can decrease the amount of deposits inside the cooler and also, if the velocity is high enough, we can exploit it to clean the cooler.

6.3 Results output

I'm show all the values and the difference between them of temperature, pollutant emissions and BSFC.

The scatter's colour indicates the following properties:

Colours	***Meaning***
Cyan	*Cold EGR*
Dark orange	*Hot EGR*
Green	*Advantage*
Red	*Disadvantage*

Table 5 – Scatter's colour properties

Also in this case, I refer all the values about the pollutant emissions and BSFC at the minimum evaluated, for not showing the real number.

The parameters evaluated are:

- Inlet temperature;
- Exhaust temperature runner 1;
- Exhaust temperature aTC;
- CO bATS and aATS;
- HC bATS and aATS;
- NO_x bATS and aATS;
- Noise;
- BSFC.

6.3.1 Inlet temperature

In the case of “Cold EGR” this is quite the same, independently from the engine speed and load because there is the PID that control the exhaust temperature after the EGR cooler.

Instead on the “Hot EGR” the temperature varies with the load and the speed and it is always higher than the “Cold EGR”.

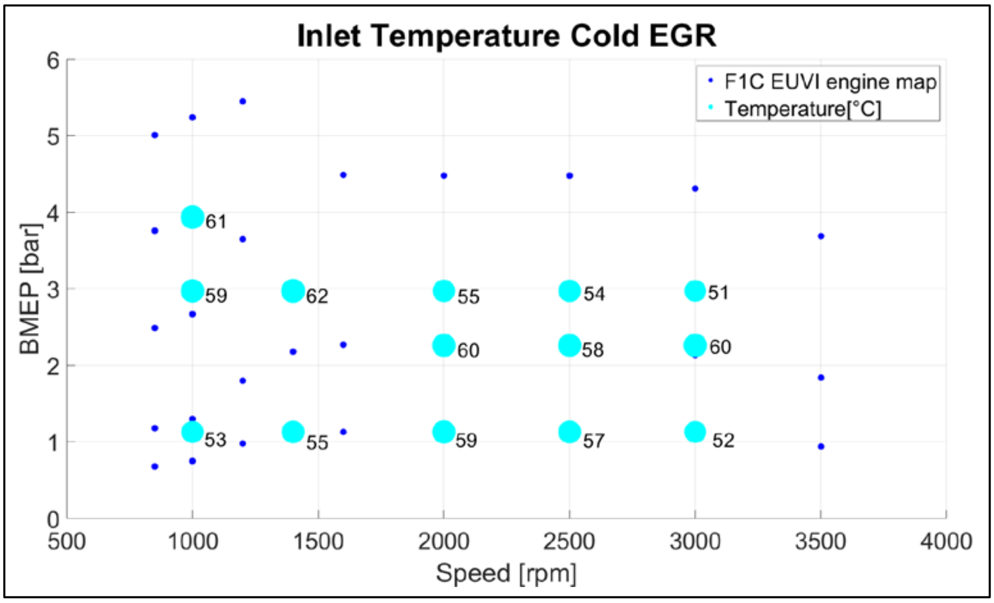


Chart 40 – Inlet T Cold EGR

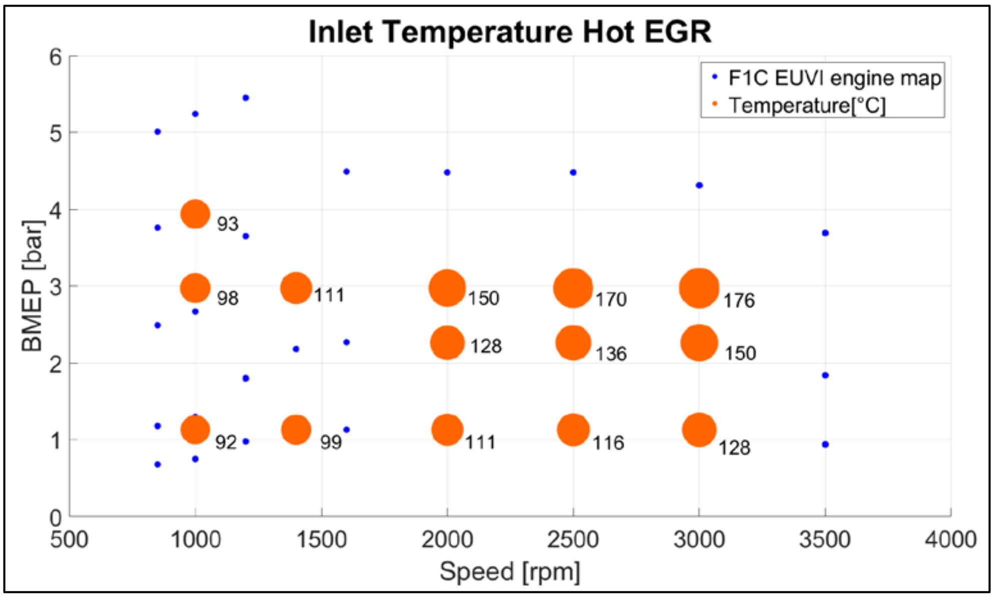


Chart 41 – Inlet T Hot EGR

6.3.2 Exhaust temperature bDOC

This temperature is very important because can gives an indication if the DOC is in the working window temperature and so our engine can be homologated for the EUVI legislation, otherwise the emissions of HC and CO can be very high.

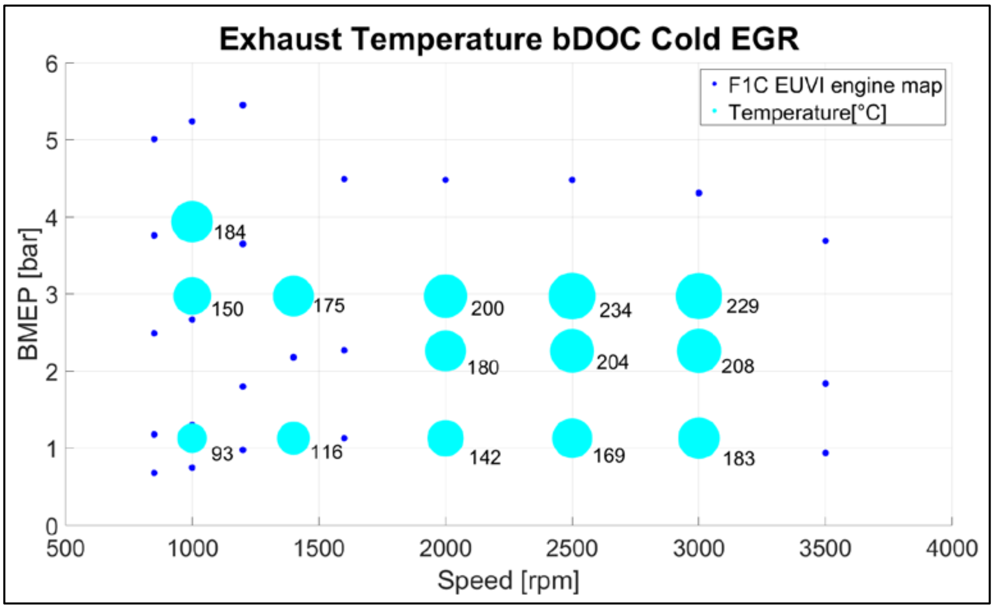


Chart 42 – Exhaust T bDOC Cold EGR

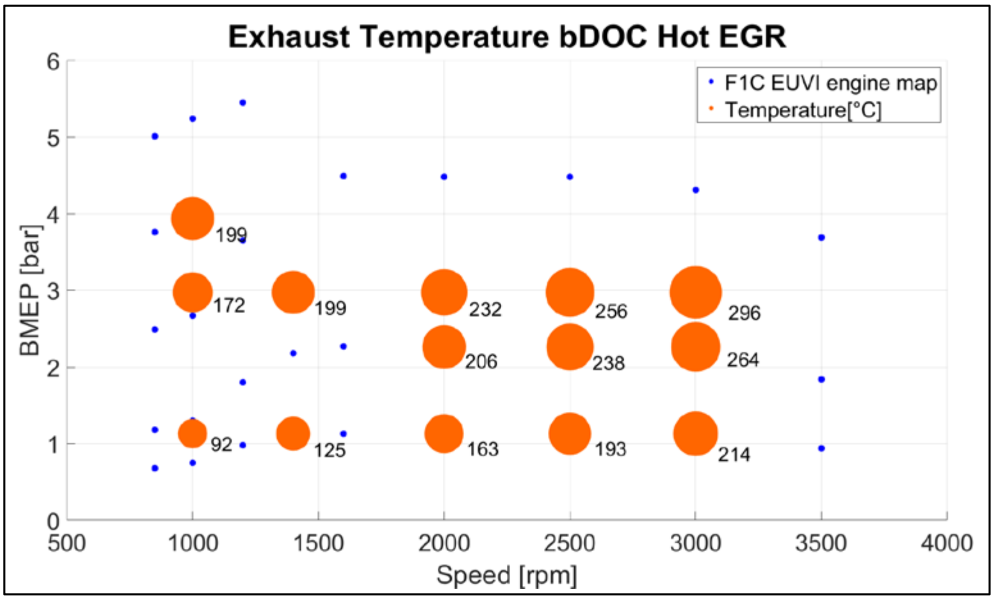


Chart 43 – Exhaust T bDOC Hot EGR

6.3.3 Exhaust Temperature difference

In the first graph () is possible to show the increment in temperature at the exhaust manifold that we have if we use the Hot EGR gas.

This can be different before the DOC because we have to use different calibration set, so the VGT and the throttle valve after the turbine can create different temperature drops. However, it is quite always good except for the lowest point.

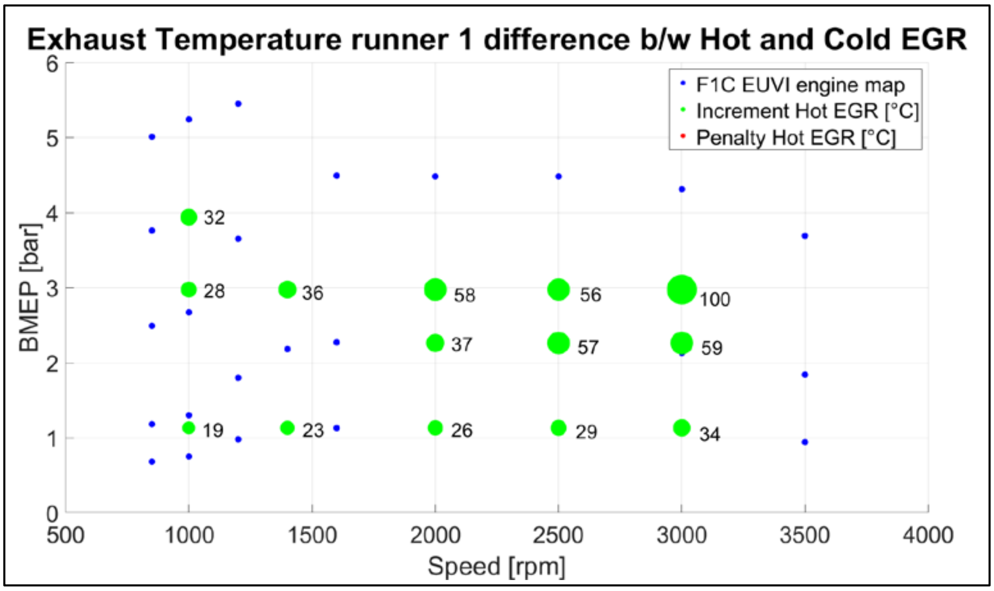


Chart 44 – Exhaust T runner 1 difference b/w Hot and Cold EGR

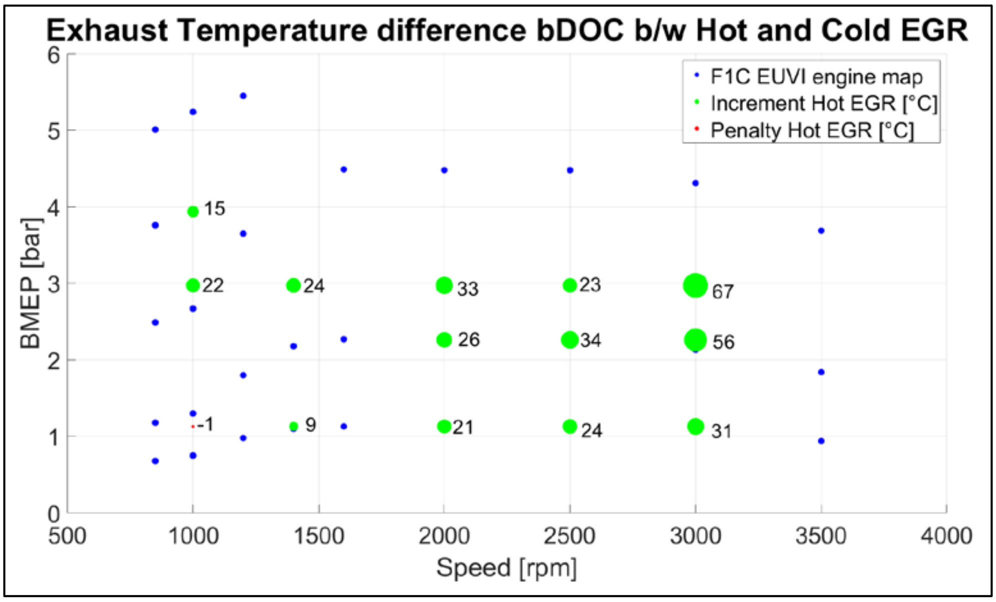


Chart 45 – Exhaust T difference bDOC b/w Hot and Cold EGR

6.3.4 Lambda “λ”

Due to the higher temperature in the intake manifold, lower lambda can be reached in the hot EGR case, without acting too much in either the exhaust or the intake throttling flap. This is due because the volumetric efficiency is decreased with the increase of the air intake temperature.

Also in this case the values of lambda are expressed in % with the same base between hot and cold EGR for having a good comparison.

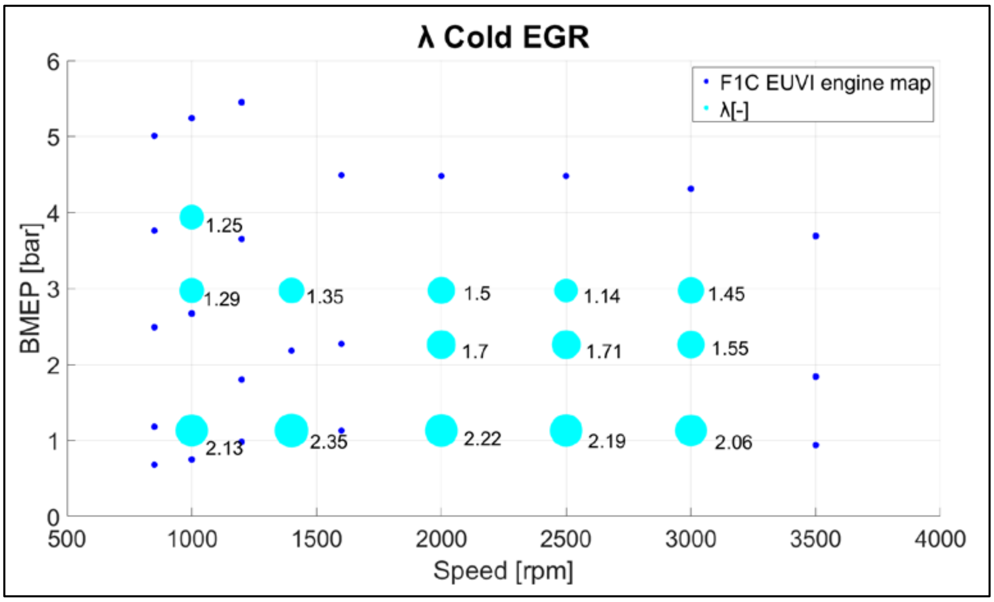


Chart 47 – λ Cold EGR

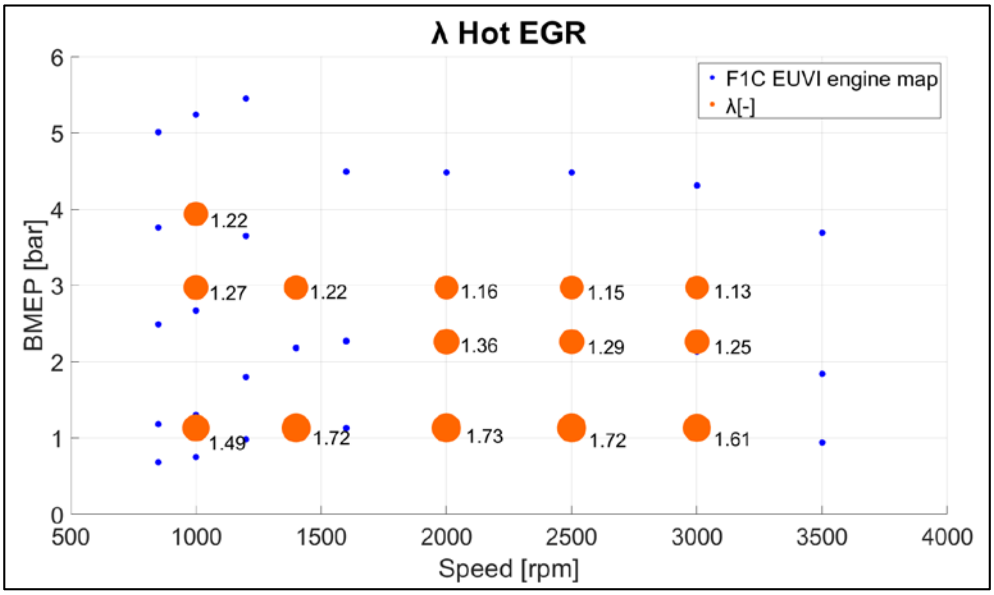


Chart 46 – λ Hot EGR

6.3.5 NO_x emissions

NOx values are almost everywhere low, with only slight increase at the lowest speed and load points in the case of hot EGR.

This is because in all the point we have a PCCI combustion stable and so we can exploit all the advantages of this combustion mode.

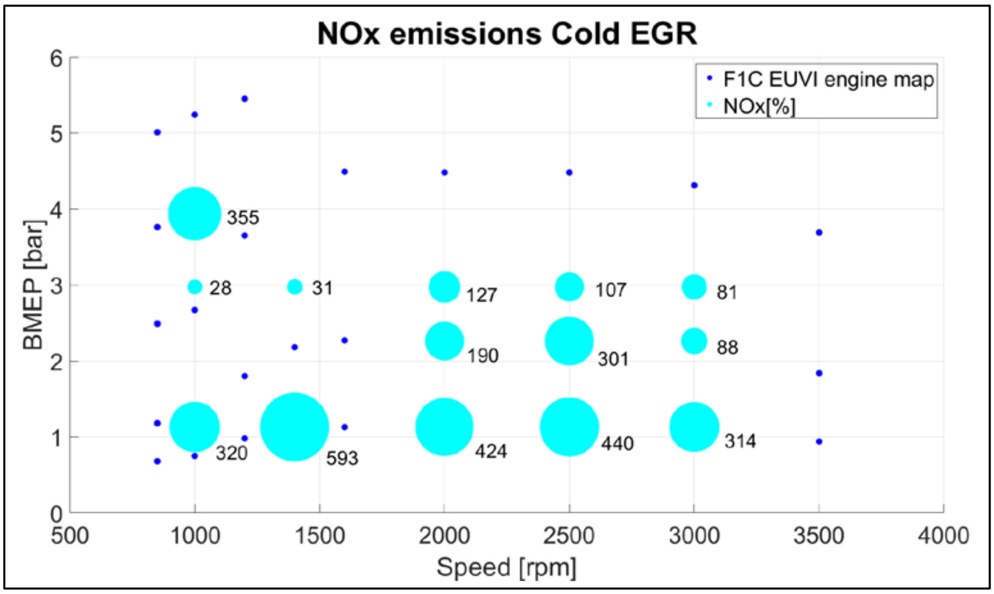


Chart 48 – NOx emissions Cold EGR

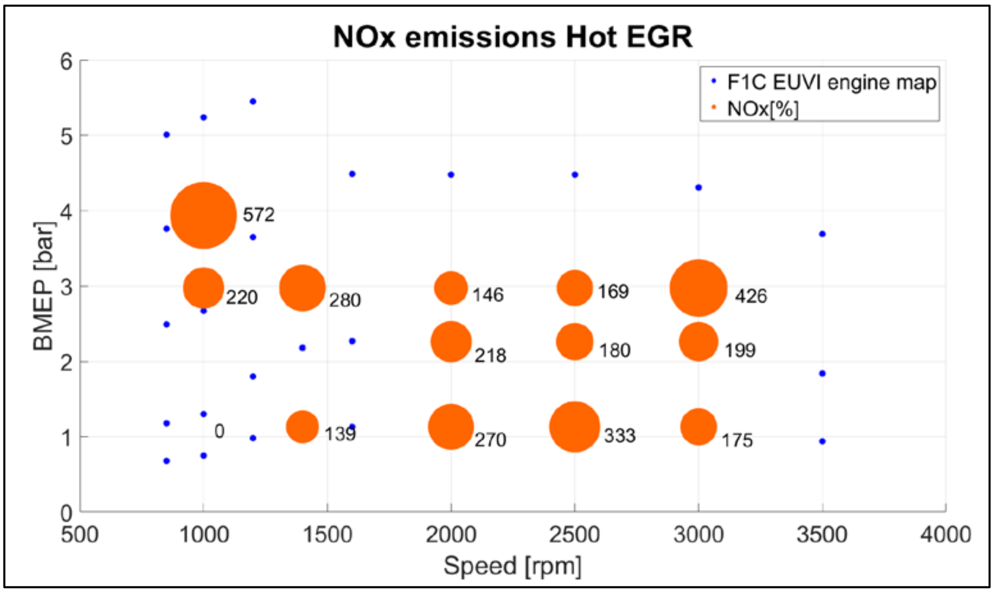


Chart 49 – NOx emissions Hot EGR

6.3.6 Soot emission

At higher loads than 3 bar BMEP, smoke values tend to increase more rapidly in the case of hot EGR, so the point at 1000x3.76 is the limit of the hot EGR with PCCI combustion, because after this limit the combustion became instable and exit from the PCCI mode due to the higher reactivity of the mixture, that doesn't allow to have the ignition delay longer than the injection time.

All the other points are in the minimum, so for the meaning of percentage difference this became zero.

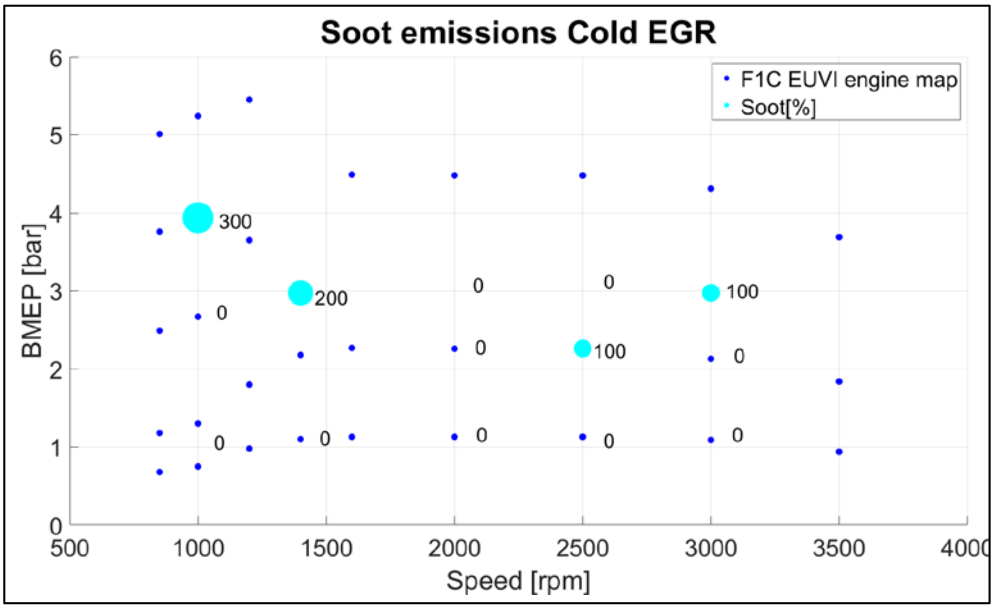


Chart 50 – Soot emissions Cold EGR

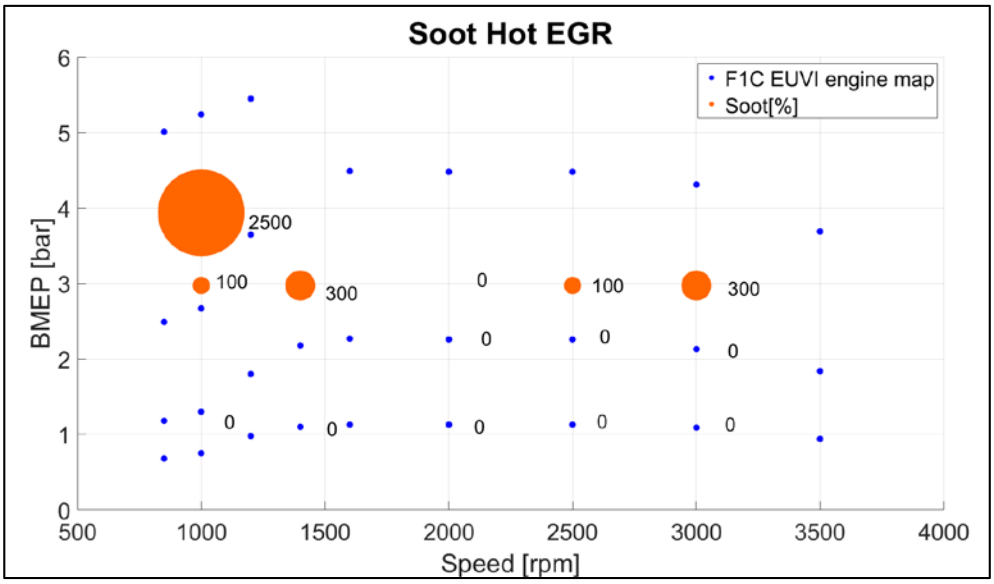


Chart 51 – Soot emissions Hot EGR

6.3.7 HC emission bATS

HC emissions after engine are generally higher in the hot EGR layout, likely due to the lower lambda values. This can be dramatic if the DOC still does not work.

Fortunately, we exploit the higher exhaust temperature and we reach the light-off temperature of DOC so, at the end, the pollutant emissions will be lower.

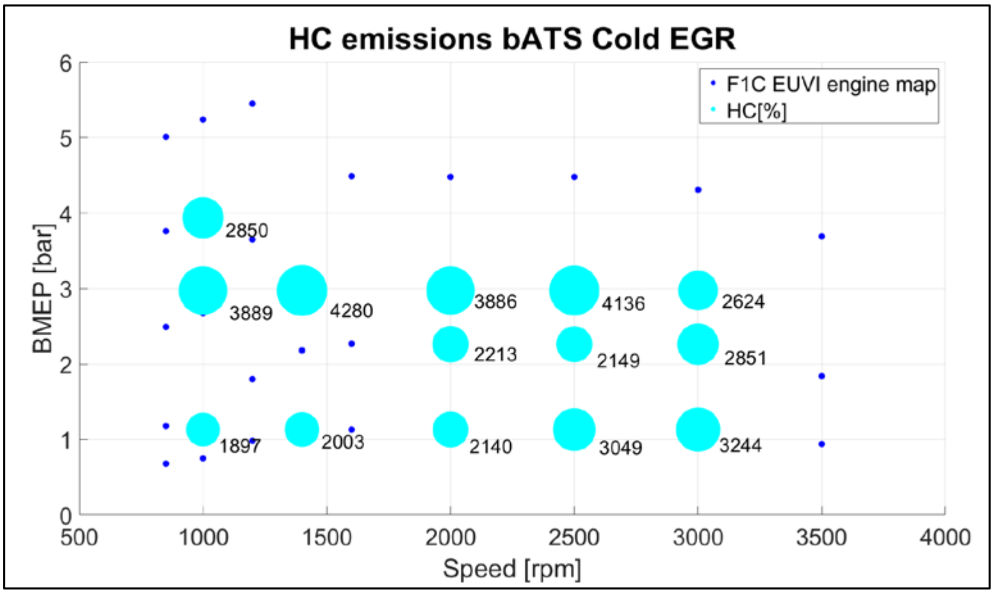


Chart 52 – HC emissions bATS Cold EGR

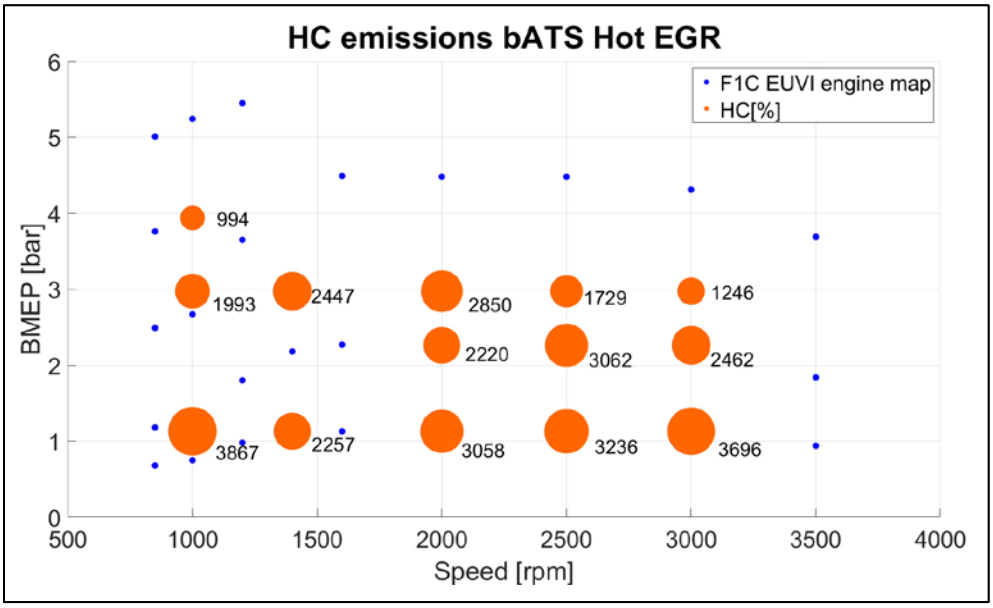


Chart 53 – HC emissions bATS Hot EGR

6.3.8 HC emissions aATS

Even if the HC emission engine out, as the exhaust temperature are higher, hot EGR gives the possibility to make DOC reach its light-off temperature in engine points in which this that was not possible with cold EGR.

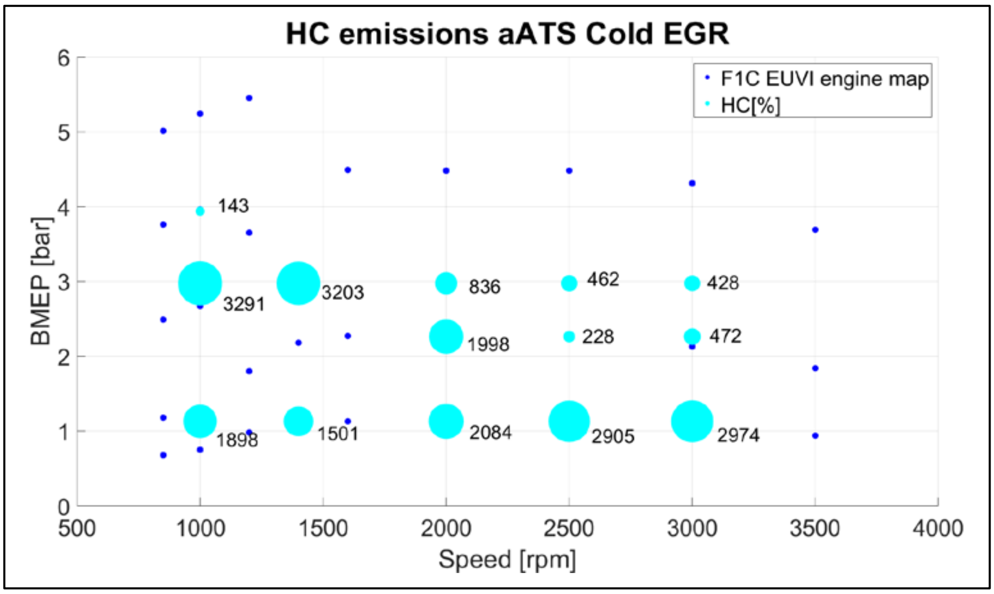


Chart 54 – HC emissions aATS Cold EGR

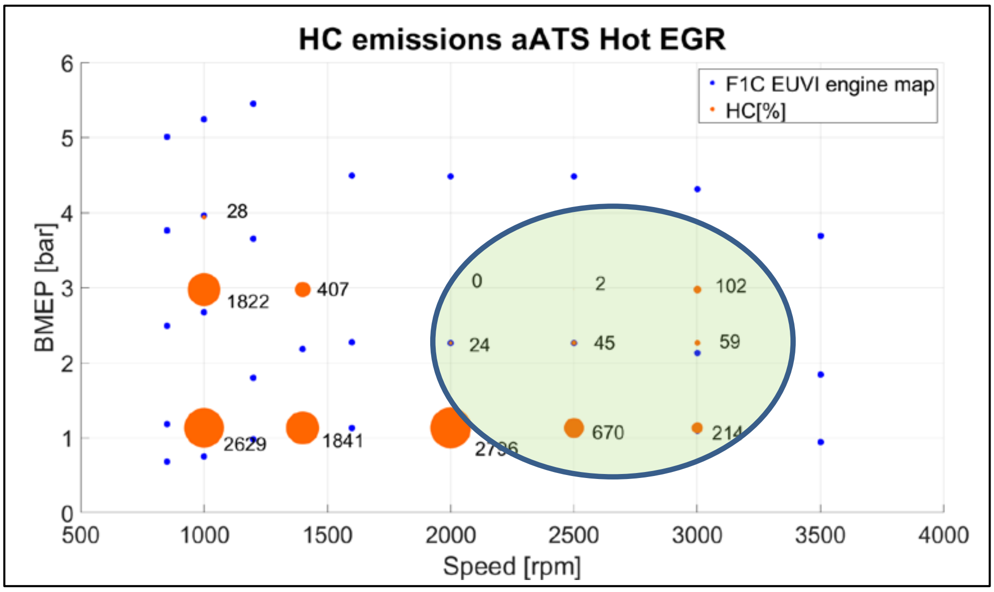


Chart 55 – HC emissions aATS Hot EGR

6.3.9 CO emissions bATS

As for HC emissions, CO after engine tends to be higher for hot EGR.

In this case, for having better graphic visualization I used different adimensional formula:

$$Value_{adimensional} = \frac{Value_{indicated} - Value_{ref}}{Value_{ref}}$$

In this way, the representation shows the real behavior of CO emissions

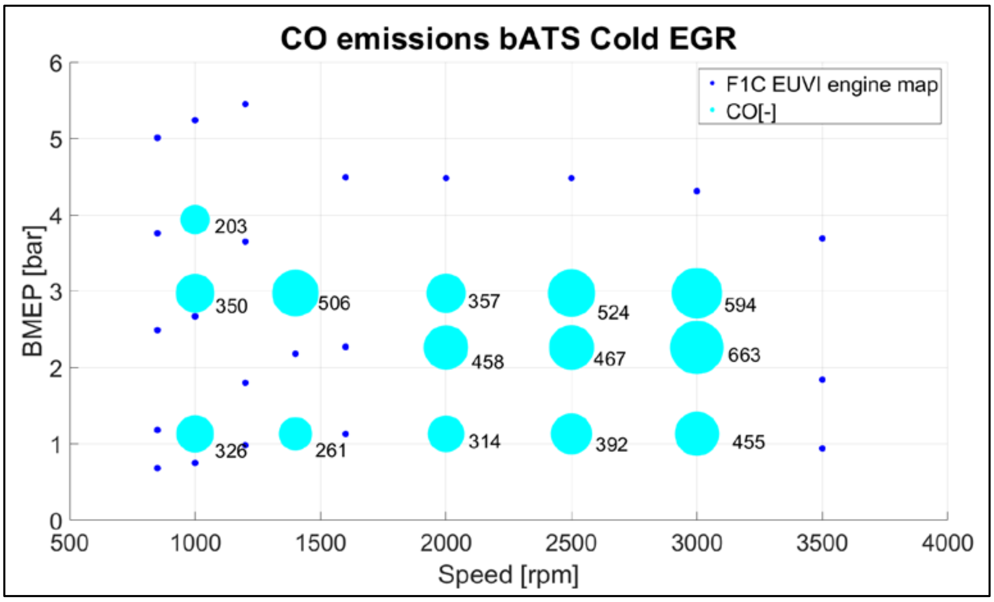


Chart 56 – CO emissions bATS Cold EGR

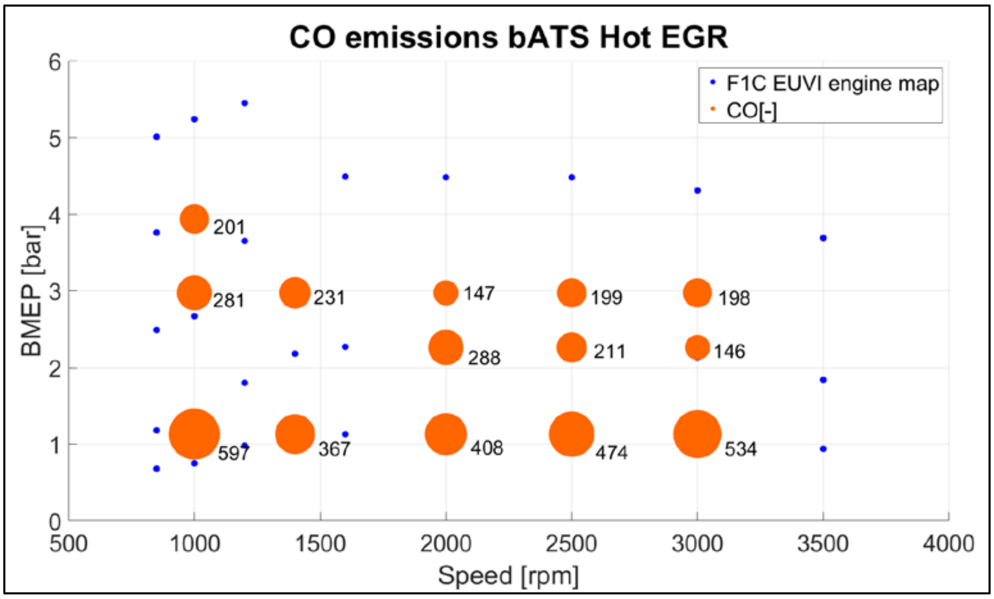


Chart 57 – CO emissions bATS Hot EGR

6.3.10 CO emissions aATS

In this case, too, as the exhaust temperature are higher, hot EGR gives the possibility to make DOC reach its light-off temperatures in engine points in which this was not possible with cold EGR.

In addition, in the point in which, also with cold EGR, we reach the light-off temperature, with the Hot EGR we have a lower light-off time.

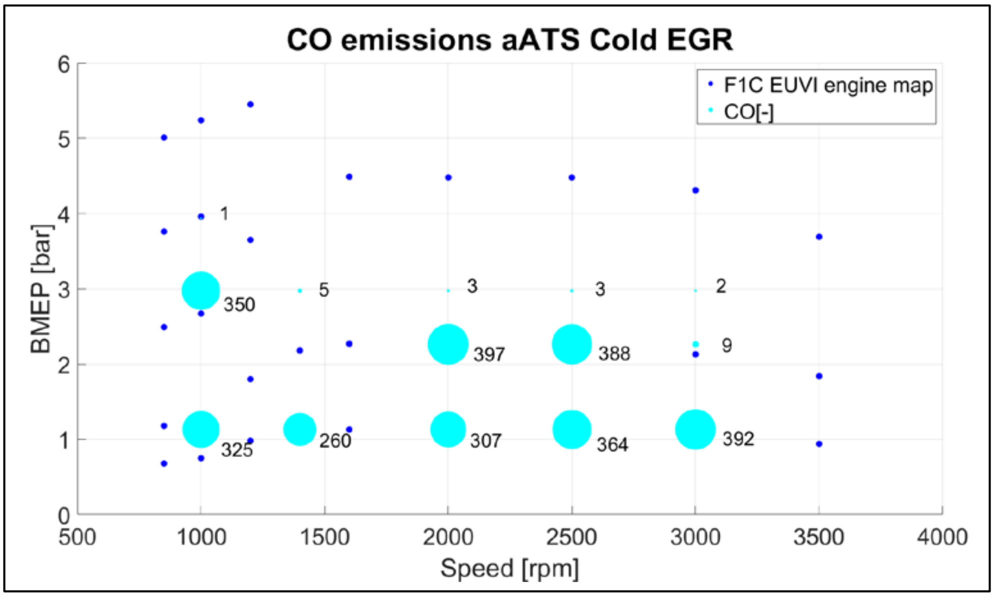


Chart 58 – CO emissions aATS Cold EGR

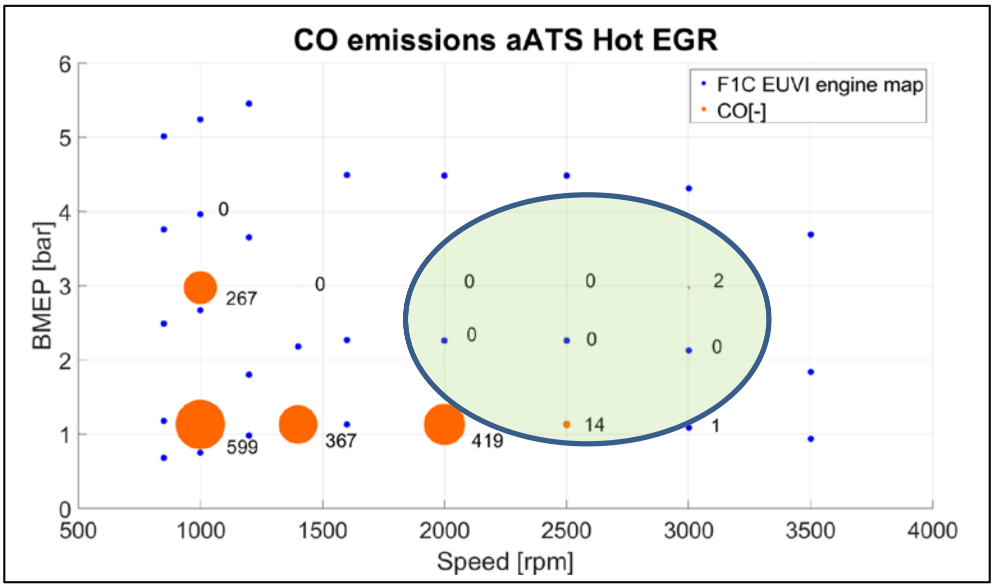
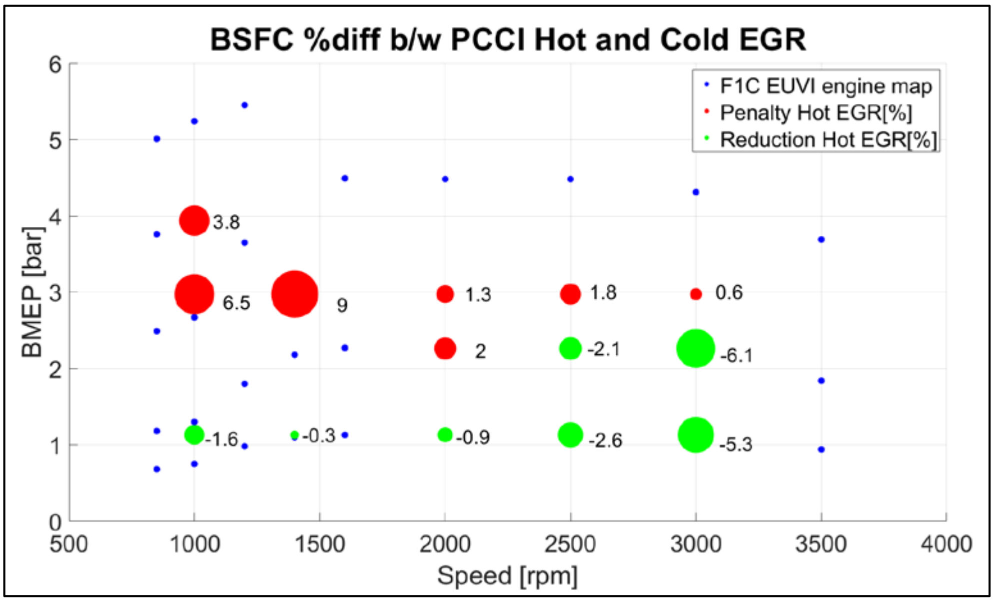
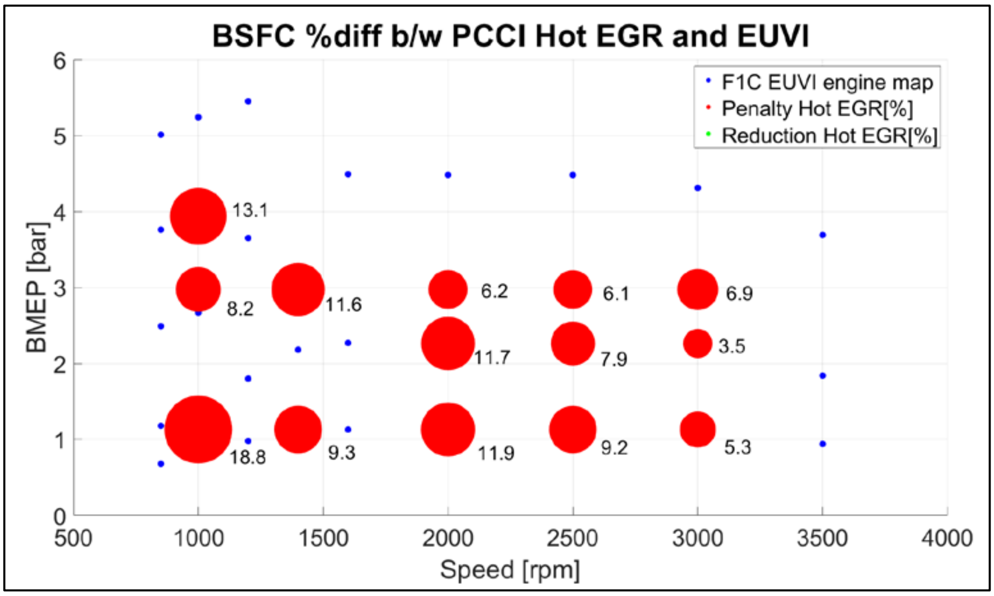


Chart 59 – CO emissions aATS Hot EGR

6.3.11 BSFC difference

In some points, hot EGR seems to be slightly beneficial with respect to cold EGR even as far as BSFC is concerned, so this can be a good results and we can by-pass the EGR cooler if we want to minimize the BSFC penalty respect to the EUVI, that is always negative.





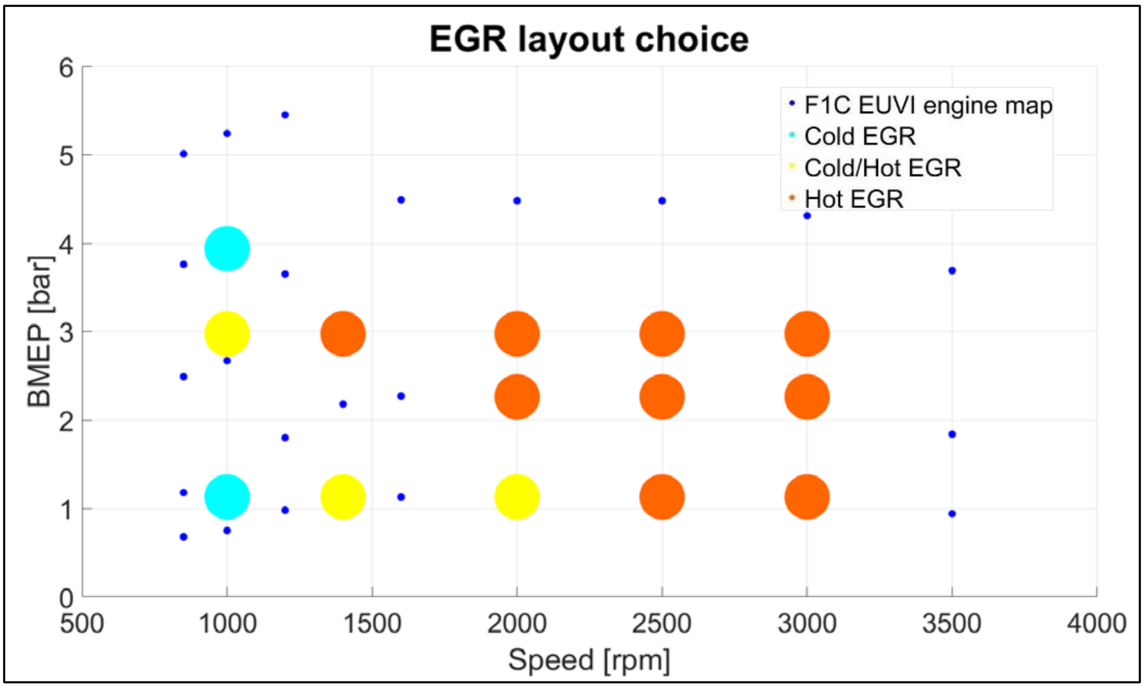
6.3.12 EGR layout choice

Based on the results shown, at lower engine speed and lower loads both cold and hot EGR do not reach DOC light-off temperature. As engine-out HC and CO emissions are generally higher for hot EGR configuration, cooling the EGR mass could be a better solution in this case.

The highest load in which we have a comparison is 3.94bar BMEP: above this load, in the case of hot EGR, a combustion deterioration is detected, which produces high levels of NOx and soot pollutant emissions. Therefore, it is not recommended to use uncooled EGR.

In the intermediate zone (yellow points) at the lowest tested load (1.13 bar BMEP), pollutant emissions (in particular engine-out HC and CO,) are lower in the cooled EGR case, but higher temperature at the exhaust are reached with hot EGR, so that a possible use of the DOC in close-coupled position could abate HC and CO after catalyst.

At higher load (2.27 bar BMEP) and high engine speed, it seems always better not to cool the EGR mass, thanks to the DOC efficiency around 100% that can be reached due to higher temperatures.



Conclusions

In the previously investigation with PCCI combustion mode we found different problems related to the fouling of EGR cooler that affects all the results and make them unreliable, especially at low load, where the exhaust gas recirculated speed inside of the cooler is low and the deposits are more likely to be form.
At low load we have also the problem related to very cold exhaust gas that make the light-off time of the DOC very long.

For this reason, in this thesis, I analysed the impact of hot EGR on the performances, at low load, on PCCI diesel engine, in order to see if I can mitigate the problem of the EGR fouling and also improve the light-off time of the DOC without change the layout of the catalyst.

The results of this solution are very good at medium and high revolution speed but there is limitation on the load because, increasing the load we increase the fuel injected so the exhaust temperature rise at level that is detrimental for the combustion stability and also for the integrity of the turbocompressor group.

At very low load the difference in temperature doesn't justify our decrease in volumetric efficiency at the intake, so it is preferable continuing to use the cold EGR and have the less BSFC.

In the future, to mitigate the fouling of EGR cooler we can try to insert a small DOC in the EGR line, before the EGR cooler, or using an active system to clean the EGR cooler without dismounted it, as I talk about in the thesis.

Last but not least, we can have another advantage if we put the DOC close to turbine, to have less heat lost by the pipe that links the 2 devices.

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Do what you love

D. Simeone