POLITECNICO DI TORINO

Master of Science
in Mechanical Engineering

Master Thesis

Development of simulation methodologies and analysis of advanced VVA and Cylinder Deactivation strategies for Diesel engines

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Abstract

Due to air pollution and global warming being very actual problems, the regulations demand increasingly stringent limits in terms of CO_2 and pollutants emissions. Thanks to their higher efficiency in respect to the gasoline counterpart, powertrains of the Diesel kind remain a key technology for reaching the CO_2 targets, even for electrified applications.

However, diesel engines suffer from greater NO_x and particulate matter emissions, and so require proper after-treatment systems. Such systems need high temperatures to work with an optimal conversion efficiency of the pollutants. Then, thermal management strategies, having as objective the warm-up of the after-treatment, are necessary to increase the engine exhaust temperatures, in particular at low loads.

The aim of this thesis, conducted in collaboration with the simulation team of FEV Italy, has been to develop and investigate simulation methodologies and control strategies for technologies designed for the warm-up of the after-treatment systems. In particular, the focus was on Variable Valve Actuation (VVA) and cylinder deactivation strategies.

The used methodology has been the co-simulation between an Engine Control Unit model in Matlab/Simulink environment and a Diesel engine model, built in the commercial software GT Power, the results of which are used for simulating an after-treatment system (made of DOC, SDPF and passive SCR). For a 4-cylinder 2-liters Diesel engine, the simulations have been run on a low-loads driving cycle, and the tested strategies have been Internal EGR (iEGR), Late Intake Valve Closure (LIVC) and cylinder deactivation (CDA).

After an analysis of the simulation results, the three strategies have been compared in terms of fuel consumption, NO_x emissions and thermal management, in respect to a baseline simulation. In particular, LIVC showed a limited impact on fuel consumption and negligible on tail-pipe emissions; iEGR provided the lowest NO_x emissions, both engine-out and tail-pipe; CDA proved to be the best both in terms of fuel consumption and after-treatment efficiency, thanks to the highest exhaust temperatures.

Sommario

Essendo l'inquinamento dell'aria e il riscaldamento globale due problemi molto d'attualità, le regolamentazioni richiedono limiti sempre più stringenti in termini di emissioni di CO_2 e inquinanti. Grazie alla loro maggiore efficienza rispetto alle controparti benzina, i powertrain di tipo Diesel rimangono una tecnologia chiave per raggiungere i target di CO_2, anche in vista applicazioni di tipo elettrificato.

Tuttavia, nei motori diesel le emissioni engine-out di NO_x e particolato sono influenti e richiedono sistemi di post-trattamento appropriati. Tali sistemi hanno bisogno di temperature elevate per lavorare con efficienze di conversione ottimali degli inquinanti. Sono quindi necessarie strategie di thermal management, con obiettivo il warm-up dei sistemi di post-trattamento, per aumentare le temperature allo scarico del motore, in particolare ai carichi parziali.

L'obiettivo di questa tesi, condotta in collaborazione con il team di simulazione di FEV Italia, è stato quindi quello di sviluppare e investigare metodologie di simulazione e strategie di controllo per tecnologie atte a migliorare il warm-up dei sistemi di post-trattamento. In particolare, si tratta di strategie di attuazione variabile delle valvole (VVA) e disattivazione cilindri.

La metodologia utilizzata è stata la co-simulazione tra un modello motore Diesel, sviluppato sul software commerciale GT Power, e un modello di Engine Control Unit in ambiente Matlab/Simulink, per poi utilizzare i risultati per la simulazione del sistema di post-trattamento (costituito da DOC, SDPF e SCR passivo). Per un motore 4-cilindri di 2 litri, sono state effettuate simulazioni transitorie su un ciclo guida a bassi carichi di due strategie di VVA, Internal EGR (iEGR) e Late Intake Valve Closure (LIVC), e una di disattivazione cilindri (CDA).

Dopo un'analisi dei risultati, le tre strategie sono state confrontate in termini di consumi, emissioni di NO_x e thermal management, rispetto ad una simulazione baseline. In particolare, la strategia di LIVC ha mostrato un impatto limitato sui consumi e trascurabile in termini di emissioni tail-pipe. La strategia di Internal EGR ha mostrato la maggiore riduzione di NOx sia engine-out che tail-pipe. La strategia di CDA si è rivelata la migliore sia in termini di consumi, che di efficienza del sistema di post-trattamento, grazie alle più alte temperature allo scarico.

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1 Introduction

Due to air pollution and global warming being very actual problems, the regulations demand increasingly stringent targets in terms of CO_2 and pollutants emissions reduction. The main way to achieve less CO_2 production is to improve the fuel economy of the vehicle, and diesel engines are usually more fuel-efficient compared to spark-ignited engines. However, diesel engines suffer from greater NO_x and particulate matter engine-out emissions, and thus require a proper after-treatment system for tailpipe pollutants abatement.

Modern Diesel engines are equipped with Diesel Oxidation Catalyst (DOC) to oxidize hydrocarbons and carbon monoxide, Diesel Particulate Filter (DPF) to reduce particulate matter emissions, and Selective Catalytic Reduction (SCR) to reduce NO_x emissions. Such components require proper temperatures in order to work effectively in reducing emissions. Thermal management strategies are then required, in order to obtain higher exhaust gas temperature, especially during cold start and low-load engine operation.

While electronically controlled fuel injection systems, Exhaust Gas Recirculation (EGR), and Variable Geometry Turbochargers have become standard in diesel engines, air management through Variable Valve Actuation (VVA) has not yet achieved a large popularity. In gasoline engines, VVA is capable of obtaining major fuel improvements, because it can replace, at least partially, the throttling for the load control, reducing the correspondent pumping losses. Instead, in diesel engines the load control is achieved by adjusting the amount of injected fuel, and therefore VVA has less potential of reducing the pumping losses. However, VVA can be a key technology for improving the exhaust warm up, aspiring at having no fuel consumption penalties, and thus is becoming highly desirable also for diesel engines.

A variable valve train can also be taken advantage of to implement Cylinder Deactivation (CDA) strategies. At low loads, the torque target can be achieved also by a lower number of cylinders than the engine has, and then cylinders can be deactivated, by interrupting their fuel injection and closing the intake and exhaust valves for the entire CDA phases. The remaining active cylinders, during CDA phases, work with higher load than normal operation, resulting not only in lower fuel consumption, but also in higher exhaust temperatures.

The work for this master thesis has been conducted in collaboration with FEV Italia s.r.l. and supported by its simulation team. An existing 4-cylinder 2-liters diesel engine model, developed in the commercial software GT Power and provided by FEV, has been modified to implement two different VVA strategies, Late Intake Valve Closure (LIVC) and Internal EGR (iEGR) through Exhaust Valve reOpening (EVrO), and CDA technologies. A calibrated Engine Control Unit (ECU) model, also provided by FEV and built in Matlab/Simulink

environment, has been revised to adapt it for the strategies above.
Co-simulations between the engine and the ECU models have been performed for evaluating fuel consumption and engine-out quantities, after a transient driving cycle with low loads. Then, a 0D after-treatment model has been used for evaluating the tailpipe emissions. The results analysis focused on the impact of LIVC, iEGR and CDA on fuel consumption, NO_x engine-out and tailpipe emissions, in respect to the baseline simulation. The exhaust temperatures have also been of interest, in order to evaluate the effectiveness of such strategies for the after-treatment system warm-up.

Figure 1.1: FEV

2 Compression Ignition Internal Combustion Engines

2.1 General overview of Diesel engines

In compression-ignition (CI) engines, air alone is inducted into the cylinder and the fuel is injected directly into the engine cylinder just before the combustion process is required to start [1].

Torque control is achieved by varying the air/fuel ratio in the combustion chamber. In naturally aspirated engines, the air flow at a given engine speed is basically constant, and it is the amount of fuel only to determine the load. Modern engines can be turbocharged, with inlet air compressed by a turbine-compressor combination driven by exhaust gases, or supercharged (even though it is not common in Diesel engines), with air compressed by a pump or blower driven mechanically. Thanks to turbocharging or supercharging, engine output torque can be increased by increasing the air mass flow per unit displaced volume, therefore allowing a greater fuel flow.

Considering the ideal thermodynamic cycles with same compression ratio ϵ (ratio between maximum cylinder volume and minimum cylinder volume), Diesel cycle of CI engines has a lower theoretical efficiency than Otto cycle, typical of spark-ignition (SI) engines. However, differently from SI engines, CI engines are not limited by combustion anomalies (knock), allowing much higher compression ratios, in the range 12 to 24, making them more efficient.

The following sequence of events describes an engine cycle of a typical four-stroke CI engine:

1. Intake: air is inducted in the cylinder, opening the intake valve;
2. Compression: air is compressed by the piston and brought to higher pressure and temperature;
3. Injection: before Top Dead Center (TDC), fuel injection into the engine cylinder starts, the jet atomizes into drops, evaporates and mixes with air;
4. Combustion: being air temperature and pressure above the fuel's ignition point, after a short delay period, auto-ignition of parts of the non-uniform fuel mixture initiates the combustion process, the flame spreads through that portion of fuel sufficiently mixed with air, and the cylinder pressure rises;
5. Expansion: the piston collects work, and mixing between fuel, air and burning gases continues, accompanied by further combustion;

6. Exhaust blowdown: during the expansion stroke, when the exhaust valve starts to open, the cylinder pressure is greater than the exhaust manifold pressure and a blowdown process occurs, with the burned gases that flow through the valve into the exhaust port;

7. Exhaust (forced): the piston displaces the burned gases from the cylinder into the manifold.

The fuels used in CI engines are typically very reactive, having a long and flexible molecular structure, so that intermediate reactions can occur rapidly at high pressure and temperature, leading to spontaneous ignition. The injection is realized using pump-injection systems or the so-called common-rail, which allows to precisely control the injection parameters, pressure and timing, and realize multiple injections (for example: pilot, main and after injections).

2.2 Combustion processes

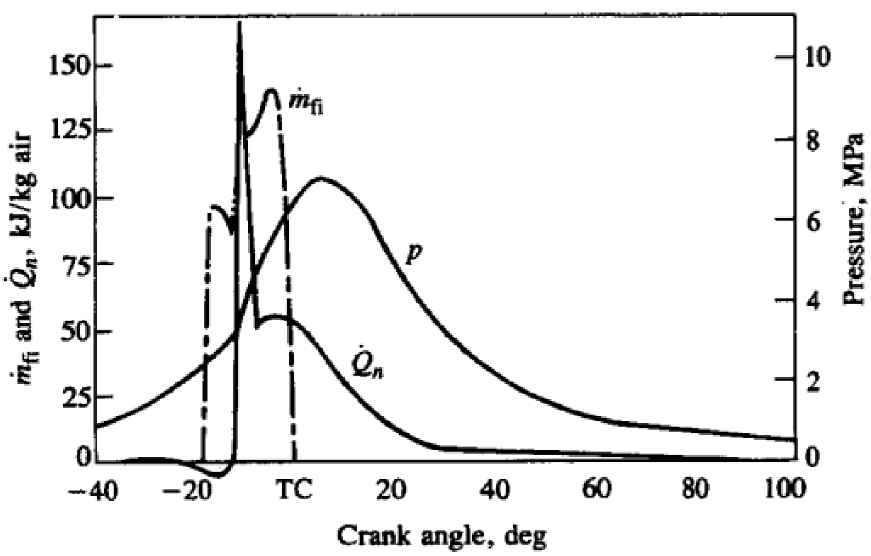

Figure 2.1: Cylinder pressure p, rate of fuel injection $\dot{m}_f$ and net heat release $\dot{Q}_n$ for direct injection diesel engine

Showing typical data of cylinder pressure and fuel mass flow rate in a Diesel engine, a delay between start of injection and start of combustion (identified by a change in slope of pressure curve) can be noticed. In fact, the injected fuel has to heat, vaporize and mix with air. When the accumulated fuel auto-ignites, pressure rises rapidly for a few crank angle degrees, then increases more slowly reaching its peak value.

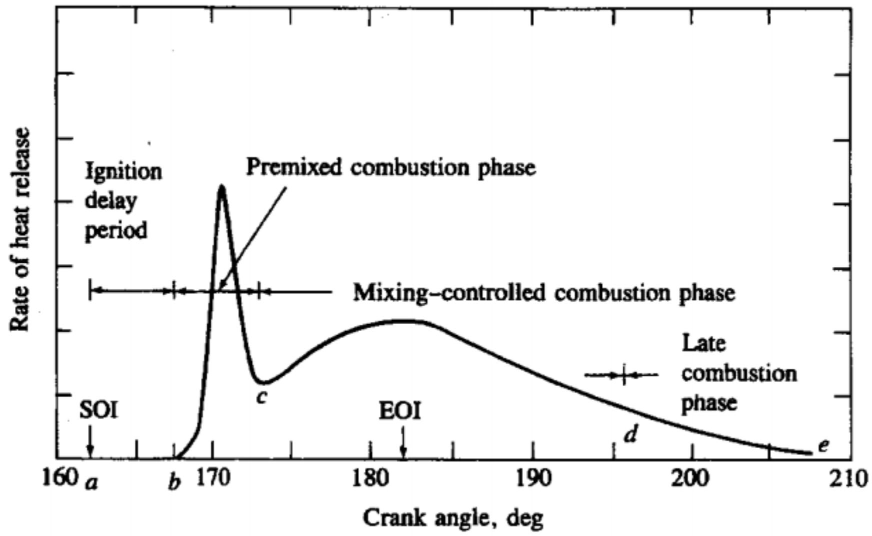

Figure 2.2: Heat-release-rate diagram of a direct injection diesel engine, identifying different combustion phases

Observing the rate of heat-release, four combustion phases can be identified:

- *Ignition delay (ab)*: after the start of injection, before the start of combustion, a slight loss of heat occurs, due to heat transfer to the walls and to fuel vaporization and heating;

- *Premixed or rapid combustion phase (bc)*: combustion of fuel which has mixed with air during ignition-delay, and reached auto-ignition conditions, occurs rapidly in a few crank degree angles. This phase is responsible of the typical noise of Diesel engines, and very high temperatures are reached, favouring NO_x formation;

- *Mixing controlled phase (cd)*: once the fuel and air which premixed during the ignition delay have been consumed, the burning rate is controlled by the rate at which mixture becomes available for burning, through the vapor-air mixing process. In this phase, the heat-release rate may reach a second (usually lower) peak;

- *Late combustion phase (de)*: heat release continues at a lower rate into the expansion stroke, because a small fraction of the fuel may not yet have burned and combustion can also involve the soot formed in the previous phase.

2.3 Emissions, NOx formation and control

The combustion reaction of a generic hydrocarbon C_aH_b ($b/a \approx 1.85$ for diesel and gasoline fuels) in ambient air ($79\%_{vol}\ N_2, 21\%_{vol}\ O_2$) is given by:

$$C_aH_b + (a + \frac{b}{4})(O_2 + 3.773N_2) \rightarrow a\ CO_2 + \frac{b}{2}\ H_2O + 3.773\ (a + \frac{b}{2})N_2$$

From the ideal combustion, no toxic products are produced, but carbon dioxide (CO_2) has to be limited, because it is a greenhouse gas and it is responsible of global warming. The only way to decrease CO_2 emissions would be increasing the powertrain efficiency, so that fuel consumption is lower.

In reality, the combustion is not ideal and complete, and pollutants are always emitted. From incomplete oxidation, carbon monoxide (CO) and hydrocarbons (HC) of unburnt fuel or intermediate species can form. Nitrogen from air can oxidize to NO_x (nitrogen oxides: NO and NO_2) because of the high temperature in the combustion chamber. In Diesel engines, particulate matter (PM) is generated as well and consists of elemental carbon particles which agglomerate and absorb other species (mainly unburned hydrocarbon and sulfates), forming complex structures sized from a few tens of nanometers to several hundred microns. The main pollutant species in CI engines are NO_x, with usually 70-90% NO and 30-10% NO_2, and PM.

As for NO_x formation, the main mechanism is the thermal one, known as the extended Zeldovich mechanism. The three chemical reactions involved are:

$$O + N_2 \rightarrow NO + N$$

$$N + O_2 \rightarrow NO + O$$

$$N + OH \rightarrow NO + H$$

The first reaction has a very high activation energy, resulting in a high sensitivity to temperature. High temperatures (T > 2000 K) are required and this mechanism is important at air/fuel mixtures that are close to stoichiometric, such as in the flame zone of a conventional burning diesel spray. As the temperature in the combustion chamber drops during the expansion stroke, the NO concentration can freeze shortly after the end of heat release.

NO formed in the flame zone can be rapidly oxidized to NO_2, with a reaction like:

$$NO + HO_2 \rightarrow NO_2 + OH$$

Then, conversion of this NO_2 to NO occurs:

$$NO_2 + O \rightarrow NO + O_2$$

The last reaction does not occur if the NO_2 formed in the flame is quenched by mixing with cooler fluid. In Diesel engines, at low loads the are several cool regions of fluid, that block the re-conversion to NO, resulting in a higher NO_2/NO ratio.
The diffusion-burning region of combustion is the primary source of NO_x, as rapid formation of NO_x begins after the start of heat release, and formation ends after the end of heat release, because temperatures of the burned gas decrease due to mixing with cool gas and expansion of the charge. As for the premixed portion of the fuel, even if NO formation is negligible during premixed combustion, this portion is compressed to a higher pressure and temperature, reaching the highest temperature of any portion of the cylinder charge.
Then, techniques to control NO_x focus on the premixed phase of combustion, aiming to reduce combustion temperatures, which can lead to penalties in HC or PM emissions, and fuel consumption. For this reason, it is common to talk about trade-offs between NO_x and PM, and between NO_x and fuel. Some operating parameters that affect NO_x emissions are: air-to-fuel ratio, injection timing advance, and EGR rate.
Despite Diesel engines always operate under lean conditions (global air-to-fuel ratio > 1), the local air-to-fuel ratio in the chamber is inhomogeneous, ranging from very rich to pure air zones. As a consequence, there are zones where combustion takes place at favorable conditions for NO_x formation. With decreasing λ (relative air-to-fuel ratio), which is equivalent to an increasing load, NO_x concentration increases.

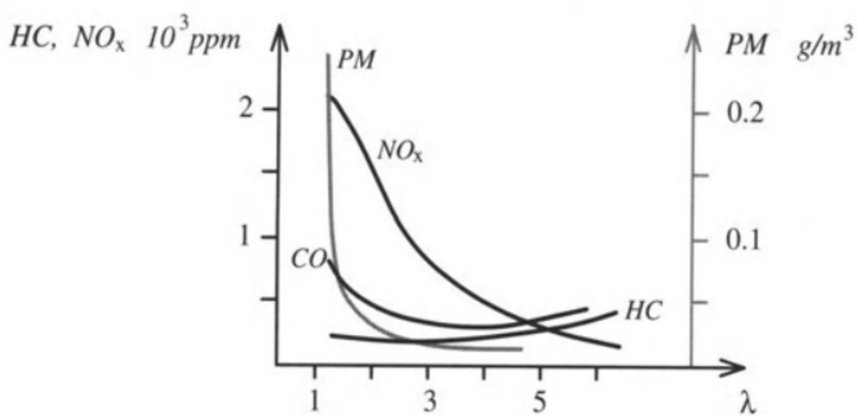

Figure 2.3: NO_x, HC and PM engine-out emission for a direct injection Diesel engine as a function of air-to-fuel ratio λ

An injection timing advance would extend the ignition delay period, because the fuel is injected when the chamber has a lower pressure and temperature; then, a greater portion of fuel is injected during the ignition delay period, with a better chance to mix with air; the premixed portion of fuel is greater and leads to more NO_x formation. For this reason, in order to decrease NO_x emissions, a retarded injection timing is required, so that the premixed fuel is reduced, but with the side effect of increasing fuel consumption.

Exhaust Gas Recirculation (EGR) is a common technique for reducing NO_x emissions, by recirculating a portion of exhaust gas back to the engine cylinders. While fresh air contains a negligible portion of CO_2, the recirculated gas has a CO_2 quantity that increases with increasing load and EGR flow rate. EGR rate can be defined as the ratio between the intake and the exhaust CO_2 concentrations:

$$EGR_{\%,vol} = \frac{CO_{2,intake\%}}{CO_{2,exhaust\%}}$$

With EGR, NO_x formation decreases because of the decreasing combustion maximum temperatures, mainly thanks to the dilution effect of the fresh air with exhaust gas, with the reduction of oxygen concentration. Moreover, exhaust gases having greater heat capacity (thermal effect), and CO_2 dissociation (chemical effect) further contribute to the decrease in combustion temperatures.

Since Diesel combustion is always lean, exhaust gases still contain significant amount of oxygen, decreasing with increasing engine load. Then, EGR% is not a very suitable parameter for predicting NO_x emissions, and should be replaced by intake oxygen concentration. The efficacy of EGR decreases with increasing load, considering that at low loads the exhaust contains higher oxygen concentration. In general, maximum EGR is limited by the consequent increase in PM, HC and CO emissions, due to the lower oxygen availability.

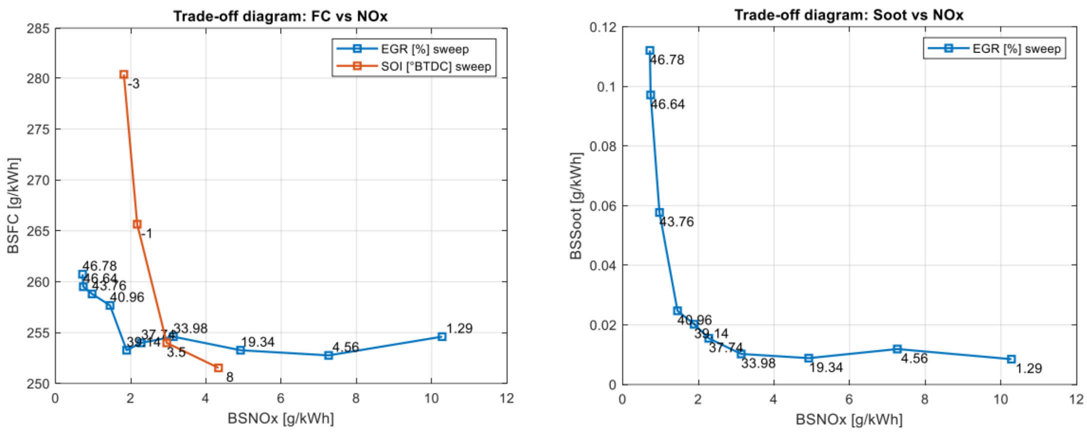

Figure 2.4: Example brake specific FC / NO_x (left) and Soot / NO_x (right) trade-off diagrams, with EGR% and injection timing advances (start of injection, degrees before top dead center)

2.4 After-treatment systems

With emissions regulations becoming more stringent, it is not sufficient to meet the target by only reducing engine-out pollutants. Therefore, pollution abatement systems become necessary, in order to reduce tailpipe emissions. In Diesel engines, the most critical issues for after-treatment systems concern the NO_x reduction and PM oxidation. Also, considering that a trade-off between NO_x and fuel consumption / PM exists, an high efficient NO_x reduction system could let the engine operate more close to its optimal BSFC (brake specific fuel consumption) and at low soot conditions.

The type of after-treatment has to be chosen depending on engine application, between low-duty and heavy-duty, and calibration parameters. In general, these systems need sufficiently high operating temperature to efficiently work, and then it is essential to develop "warm-up" strategies and technologies, that mostly increase the exhaust gas temperature of the engine. A standard configuration of after-treatment for Diesel engines should have in order DOC, DPF, and LNT or SCR. An overview of these technologies will be given.

Diesel Oxidation Catalyst (DOC) promotes the oxidation of CO and HC by means of platinum (Pt) on an alumina washcoat (Pt/Al_2O_3), with a ceramic monolith substrate. With sufficiently high exhaust temperatures, the conversion efficiency of CO and HC can reach values over 90%. The catalytic oxidation of Soluble Organic Fraction (SOF) also occurs, resulting in reduction of PM emissions, with an efficiency between 15% and 30%. Another major effect is the oxidation of NO into NO_2, resulting in increased NO_2/NO_x ratios downstream of the DOC, which leads to benefits for the following after-treatment units (SCR in particular).

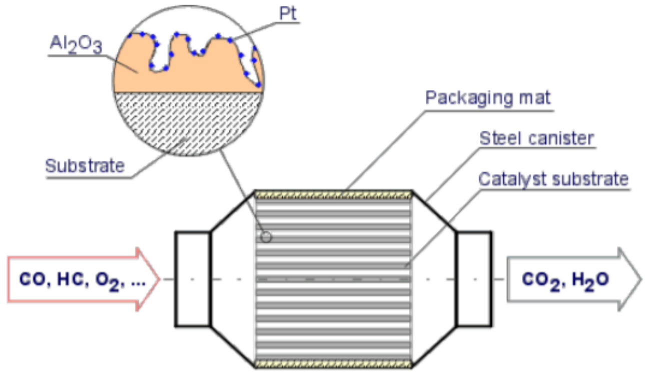

Figure 2.5: Diesel Oxidation Catalyst (DOC)

Diesel Particulate Filter (DPF) operates like a mechanical filter, physically capturing solid particles in the gas stream, to prevent their release to the atmosphere, with the potential to reach filtration efficiencies of 90%. While filters are most effective in controlling the solid fraction of diesel particulates, including soot (elemental carbon), they may have limited effectiveness in controlling non-solid fractions of PM, such as SOF and sulfates. At first, filtration occurs through the porous wall (deep bed filtration) and then, when the first layer of the porous media has been saturated, soot starts accumulating on the wall surface (cake filtration). The more the soot accumulated, the more is the pressure drop through the DPF. Then, DPF needs to be periodically regenerated, in order to burn the accumulated soot. Regeneration needs high temperatures (above 600 °C), which may be hardly reached in the exhaust gas, especially under partial loads. Specific regeneration (RGN) strategies are necessary, with the objective of increasing the exhaust gas temperature, for example using multiple injections, but with several issues, such as wasting fuel into heat and so increasing fuel consumption.

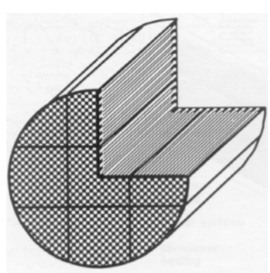

Figure 2.6: Diesel Particulate Filter (DPF)

Lean NO_x Traps (LNT) are used for reduction of NO_x emissions, by working under alternative cyclic condition of lean and rich phases. Barium and lanthanum oxides can adsorb the NO_2 formed by catalytic oxidation during lean operating conditions. When additional fuel is post-injected, nitrates reduction to nitrogen is induced, and the regeneration of the adsorber occurs. The major issue of NO_x adsorbers is their susceptibility to sulfur poisoning, and then also dedicated combustion modes with rich mixture and high temperature are needed for periodic desulfurization.

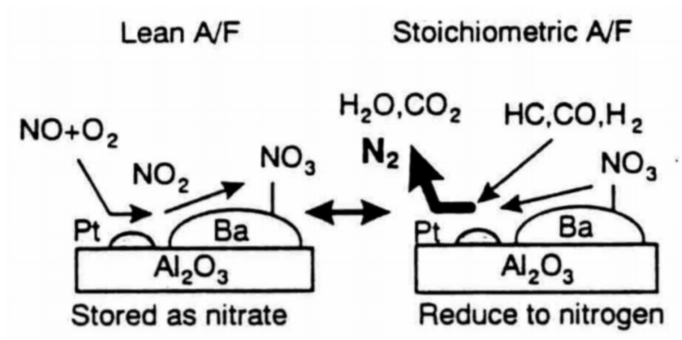

Figure 2.7: Lean NO_x Trap (LNT) storing and reduction mechanisms

Selective Catalytic Reduction (SCR) is a more efficient but also more costly and space requiring alternative to LNT, and then is more common for medium and heavy duty applications. A reducing agent, such as urea $CO(NH_2)_2$, is injected into the gas upstream of the catalyst bed, resulting in high NO_x conversion efficiency, reaching 90%. The injected fluid is referred to as Diesel Exhaust Fluid (DEF), and one of the more famous commercially available is "AdBlue", an homogenous solution of water and urea. The DEF is stored in a specific tank and injected through a dosing valve on the exhaust line. The injected fluid originates ammonia (NH_3), through thermolysis and hydrolysis, which deposits on the substrate of the catalyst, where the NO_x reductions take place, depending on the SCR temperature. One of the methods to improve the low temperature performance of the catalyst is to increase the NO_2 content in the exhaust gas (for example, through a DOC). One of the issues of SCR is the generation of ammonia emissions (ammonia slip), mostly in dynamic operating conditions, that can be limited through a closed loop control of injected urea. Selective Catalytic Reduction with hydrocarbons (instead of ammonia as reducing agent), namely deNO_x or lean NO_x catalyst (LNC), is also possible, using either HC emitted by the engine (passive deNO_x) or else through the addition of extra HC, such as diesel fuel (active deNO_x).

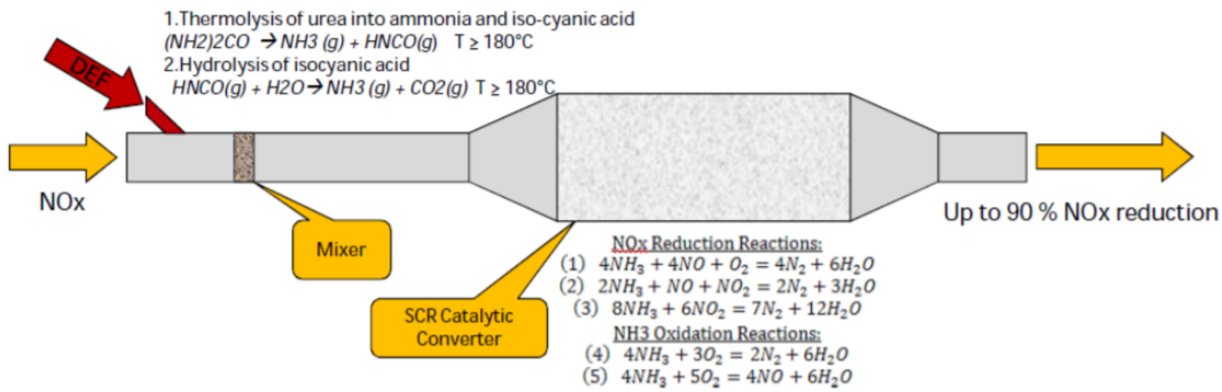

Figure 2.8: Selective Catalytic Reduction (SCR) with urea

The described after-treatment systems can also be combined into a more advanced single unit. The four-ways catalyst developed by Toyota integrates a NO_x adsorber and a DPF on one substrate.
Another example is the SDPF (SCR coated DPF), that integrates the SCR on the Filter, with light-off advantages and the possibility of NO_x reduction during regeneration phases. This configuration eliminates the flow-through SCR substrate and reduces the space for packaging. Thanks to the lower volume, the thermal capacity is lower and the warm-up is quicker.

2.5 Diesel engines control structure

The objective of electronic Diesel engine control systems is to provide the required engine torque, under the constraint of meeting the given exhaust gas and noise emission regulations [2]. In stationary and dynamic conditions, three important paths have to be optimally coordinated: the fuel path, the air path, and the EGR (exhaust gas recirculation) path. A speed feedback controller is also common, which at least has to guarantee the top speed limit is not reached.
The fuel path receives its inputs from the speed controller and outputs torque, speed and exhaust gas emission. Its control inputs are start of injection, injection duration, and injection pressure; with common-rail systems, the injection can be split into pilot, main and after quantities. Depending on the air mass flow into the engine, the control limits the quantity of injected fuel, in order to limit soot emissions (smoke limit, for which air/fuel ratio has a lower boundary).
The air path is mainly dominated by the turbocharger. At high loads, boost pressure has to be limited, because after the in-cylinder compression it would lead to very high engine

pressures. In order to do so, a fraction of the exhaust gas has to bypass the turbine through a waste gate valve, that has the disadvantage of generating considerable losses. As a more efficient alternative, Variable Nozzle Turbochargers (VNT) can be used and controlled with a closed-loop approach, where the target is the boost pressure.
Exhaust Gas Recirculation (EGR) is used for NO_x emissions reduction. From the control point of view, the EGR path doesn't take as objective an EGR rate, but it is the measured air mass flow to be the feedback. The air flow meter measures the intaken Mass Air Flow (MAF), the control compares the actual MAF value with a target value, which has been set up for each engine operating conditions, and actuates the EGR valve trying to reach the target MAF value. The EGR valve position is determined from the difference between the total intake flow and the measured air flow.
Particular attention has to be made in order to coordinate the controls. For example, the Engine Control Unit (ECU) controls the VNT position in order to achieve a boost target, but a greater boost would increase the quantity of EGR mass flow, assuming the EGR valve opening is the same. A simple way to solve the coordination problem would be giving a priority to one of the two paths and control the other one in open-loop, usually for a temporary period of time.

3 Emission standards and driving cycles

3.1 European emission standards

In the European Union, the need to tackle the problem of air pollution has lead to the progressive introduction of increasingly stringent standards for emissions. Such regulations limit CO_2 (carbon dioxide) emissions, as well as pollutants like NO_x (nitrogen oxide), THC (total hydrocarbons), NMHC (non-methane hydrocarbons), CO (carbon monoxide) and PM (particulate matter), depending on vehicle type (light-duty, heavy-duty, non-road) and engine type (gasoline, diesel).

The EURO standards have been introduced since 1992, being periodically updated, and since 2015 the EURO-6 standard has to be met by newly registered vehicles. Emission limits for passenger cars and light commercial vehicles are distance-specific (g/km or #/km for particulate number PN), while limits for heavy duty vehicles (trucks and buses) are energy specific (g/kWh). As for carbon dioxide, the new legislation defines the permitted emissions

Engine type	**CO (g/km)**	**HC**	**NO_x (g/km)**	**HC + NO_x (g/km)**	**PN (#/km)**
Petrol	1.0	0.1	0.06	-	6×10^{11}
Diesel	0.5	-	0.08	0.17	6×10^{11}

Table 3.1: Limits of EC Regulation EURO 6 for light duty vehicles

of CO_2 for new vehicles, depending on their mass. In this way, the target for 2015 was 130 g/km of CO_2 for the fleet average of all new cars, and the long term target will be 95 g/km for 2020. If average emission levels of their fleet are above the limit value curve, the manufacturers have to pay an excess emissions premium, based on the g/km that an average vehicle sold is above the curve, multiplied by the number of vehicles sold by the manufacturer.

Starting from the Euro 3 stage, vehicles must have an Engine On-Board Diagnostic (EOBD) system for emission control, that notifies the driver, in case of malfunction of the after-treatment system, that would cause emissions to exceed the EOBD regulated thresholds.

The emissions produced by a vehicle are strictly correlated to the engine operating conditions. Therefore, the emissions limits refer to a driving cycle, which consists of a standardized vehicle speed trace, based on statistical data, reproduced on an emissions test rig.

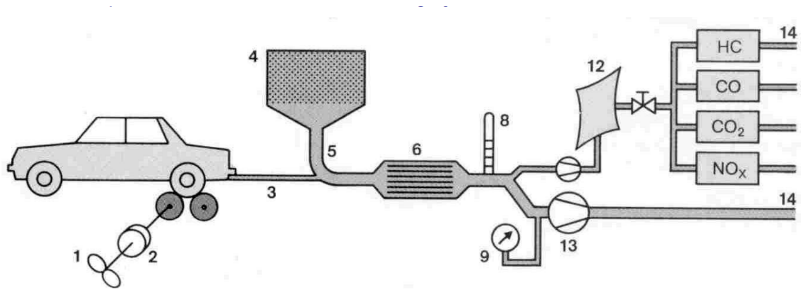

Figure 3.1: Vehicle emissions test rig

3.2 NEDC driving cycle

New European Driving Cycle (NEDC) was the designed cycle for EURO homologation lab-bench procedure, supposed to represent the typical usage of a car in Europe.

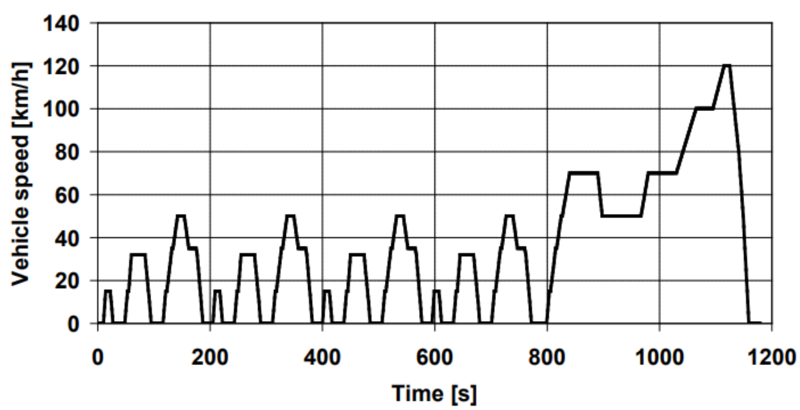

Figure 3.2: NEDC speed trace: UDC repeated four times + EUDC

The cycle consists of two parts, the ECE-15, repeated four times, and the EUDC. The Urban Driving Cycle (ECE-15 or UDC) represents city driving conditions and is characterized by low vehicle speed (maximum speed = 50 km/h), low engine load, and low exhaust gas temperature. The Extra Urban Driving Cycle (EUDC) is performed after the fourth ECE cycle, with more aggressive and high speed driving modes (maximum speed = 120 km/h).
NEDC has been criticized for the discrepancy between the type approval test and the real driving conditions. In fact, it can be considered out-of-date, as its most recent update was made in 1997 and it was conceived when European vehicles were lighter and less powerful. Also, real world driving contains way more accelerations and decelerations in respect to the NEDC, due to driver behavior and traffic conditions.

Cycle Phase	Repeated for	Time (s)	Distance (km)	Average speed (km/h)
UDC	4 times	4 x 195 = 780 s	4 x 1.013 = 4.052	18.7
EUDC	1 time	400	6.955	62.6
Total (NEDC)	-	1180	11.007	33.6

Table 3.2: Metrics of NEDC

3.3 WLTC driving cycle

From the 1st of September 2019, all the light duty vehicles that are to be registered in Europe must comply with the World harmonized Light-duty vehicle Test Procedure (WLTP), replacing the NEDC. The WLTP procedures includes three classes of Worldwide harmonized Light vehicles Test Cycles (WLTC), depending on the power-to-mass ratio (PMR), ratio of rated power (W) and curb mass (kg) not including the driver, and the maximum speed of the vehicle, declared by the manufacturer. Class 1 is for low power vehicles with PWR $\leq$ 22, class 2 is for vehicles with 22 $\leq$ PWR $\leq$ 34, and class 3 is for high-power vehicles with PWR $>$ 34.

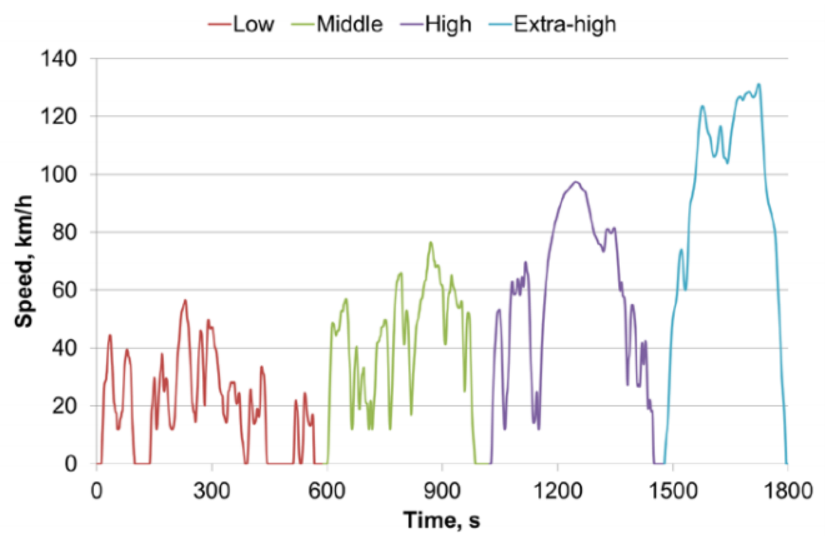

Figure 3.3: WLTC speed trace for class 3

Most common european cars belong to class 3. The cycle is mainly divided into four phases: low speed (L), middle speed (M), high speed (H) and ex-high speed (ExH). For example, in case of class 3, WLTC has a total duration of 1800 s, a total distance of 23.3 km, an average velocity of 46.5 km/h, and a maximum velocity of 131.3 km/h.

3.4 RDE test procedure

The Real Driving Emissions test consists of measuring the pollutants on-road, during a real driving divided in urban, extra-urban and motorway phases. A Portable Emissions Measurement System (PEMS) provides real-time monitoring of the pollutants.
Obviously, this procedure reflects more the real driving emissions, and allows to account for factors like the environmental conditions (such as external temperature) and the road incline, that the test cycle procedures wouldn't account for. An emission rig test procedure, with a standard cycle, can be subject to "cycle beating", which is forbidden by laws and consists of switch to a mode optimised for emissions performance, with an engine control system that recognizes that the vehicle is being tested. The high degree of randomness provided by RDE, prevents cycle beating.
For new EURO cars registered from September 2017, RDE testing over a range of conditions must be performed. RDE emissions limits are increased by a conformity factor CR in respect to the regulated emission limits. For example, with a conformity factor of 2 for NO_x, the on-road test limit is twice the regulated one. From January 2020 onward, a NO_x conformity factor of 1 plus an error margin set to 0.5 (for statistical and technical uncertainties) will apply for new vehicles.

3.5 TFL driving cycle

Transport For London (TFL) driving cycle [3] has been developed for laboratory testing of passenger cars under London driving conditions. In order to derive this test cycle, data logging has been made under a real route in London, representative of road and traffic conditions of the city. This route includes varying traffic conditions: free-flow, morning peak and mid-day inter-peak.

Cycle Phase	**Description**	**Time (s)**	**Distance (km)**	**Average speed (km/h)**
Phase 1	Urban	1656	8.38	18.2
Phase 2	Suburban	528	4.69	32
Phase 3	Motorway	216	4.48	74.8
Total	-	2400	17.56	26.34

Table 3.3: Metrics of TFL Passenger Car Drive cycle

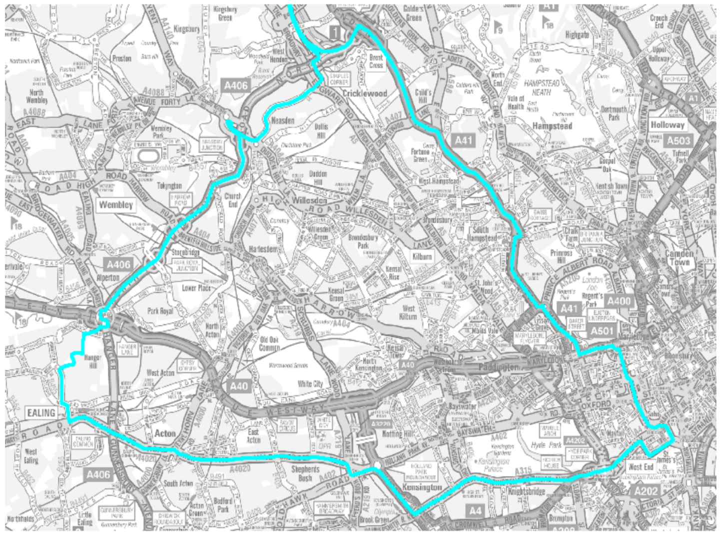

Figure 3.4: Map of the route chosen for TFL drive cycle development

The cycle consists of three different road types: urban (central and inner London), sub-urban (outer London) and motorway.

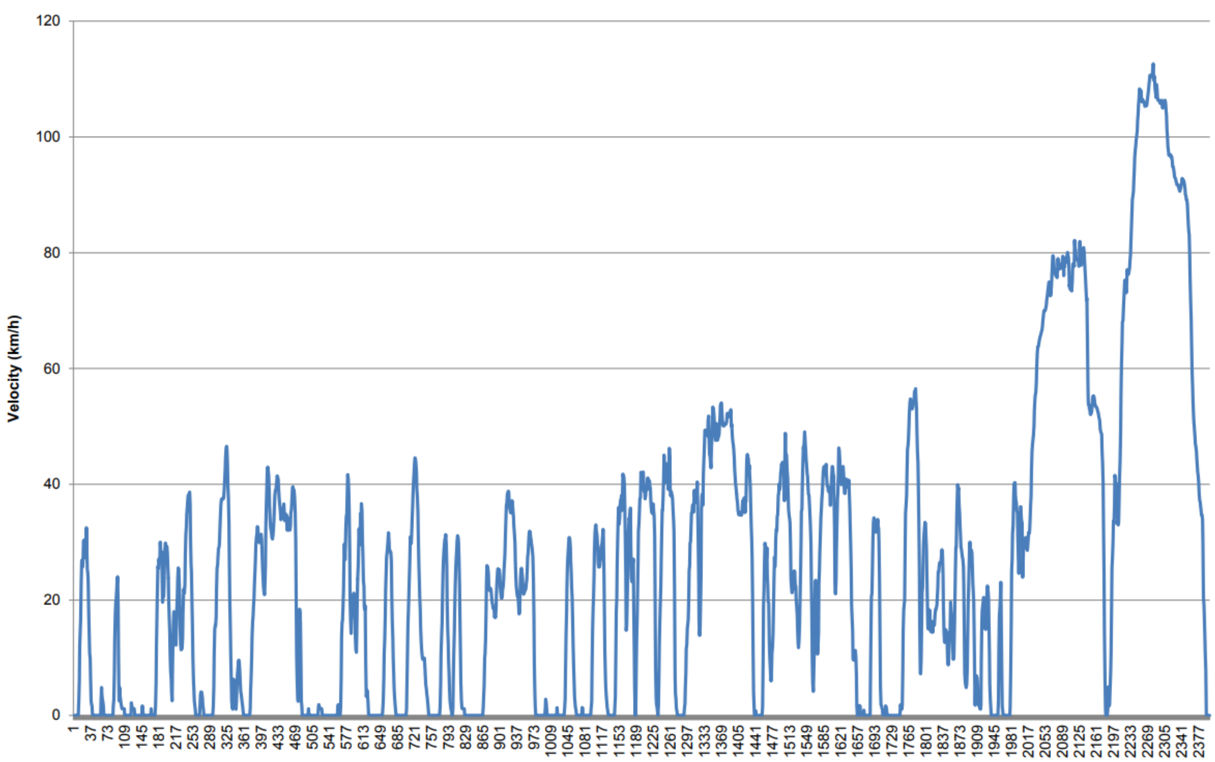

Figure 3.5: Speed trace of TFL drive cycle

From the speed trace, it can be seen how the TFL drive cycle is more transient than NEDC cycle, reflecting the real driving conditions in London, which are more of the "stop-start" kind. At 2,400 seconds, it is twice as long as the NEDC, with a 26.34 km distance in comparison to the 11 km of the NEDC.

4 VVA technologies

4.1 Variable Valve Actuation

While VVT (Variable Valve Timing) systems allow to obtain a variable phasing, VVA (Variable Valve Actuation) systems are capable of changing the entire lift profile of the intake or exhaust valve, in order to decrease fuel consumption and emissions at low loads.

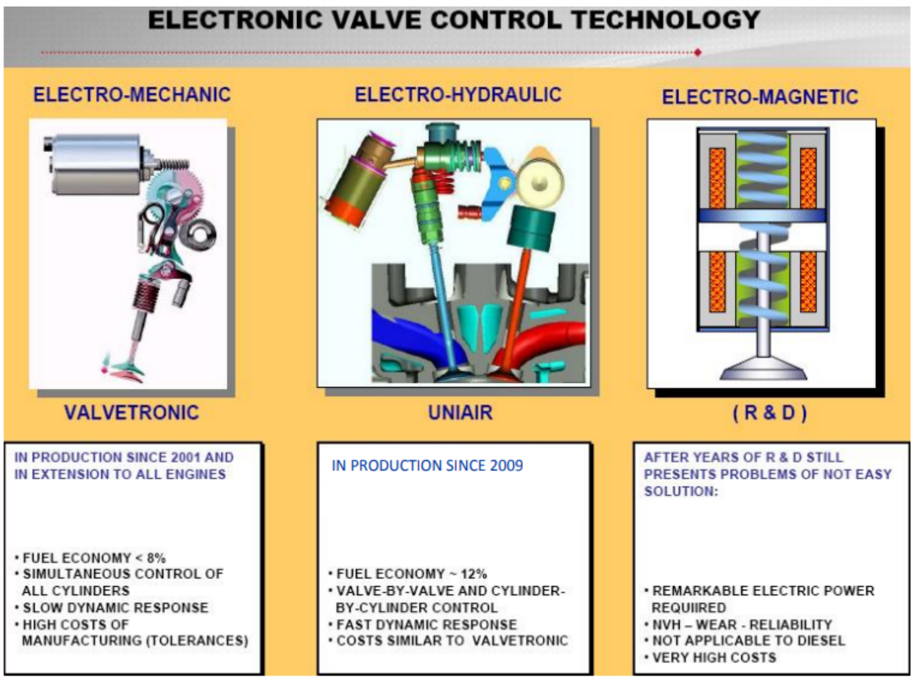

Figure 4.1: Most common VVA technologies

Electro-mechanic VVA, like the Valvetronic designed by BMW, can be used to obtain a continuous variation of valves lift and timing on the intake system. At low loads, this allows to eliminate the need of a throttle butterfly and so reduces the pumping work, using Early Intake Valve Closure or Late Intake Valve Closure strategies.
Similarly, electro-hydraulic system like Fiat MultiAir can be used to precisely control air intake without a throttle valve, with the same benefits described above, but with a faster dynamic response and a better behavior at high speeds.
Electro-magnetic camless VVA is currently in research and development. Actioning the valves with an electro-magnet, theoretically an instantaneous opening or closure is possible, but this would require a very high electric power. Opening and closure speeds are also limited by NVH (noise, vibration, harshness).

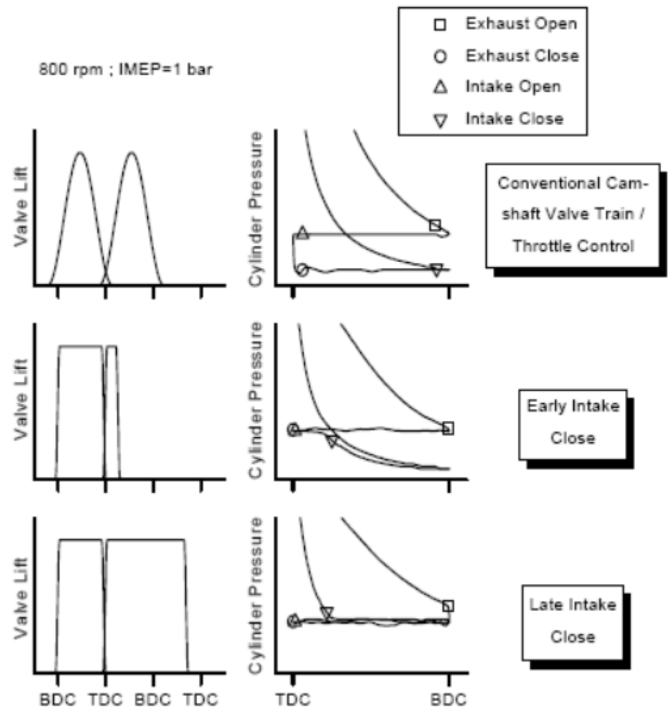

Figure 4.2: Effects of EIVC or LIVC on the pumping cycle

4.2 Internal EGR

VVA systems can also be used for an internal control of EGR rate, in order to achieve NO_x (nitrogen oxides) emissions reduction on Diesel engines. The recirculation of exhaust gas from the previous engine cycle is obtained by reopening the exhaust valve during the intake stroke or by a pre-lift of the intake valve during the exhaust stroke. The internal EGR increases the concentration of in-cylinder exhaust gas, thus reducing the combustion maximum temperatures due to the dilution effect.

"Analysis of Different Internal EGR Solutions for Small Diesel Engines" [4], by Millo et al., presented a comparison of the two iEGR solutions, exhaust valve post-opening and intake valve pre-opening, for a small Diesel engine. Through an experimental analysis, the exhaust post-opening valve profile is shown to produce an higher NO_x reduction, despite giving a lower value of internal EGR, in respect to the intake pre-opening solution, thus suggesting that the lower temperatures of the recirculated exhaust gases play a major role. Moreover, while the post-opening profile nearly does not influence the soot emissions, the pre-opening profile is quite penalizing. Both modified cam profile have negligible effects on the fuel consumption.

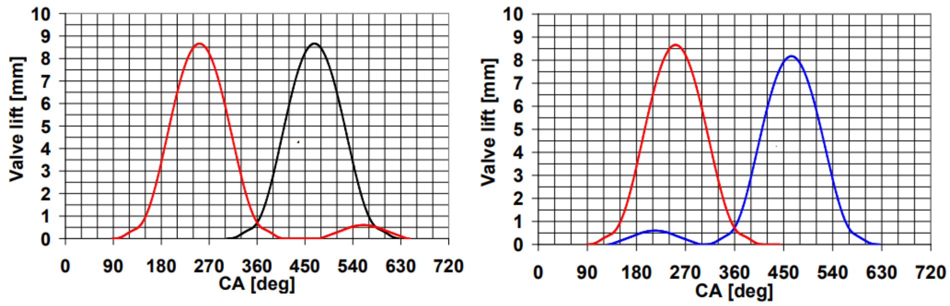

Figure 4.3: Internal EGR with Exhaust Valve reOpening (left) or Intake Valve preOpening (right)

”Internal Exhaust Gas Recirculation for Efficiency and Emissions in a 4-Cylinder Diesel Engine” [5], by Gonzales and Di Nunno, studies the impact of Internal EGR by exhaust rebreathing on transient test cycles. With the exhaust valve reopening, it can be noticed a reduction in exhaust back-pressure, and the significant effect of exhaust rebreathing on reducing pumping cycle losses. Thanks to the lower pumping losses, at lower loads (i.e. 2 bar) Internal EGR shows potential for not penalizing the BSFC. Under increasing engine load (i.e. from 2 bar to 5 bar), charge compression occurs at a higher initial temperature, resulting in greater heat transfer losses, and this tends to produce a less efficient compression process, penalizing the BSFC.

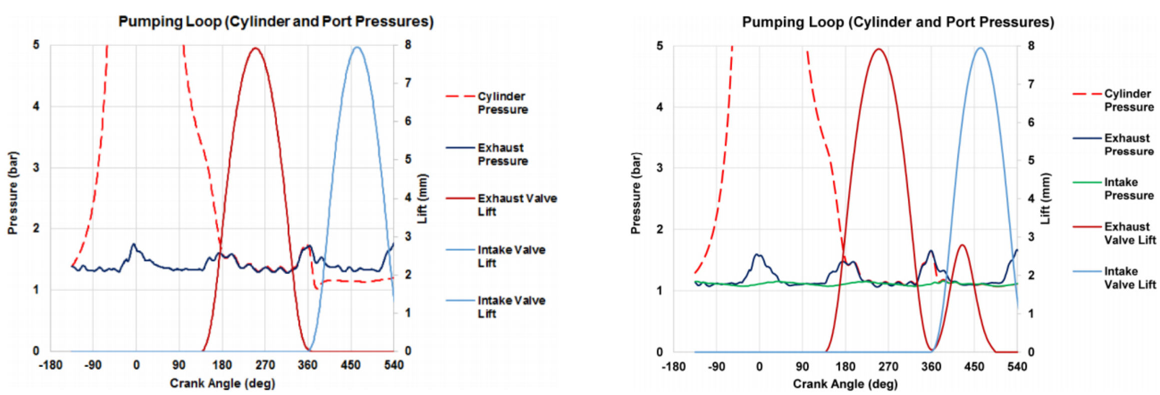

Figure 4.4: Engine system pressures for intake and exhaust events: baseline (left) and iEGR with exhaust reopening (right) [5]

Another benefit of Internal EGR is that it can accelerate the warm up of exhaust system after cold start, improving the after-treatment efficiency. However, the increase in exhaust gas temperature, which is particularly significant at low loads, is limited by the resulting increase in soot.

4.3 Late Intake Valve Closure

With LIVC (Late Intake Valve Closure), the intake valve is held open longer than normal, allowing a reverse flow into the intake manifold. Excess gas mass is pushed back during compression and then the effective compression ratio is reduced, getting smaller than the effective expansion ratio. The working cycle obtained with the retarded closure is called Atkinson cycle (also referred as Miller cycle).

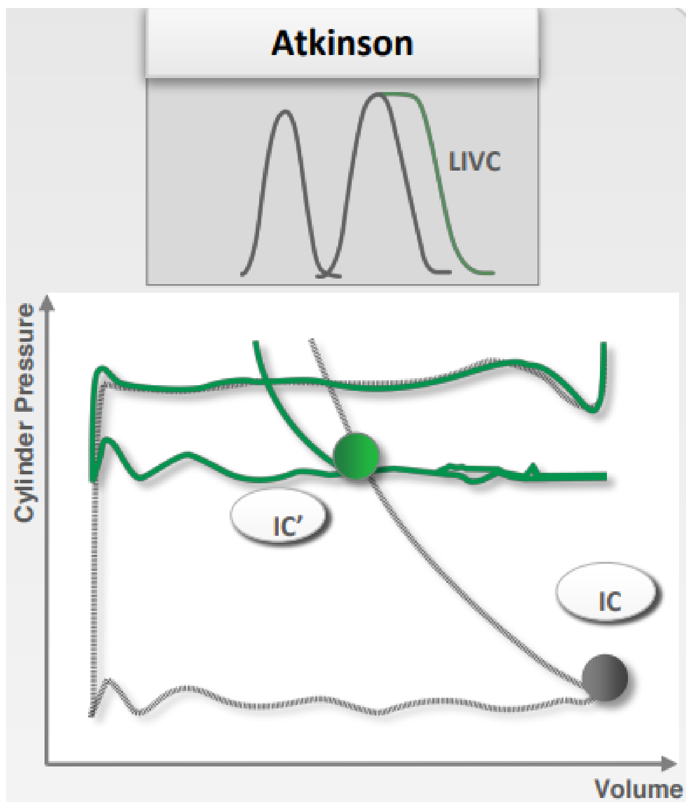

Figure 4.5: Atkinson cycle

As an advantage, the same charge mass can be trapped at lower temperature, or a bigger charge mass can be trapped at the same temperature.
"Numerical Assessment of the CO2 Reduction Potential of Variable Valve Actuation on a Light Duty Diesel Engine" [6], by Andrea Piano and Federico Millo, investigates on the effects of different LIVC angles with focus on fuel consumption. With the "Millerization" of the working cycle enabled by LIVC, significant temperature reductions at the end of compression stroke can be achieved, resulting in NO_x emissions reduction.

At low loads, LIVC allows efficiency improvements, because of the reduction of maximum pressure and the resulting friction. With a larger lift profile, the BSFC benefit is higher (figure 4.6, left). However, at higher load, the trend is opposite (figure 4.6, right), due to the increasing heat transfer losses.
In particular, by retarding the closure of the inlet valve, the air-to-fuel ratio is reduced because of the amount of air lost. Being the pressure level lower, with LIVC the heat transfer coefficient is reduced. However, the maximum temperature in the engine cycle is increased. While at lower load, the increase in temperature is compensated by a reduction in the heat transfer coefficient, at higher load, the maximum temperature increment is greater and causes a heat transfer losses increment.

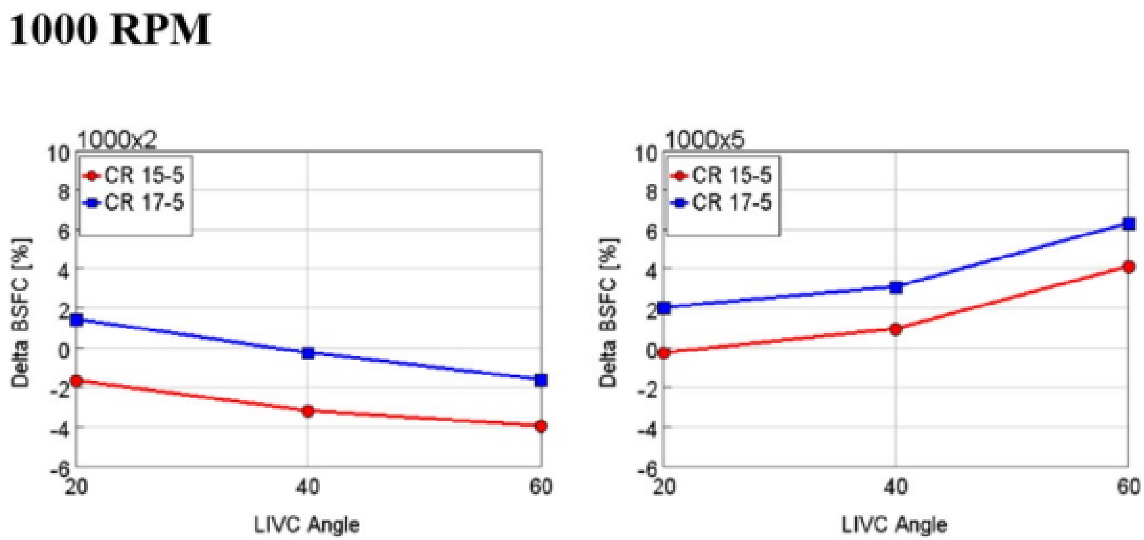

Figure 4.6: Trend of Brake Specific Fuel Consumption at low load (left) and high load (right) with different LIVC angle [6]

In the same publication [6], a transient analysis is also shown in order to highlight the impact of the LIVC technique in terms of BSFC reduction along the whole WLTC cycle. Differently from the steady state analysis, the BSFC benefits in transient analysis were not significant, with a BSFC reduction lower than 1%. The reason is that the actuation of LIVC is limited to a part load zone (BMEP $\leq$ 2 bar in that case), that has a low impact on the total fuel consumption over the driving cycle.

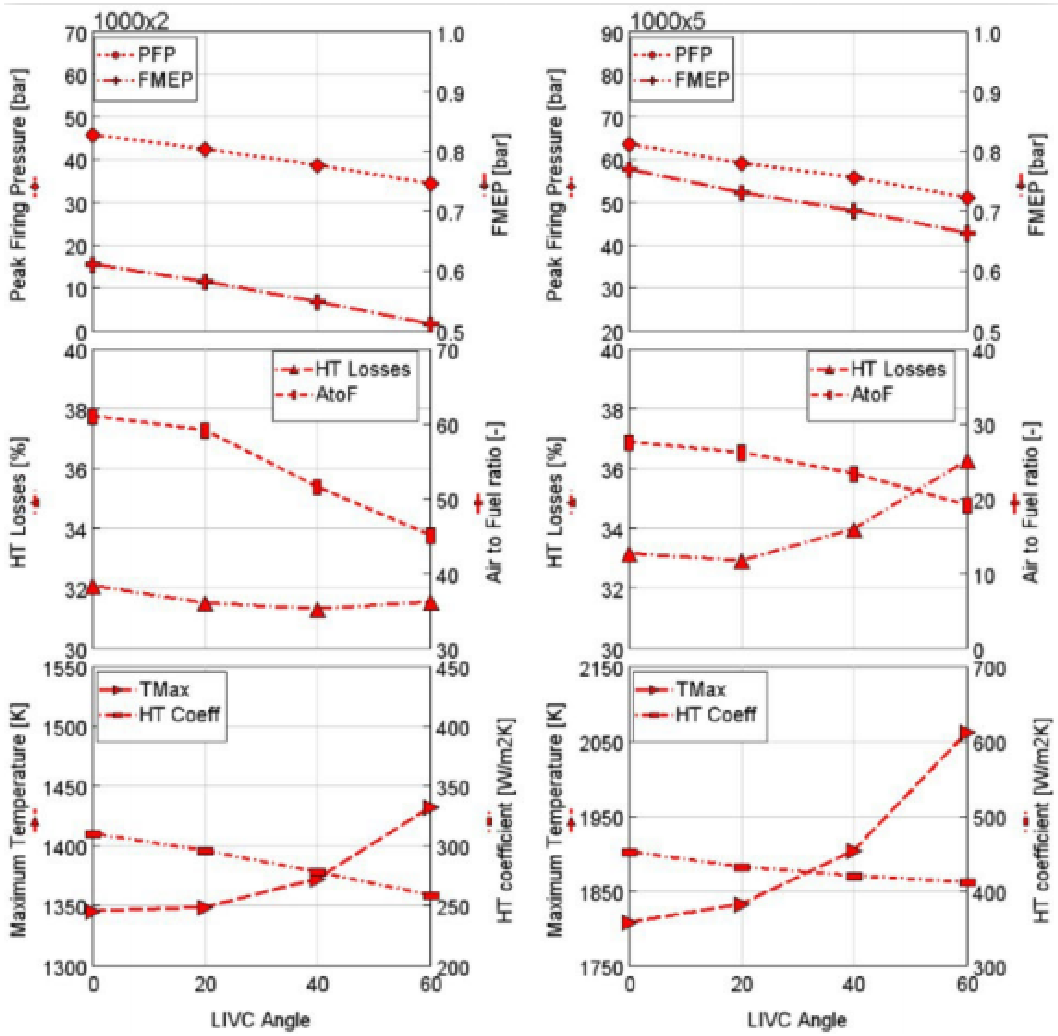

Figure 4.7: Peak firing pressure and friction losses, heat transfer losses and air-to-fuel ratio, maximum temperature and heat transfer coefficient for low load (left) and high load (right) with different LIVC angle [6]

5 CDA technologies

5.1 Introduction to Cylinder Deactivation

Downsizing is a common approach to provide more efficient vehicles with lower emissions and consists of substituting naturally aspirated engines by turbocharged/supercharged engines with a smaller displacement (and/or less cylinders). A further approach to achieve benefits in fuel consumption is Cylinder DeActivation (CDA), that allows the engine to work with a "variable displacement" (or "temporary downsizing"), depending on the operating point. CDA usually refers to deactivating half of the cylinders at low load, by cutting off their fuel supply and, at the same time, keeping the intake and exhaust valves closed.

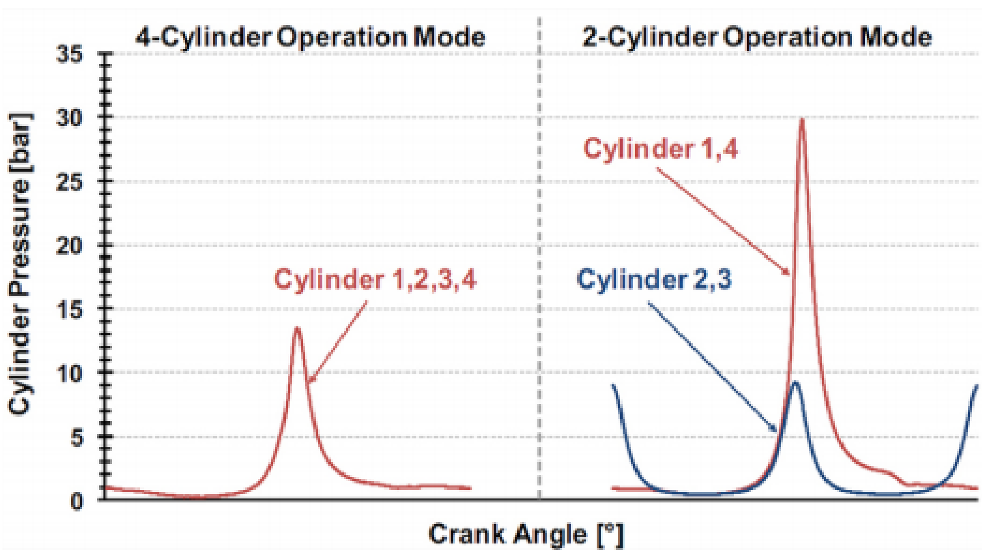

Figure 5.1: Comparison of cylinder pressure between full engine and half engine operation mode [7]

In the deactivated cylinders, the same enclosed air works like a pneumatic spring, that is repeatedly compressed and expanded, thus avoiding overall pumping losses for these cylinders. Also, the exhaust gas after-treatment efficiency is preserved, because if valves were not closed, cool gas would get to the exhaust. The unfired cylinders are dragged and contribute only with low friction losses, due to heat transfer and blow-by.

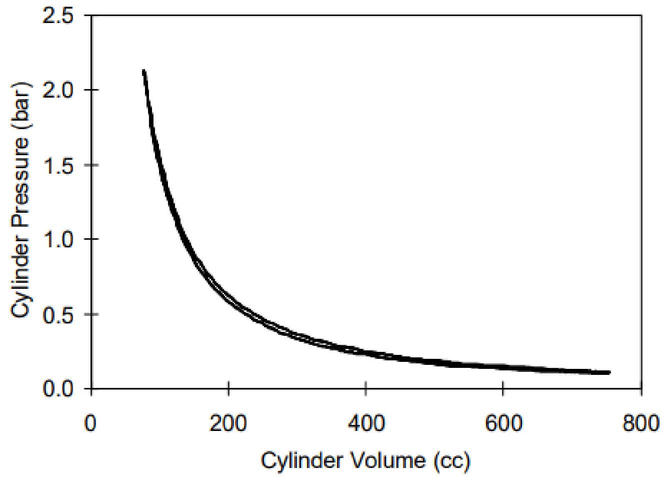

Figure 5.2: Pressure in a deactivated cylinder [8]

In order to keep the same engine torque and to compensate the friction losses of the unfired cylinders, the load of the active cylinders must be higher, resulting in an IMEP (Indicated Mean Effective Pressure, referred to the single cylinder displaced volume) shift. The effective compression ratio of the firing cylinders increases and their thermal efficiency is improved. Then, the operating ranges associated with the lowest specific fuel consumption are approached in cylinder deactivation mode and not when all cylinders are operating.

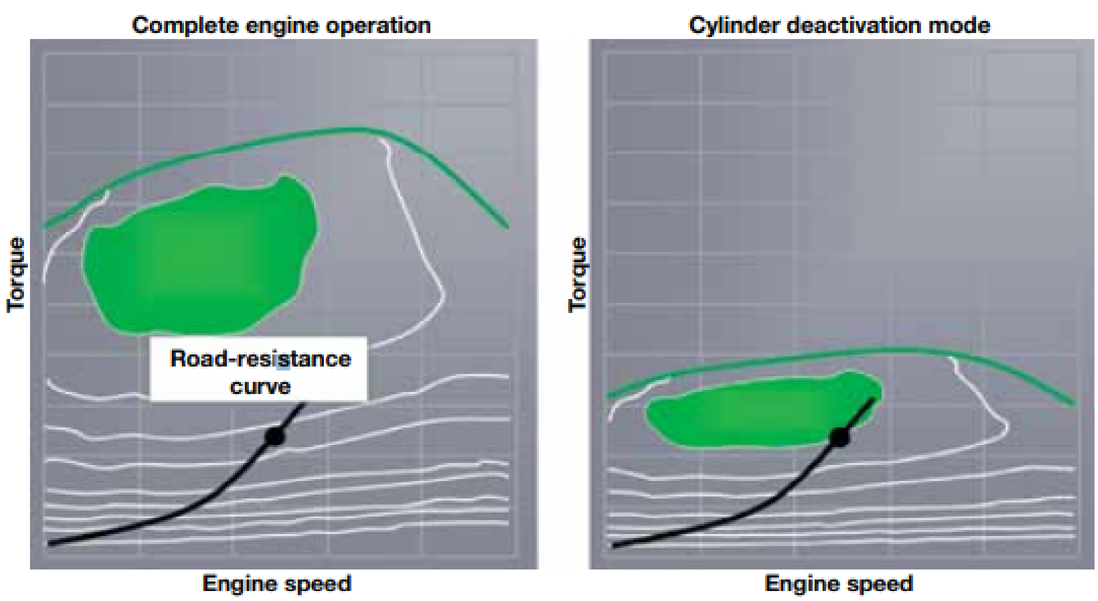

Figure 5.3: Operating data map (iso-BSFC lines) and driving resistance curve: complete engine operation (left) vs CDA mode (right) [9]

NVH (Noise, Vibration, and Harshness) is a fundamental limitation for CDA, as deactivating half the cylinders results in lower frequency and higher amplitude torque pulsations at the crankshaft, which may not be acceptable under all conditions.

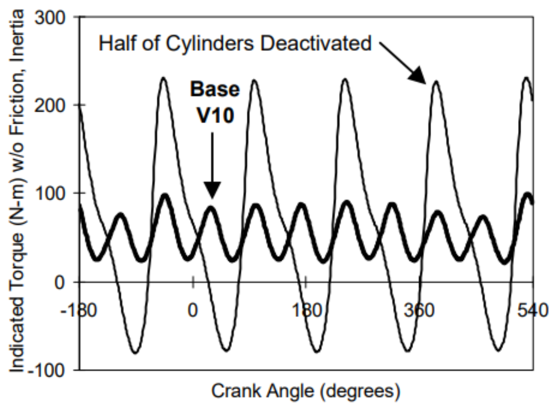

Figure 5.4: Torque pulsations with cylinder deactivation, compared to 4-cylinder mode [8]

5.2 CDA advantages and issues in Gasoline engines

In Spark Ignition (SI) engines, load control is usually achieved through a butterfly throttle valve, that regulates the intake manifold pressure in order to control the amount of air charge in the cylinders. Air-fuel-ratio must be nearly-stoichiometric, for the three-way catalyst to work properly. In this case, in order to supply the required work with only half of the cylinders, each cylinders requires more air and fuel than it would with all cylinders firing. Therefore, the intake manifold pressure must be higher, so the butterfly valve less throttled and the pumping work is reduced [8].

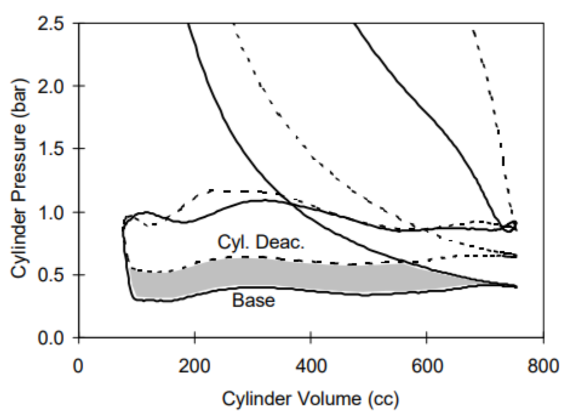

Figure 5.5: Pumping work benefit of CDA for SI engines [8]

"Cylinder Deactivation with Mechanically Fully Variable Valve Train" [7], by Rudolf Flierl and Frederic Lauer, discusses the advantages of CDA for a SI unthrottled engine, where intake charge control is instead achieved by EIVC (Early Intake Valve Closure). In comparison to the base engine which is run in throttled mode, the gas exchange work can already be significantly reduced by utilizing EIVC, with the remaining losses that are due to the flow losses at small valve lifts, required to meet the low engine load. After inlet valve closing, an expansion and re-compression ensues on a similar level, hence does not contribute to the pumping work. The same situation applies also for the 2-cylinder mode; however, the load required for the fired cylinders is more than doubled, and the valve opening duration has to be expanded. This is hence accompanied by a greater peak lift, which results in lower pumping losses through the valves. Then, cylinder deactivation can be used in combination with EIVC, with a, although only partial, combination of benefits.

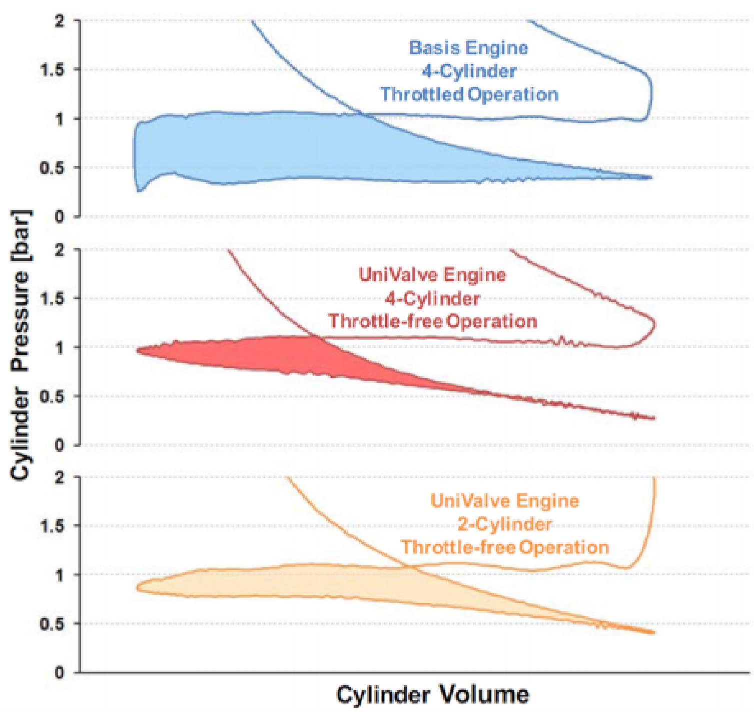

Figure 5.6: Comparison of charge change cycle for normal throttled operation, throttle-free (EIVC) operation and throttle-free CDA operation [8]

The same study compares the different BSFC (Brake Specific Fuel Consumption) between the 2-cylinder and 4-cylinder engine mode, but run with EIVC (throttle-free). CDA generates considerable advantages in fuel consumption for low engine loads (in this case, < 5 bar), with the highest reduction close to engine idle. In the case of SI engines, knock (detonation) has to be avoided and so is a constraint. At high engine loads, the higher cylinder pressure in the fired cylinders tends to provoke knock. Hence, spark advance has to be retarded, and the optimal combustion timing can no longer be obtained. Therefore, at higher engine loads, CDA ends up in a fuel penalty. However, with increasing engine speeds, the limit, at which CDA has zero BSFC benefits, shifts slightly towards higher engine loads, due to the decreasing knock tendency and the higher pumping losses weight, which increase the advantage of cylinder deactivation.

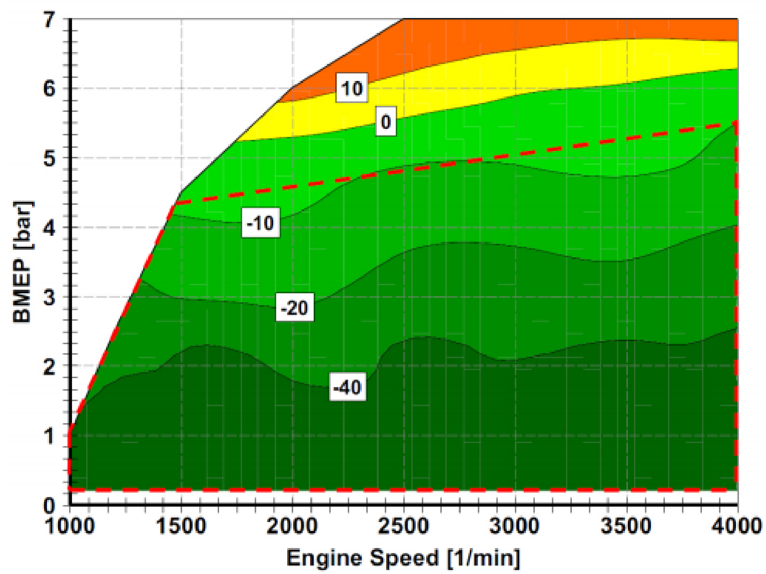

Figure 5.7: Difference of fuel consumption in 2-cylinder-mode to 4-cylinder-mode (both throttle free, SI engine) [8]

When the fuel injection is cut off on the cylinders to be deactivated, the intake valves are still open, thus additional pumping losses are provoked. This increased load demand has to be compensated by the active cylinders. Then, a certain torque reserve is needed to temporarily cover the additional losses during mode change, and this is obtainable only up to certain engine loads, therefore marking an upper limit of the 2-cylinder mode. Boost pressure from turbochargers can not be used during transition because of the system's high inertia. Then, it is advisable to provide hysteresis when applying the cylinder deactivation, to avoid the system changing mode very frequently.
When switching to cylinder deactivation mode, there are two different strategies for keeping a charge in the deactivated cylinders. With the "exhaust trapped" strategy, the exhaust gas is confined in the chamber after the combustion process has been completed; with the "fresh air trapped" strategy, new fresh air is introduced, but it is only possible for direct injection

engines. In both cases, the confined gas acts as a pressure spring. With the former strategy, the heat generated by the confined exhaust gas makes the cylinder cool down more slowly, but peak pressures drop due to cool down as well as blow-by, causing irregularities on the crankshaft. With the latter strategy, the difference in compression between the cylinders are less pronounced, but the air in the combustion chamber loses all tumble or swirling motion produced at the intake point after just a few cycles; therefore, it may still be possible to re-fire the engine in this operating state, but the ignition timing will have to be adjusted. Care must also be taken to ensure that no suction or vacuum effect is produced in the combustion chamber, since this would lead to engine oil being drawn in [9].

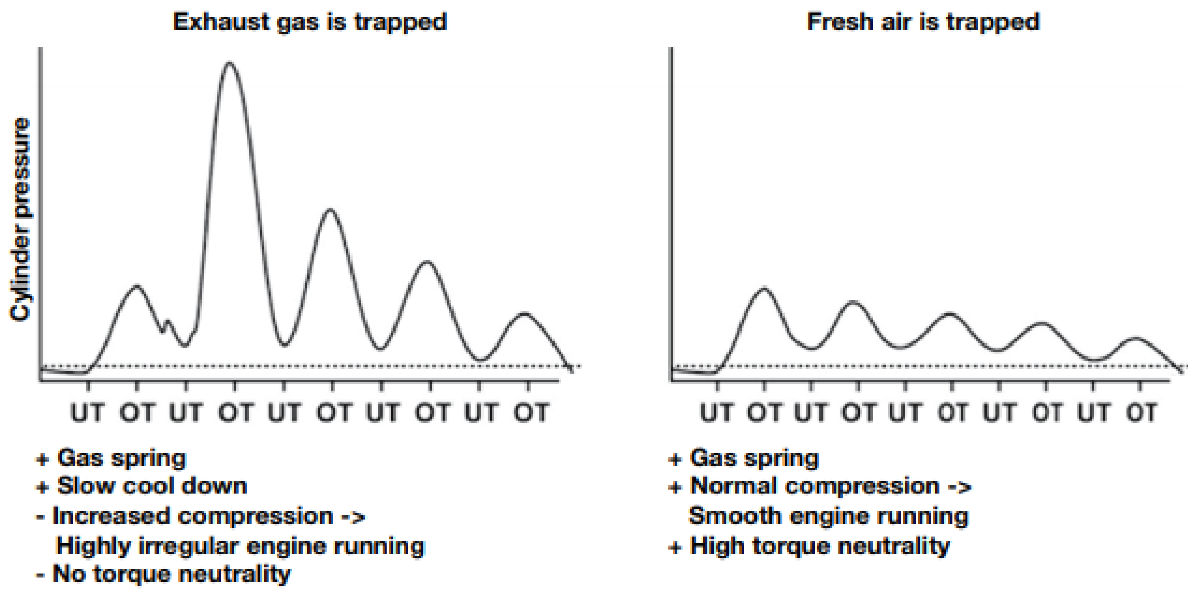

Figure 5.8: Trapping strategies for deactivated cylinders [9]

5.3 CDA advantages and issues in Diesel engines

In Compression Ignition (CI) engines, load control is achieved by acting on the air-to-fuel ratio, which doesn't need to be stoichiometric like in SI engines. Therefore, the butterfly throttle is usually not present, avoiding the respective losses, the reduction of which constitutes a major advantage of CDA in SI engines. Then, CDA potential appears to be less promising for Diesel engines. However, CDA is still capable of obtaining fuel consumption benefits, because of the lower pumping losses and higher thermal efficiency, and knock is not a limitation for this kind of engines. Moreover, CDA is beneficial for the thermal management of the after-treatment system, because the combination of lower air flow, higher firing cylinder temperature, and reduced heat transfer to the cylinder walls is capable of increasing the exhaust gas temperatures.

In "Analytical and Experimental Evaluation of Cylinder Deactivation on a Diesel Engine" [10], the BSFC benefits of CDA are evaluated analytically for a six cylinder diesel engine, identifying changes in pumping work, heat transfer and friction as having the greatest influence on the fuel efficiency results achieved. While the firing cylinders have similar pumping in three-cylinder and six-cylinder mode, the deactivated cylinders have negligible pumping losses, resulting in an overall pumping torque reduction at light loads. In three-cylinder mode, there is a reduction in the in-cylinder heat transfer, because the surface area for combustion is less than with six firing cylinders. For three-cylinder mode there is an overall friction reduction at equivalent brake torque in six-cylinder mode. In fact, FMEP (Friction Mean Effective Pressure) depends on the maximum cylinder pressure (according to Chenn-Flynn model), and the increase in firing cylinder pressure in three-cylinder mode is less than the pressure drop in the deactivated cylinders.

In "Engine strategies to meet Phase-2 greenhouse gas emission legislation for Heavy-Duty Diesel engines" [11], a study on a turbocharged six cylinder diesel engine that can switch to three cylinder operation mode shows how CDA benefits depend on engine speed. The amount of boost pressure possible for a turbocharged engine is mostly dependent on the energy provided by the exhaust mass flow rate. The exhaust flow rate, and thus the air flow rate, increases as the engine speed is increased, but when operation is switched from six to three cylinders, the air flow rate is reduced by half. Also, the non-firing cylinders are still active in a closed configuration and contribute to the overall parasitic losses. Therefore, at low engine speeds, no improvement in BSFC is observed during CDA operation. Instead, at higher engine speeds, the three cylinder operation is still able to produce enough air flow to reach the power requirements, resulting in lower BSFC values with cylinder deactivation, when compared to the baseline configuration.

With the same turbocharger configuration, matched for the volume flow of the normal engine, during CDA operations boost pressure drops because of the lower capacity. Therefore, cylinder charge is decreased, causing lower air-to-fuel ratios than the desired one, with lower fuel benefits than expected. To reach higher boost pressure, a smaller exhaust turbine would be needed, but this variability can be obtained with a Variable Geometry Turbine (VGT) [12].

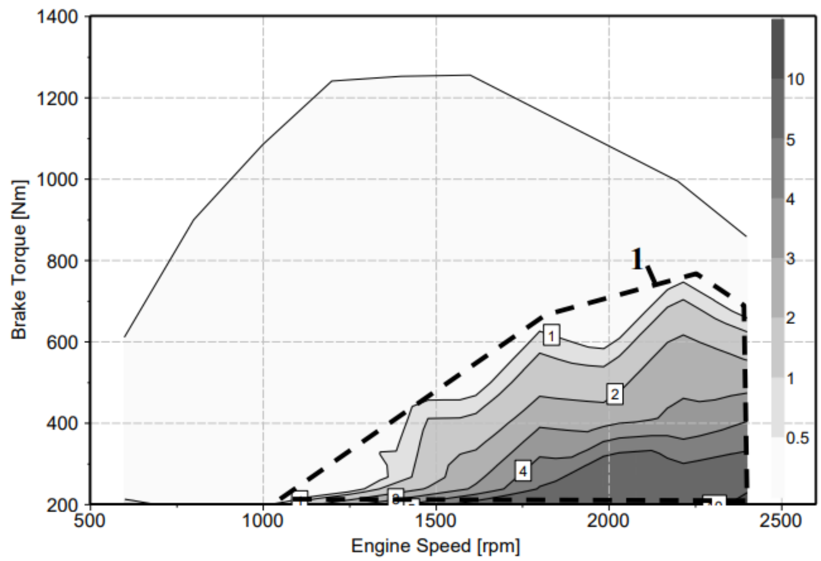

Figure 5.9: BSFC reduction for the engine with cylinder deactivation when compared to baseline: at low engine speeds (< 1200 rpm), no benefits are observed [11]

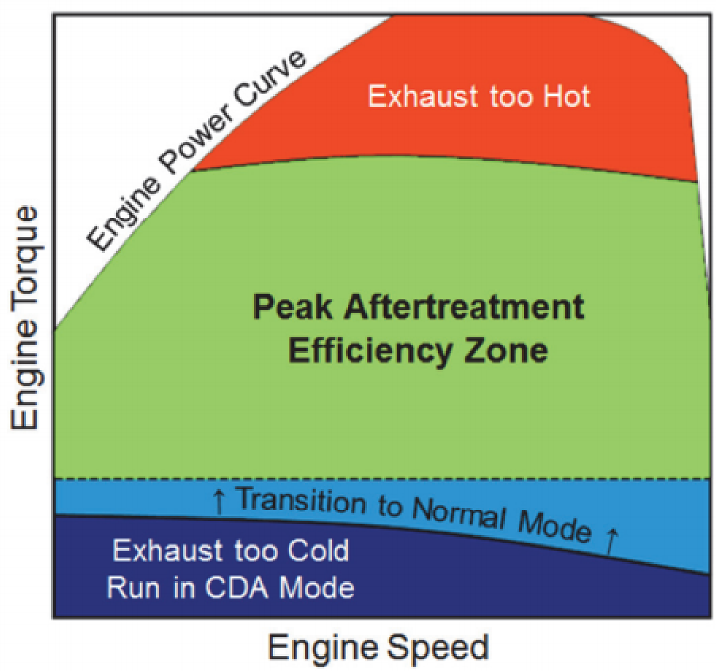

Figure 5.10: Hypothetical turbine out temperature map of medium duty diesel engine, showing areas of load/speed space appropriate for half engine CDA operation [13]

The higher exhaust temperature with CDA can be used for the after-treatment system, but it must be considered that CDA operation gives a reduced exhaust gas mass flow (approximately half than normal mode) and thus the exhaust gas enthalpy can be lower. According to the engine operating point, and the correspondent exhaust temperatures, the NO_x after-treatment system can reach a different efficiency. At low engine load, the exhaust is too cold for having an high efficiency, while at high engine load, the exhaust can be hotter than desired. "Cylinder deactivation improves Diesel after-treatment and fuel economy for commercial vehicles" [13] shows how incorporating half engine CDA (on a six cylinder engine) at low engine loads significantly reduces the area of speed-load space where the exhaust temperature is too cold to bring the engine's after-treatment system into its high efficiency zone.

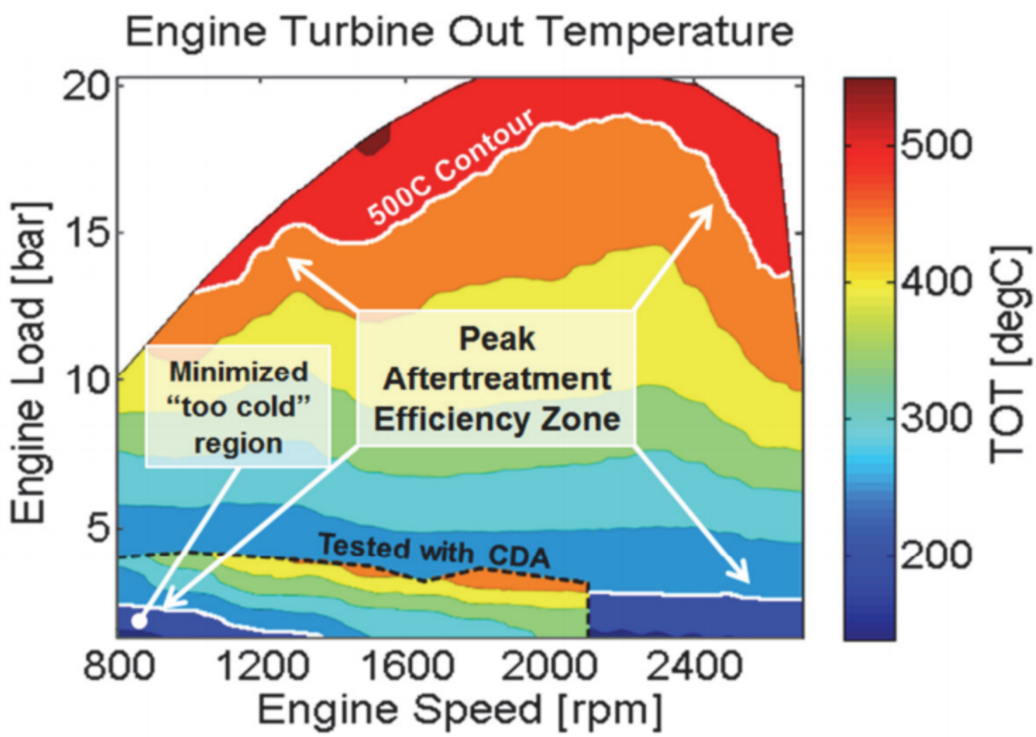

Figure 5.11: Turbine out temperatures function of engine speed and load, with a zone for half engine CDA mode [13]

In "Cylinder Deactivation for Increased Engine Efficiency and Aftertreatment Thermal Management in Diesel Engines" [14], the benefits of deactivating cylinders on reducing the motoring torque are presented. Motoring is defined as engine operation where no fuel is injected in the cylinders, and occurs regularly during decelerations or during vehicle coast. The motoring torque, required to externally run the engine, is shown (figure 5.12) for various degrees of CDA, ranging from 6-cylinder operation to 0-cylinder operation (full cylinder deactivation), and results decreasing as more cylinders are deactivated. Full engine CDA, with no active cylinders, could offer utility as part of a fuel saving strategy during vehicle coast, and has the advantage of requiring zero fuel input, unlike strategies which disconnect the engine from the drive-train but leave the engine idling. Since there is no exhaust flow, heat transfer is also eliminated when all cylinders are deactivated, which can help at maintaining the after-treatment temperatures.

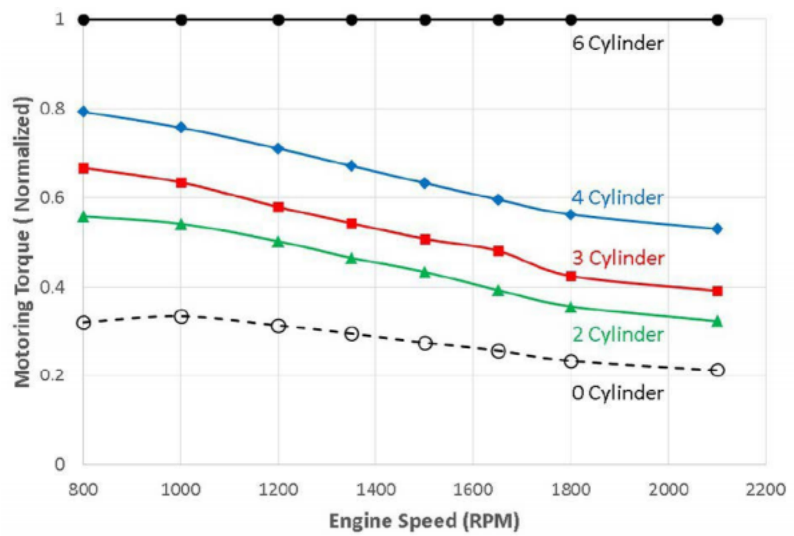

Figure 5.12: Normalized motoring torque required to run the studied engine at speeds from 800 to 2100 RPM [14]

5.4 Dynamic Skip Fire

Conventional Cylinder Deactivation systems generally allow only to switch off half of the engine cylinders, or switch to fixed reduced sets of cylinders, such as eight to four cylinders and six to four to three cylinders, with drawbacks in terms of NVH. Dynamic Skip Fire (DSF®), developed and marketed by *Tula Technology, Inc.*, is an advanced cylinder deactivation control strategy, that considers activation or deactivation of each cylinder event independently. When a decision is made to skip a cylinder, intake and exhaust valve are held closed. Load control can be achieved by varying cylinder firing density (number of firing events out of total opportunities), making fire / no fire decisions at each firing opportunity, depending on engine speed and driver torque demand.

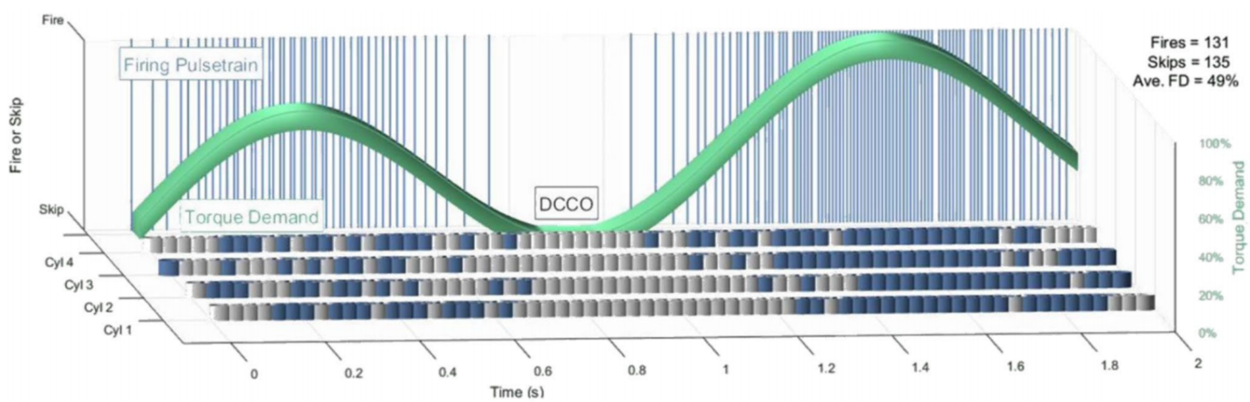

Figure 5.13: Illustration of the DSF approach [15]

Figure 5.13 shows an example of DSF approach in a four-cylinder engine. Given a varying torque request, for each engine cycle blue cylinders are fired and gray cylinders are skipped. The firing pulse train illustrates the overall firing density. At high load operation (100% torque demand), all cylinders fire. At low load operation (near-zero torque demand), 25% or fewer cylinders fire. When torque demand is zero or negative, Deceleration Cylinder Cut-Off (DCCO) phase is entered, with both fuel cut-off and valve deactivation for all cylinders.

Compared to conventional CDA, DSF can improve transition smoothness by avoiding switching sets of cylinders on and off. Moreover, cylinder wall temperature cool-off and other thermal effects can be prevented, by avoiding deactivating individual cylinders for extended periods of time. DSF can also achieve a finer control of the "effective" cylinder displacement. For example, an effective displacement of 1/3 can be obtained by repeating a fire followed by two skips; in a four-cylinder or eight-cylinder engine, this configuration is not possible with conventional CDA, where the sets of deactivated cylinders are fixed.

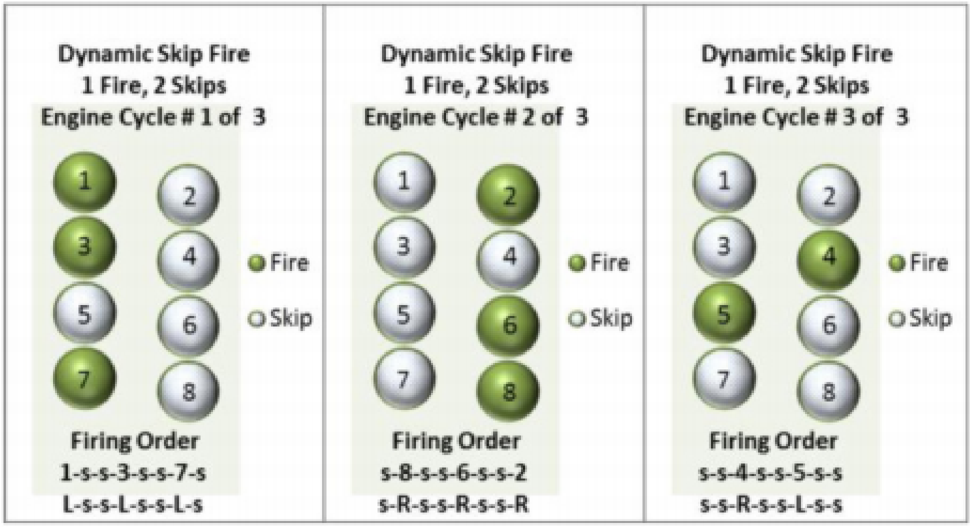

Figure 5.14: Example of firing sequence that fires one cylinder followed by two skipped cylinders (1/3 firing density), for a eight-cylinder engine (firing order: 1-8-7-2-6-5-4-3) [16]

The responsive valve-train deactivation system should be capable of deactivating and reactivating the intake and exhaust valves of each cylinder independently. *Delphi Technologies, Inc.* developed an hardware for this purpose, the "deactivation Roller Finger Follower" (dRFF). The dRFFs switch valve actuation between full and zero lift. Hydraulic pressure, routed through the lash adjuster, moves a spring-loaded pin in the body of the dRFF. When oil pressure is applied while on the cam base circle, it pushes the pin out of engagement, and, with the pin disengaged, the RFF follows the cam profile in lost motion. When the oil pressure is released while on the base cam circle, the spring forces the pin back into engagement and the dRFF then imparts the cam profile onto the valve stem for normal valve operation [17].

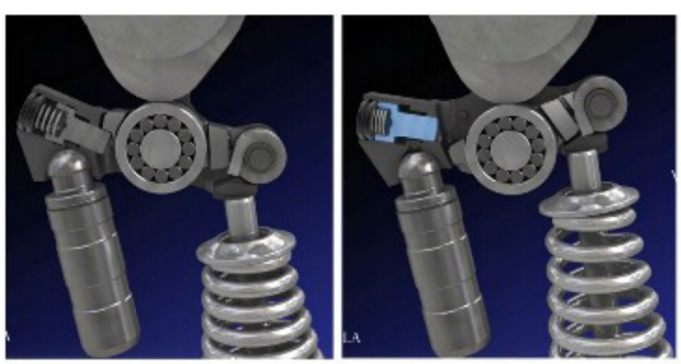

Figure 5.15: Deactivation Roller Finger Follower at peak cam lift: activated mode (left) and deactivated mode (right) [17]

An electrically controlled Deactivation Control Valve (DCV) provides the oil pressure that drives the lost motion pin. Only one DCV for each cylinder is sufficient for controlling all four intake and exhaust valves, because intake and exhaust RFFs are on the cam base circle at different times. The DCVs are similar to production two step oil control valves, but have faster response time and higher durability, needed because the number of deactivation cycles in DSF is much higher than in conventional cylinder deactivation [17].

Figure 5.16: Deactivation Control Valve (left) and Deactivation Roller Finger Follower (right) [17]

DSF achieves fuel consumption benefits, by reduction of thermal efficiency losses, primarily pumping losses and combustion inefficiency. In fact, in throttled engines, the highest thermal efficiency spot is at high loads, due to minimized pumping loads. Due to the cylinders operating at higher loads with DSF, the engine combustion system can also be potentially optimized to match this operating area. Moreover, inactive cylinders are prevented from pumping air through the engine, reducing oxygen saturation of catalysts during deceleration fuel cut events.

The full dynamic control of firings and non-firing of engine cylinders gives wide authority over generation of vibrational and acoustic excitation, because the excitation spectra are additionally controlled by the number and sequence of firing cylinders. As a result, NVH can then be dealt with algorithmically, in a flexible and systematic way [18], avoiding known resonance patterns within the engine. Moreover, the cylinders being deactivated are not always restricted to the same set, but deactivation is uniformly distributed among the cylinders, solving problems of differential wear.

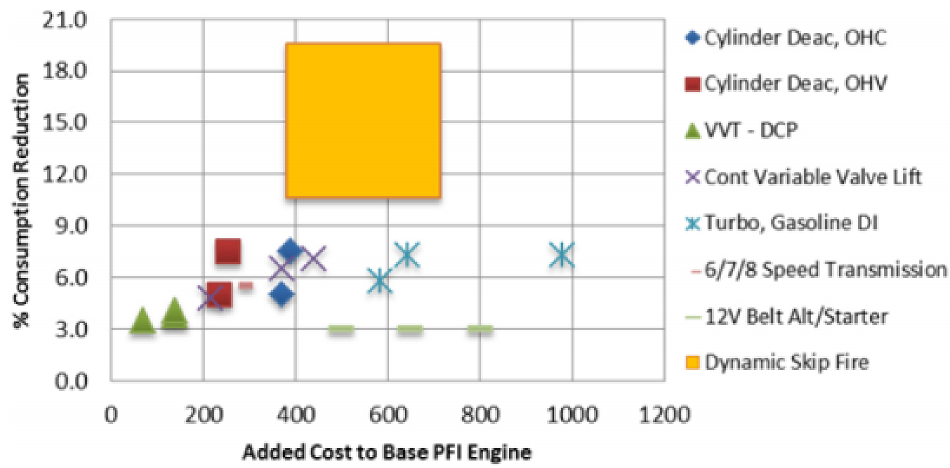

Figure 5.17: Comparison of DSF fuel economy gains and costs with competing technologies (spark-ignition engines) [18]

The fuel economy benefit potential of DSF in Diesel engines, as for CDA, is lower because of the load control not being achieved by a throttle valve. However, diesel DSF (dDSF) [19] still proved capable of obtaining a fuel consumption reduction up to 5%, while realizing tailpipe NO_x emissions reduction up to 40%, by improved conversion efficiency in the NO_x after-treatment system, due to the increased exhaust temperatures.

During DSF operation, torque pulses have greater magnitude for fired cylinders and are more spaced from each other in time than in all-cylinder operation. For this reason, the torque excitations on the drive-train have more low frequency content. In applications with power-train electrification, electrified DSF (eDSF) [17] is a concept for countering the low frequency excitation and achieving torque smoothing, by using a Motor-Generator Unit (MGU). The large, wider-spaced torque pulses from combustion are partially stored by a MGU generating (negative) torque pulse, and stored in a battery or capacitor. During skipped cylinder events, this energy is released from the MGU and reapplied on the powertrain, in the form of a positive torque pulse. However, the electric torque pulses only partially reproduce the missing combustion ones, due to MGU torque and power limits. With this method, the lower frequency components of the resultant combined torque waveform are reduced, and then the torque excitation spectral components are shifted to higher frequencies, where vehicle attenuation is generally better. As a result, DSF operating regions can be extended, allowing skip-fire sequences that normally would exceed the NVH targets.

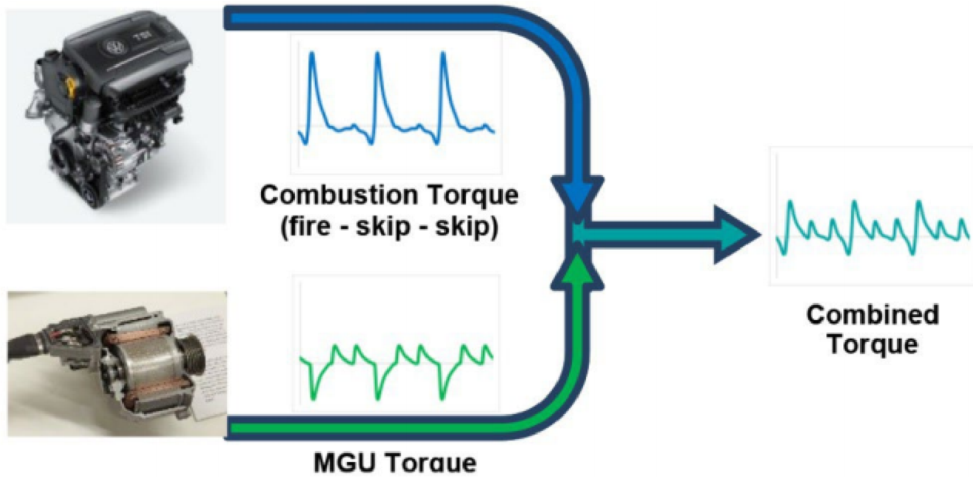

Figure 5.18: Torque smoothing principle, using eDSF [17]

6 Modelling of engine physics and control strategies

6.1 GT Power 1D engine model

A Fast Running Model (FRM) has been used, in order to simulate the 1D physics of the engine. This model has been developed and calibrated by FEV with the commercial software GT Power.

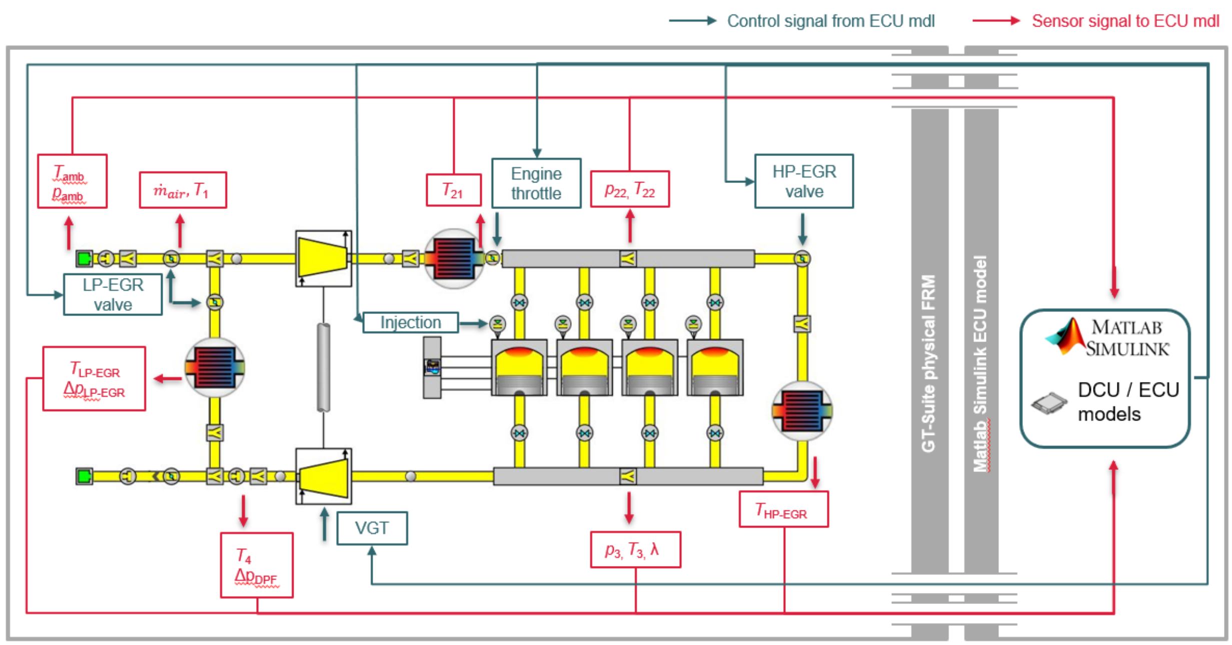

Figure 6.1: Functional scheme of the engine model

The Diesel engine works with four cylinders, having firing order 1-3-4-2, modelled with DI Pulse, a predictive combustion and emissions model for direct-injection compression-ignition engines, available in GT Power; this combustion model is capable of handling multi-pulse injections, but for simplicity the simulations will use only a single injection.

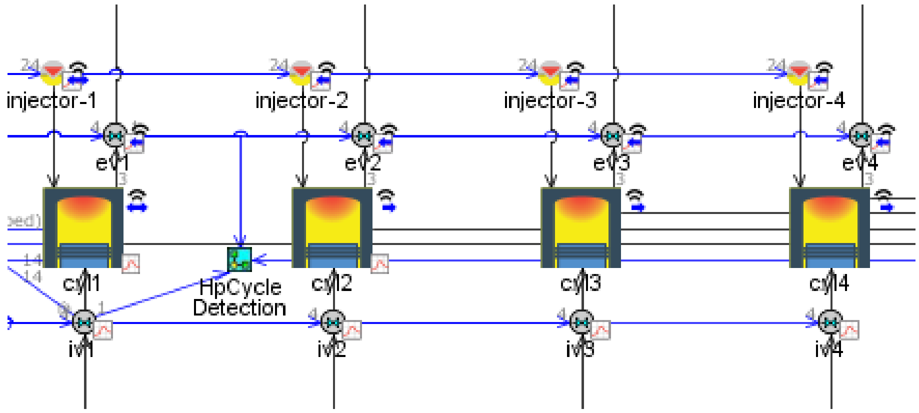

Figure 6.2: Four cylinders, connected each one to the respective injector, intake valve and exhaust valve

The intake air boosting is given by a Variable Geometry Turbocharger (VGT), which means that the rack position of the turbine can be adjusted in order to reach the desired boost level. The turbine map is obtained by interpolating, according to the rack position, different maps defined at discrete rack positions.

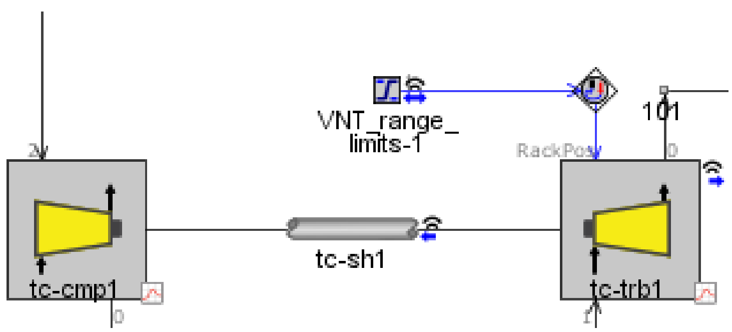

Figure 6.3: VGT: compressor and turbine connected by a shaft

The exhaust gas recirculation is performed through High Pressure and Low Pressure EGR valves. With High Pressure EGR, exhaust gases are recirculated in a short route, from the exhaust manifold to the intake manifold. With Low Pressure EGR, exhaust gases are recirculated in a long route, from the outlet (after the turbine and the after-treatment) to the inlet (before the compressor). The amount of EGR is regulated by the throttle angle of the two valves. The recirculated gases are cooled through dedicated heat exchangers. Other elements of the engine model are the Charge Air Cooler, modelled as an heat exchanger before the intake manifold, and the exhaust pressure losses given by the DPF (Diesel Particulate Filter) and the muffler.

6.2 Simulink control model and co-simulation

When simulating a cycle, the engine has an actuated engine speed and a target brake torque. A Simulink model of the ECU (Engine Control Unit) has the aim of selecting the proper engine parameters in order to reach the target torque and comply with mapped emission limits. In fact, the ECU model is a 0D Mean Value Model of the engine, that uses a map-based approach.

The co-simulation loop links the GT engine model and the Simulink ECU model. While GT works with a varying time step, the Simulink model works with a fixed time step at 100 Hz. The ECU passes to the engine these parameters:

- fuel mass to be injected in a single cylinder;
- VGT position;
- HP EGR valve position;
- LP EGR valve position;
- injection rail pressure;
- angle of SOI (start of injection).

A series of sensors passes from GT to Simulink quantities like the actual brake torque, temperatures and pressures (downstream compressor, upstream turbine, downstream turbine, upstream DPF, downstream DPF, ...), O_2 and NO_x concentration of exhaust gases, and feedbacks of boost and EGR.

In addition, a Simulink model for emissions calculation is coupled with the GT model. This has been preferred for NO_x emissions computation in respect to the model integrated in GT Power. The emissions model was calibrated for the engine and receives as inputs the engine operating point (engine speed n, indicated mean effective pressure IMEP), the mass flows (exhaust gas, hp egr, lp egr), oxygen concentrations before and after combustion, and other parameters. As an output, the NO_x emission is given, as a mass flow, a concentration, or a cumulative mass.

From the GT Power / Simulink co-simulations, the engine-out traces, emissions in particular, can be obtained. For the purpose of obtaining the tail-pipe quantities, a 0D vehicle and engine model, developed by FEV, has been used and in this case configured for simulating the exhaust line only. The after-treatment layout, typical for Euro 6-D passenger car application, contains in order: DOC, SDPF, and passive SCR. The LP EGR exchange line is located between SDPF and SCR.

6.3 Fuel injection loop closure

The ECU Simulink model has complete engine controls incorporated, making it requiring proper calibration. However, the model was still in development and did not have a closed-loop control of the fuel injection, in order to match the actual torque with the desired one. This could be an issue especially when simulating the VVA heat-up strategies or Cylinder Deactivation, using the baseline calibration.
The ECU model, computes the fuel to be injected, and also the other engine parameters, starting from a desired IMEP (Indicated Mean Effective Pressure):

$$IMEP = BMEP + PMEP + FMEP$$

Where BMEP (Brake Mean Effective Pressure) is calculated from the desired torque, depending also on the number of active cylinders, FMEP (Friction losses) and PMEP (Pumping work) are estimated through calibrated maps according to the operating conditions (BMEP, engine speed, ambient corrections, ...).
In order to achieve the loop closure for matching the actual engine torque with the desired one, as a first approach, a PID (Proportional, Integrative, Derivative) Controller was put in the injected fuel calculation. Then, assuming that the torque error was due to a not suitable calibration of FMEP/PMEP maps, it has been decided to move the PID Controller at the IMEP level.

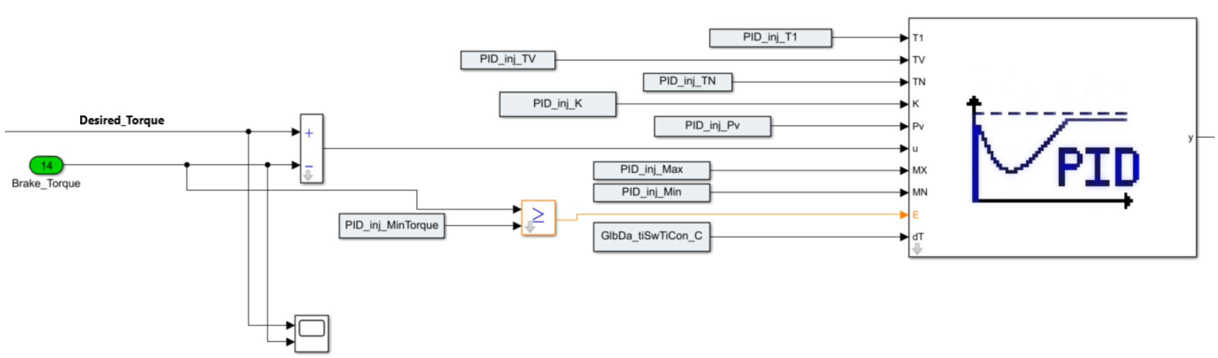

Figure 6.4: PID controller for torque matching

The PID Controller receives in input the error between the desired torque and the actual brake torque. The value, the integral and the derivative of the error are weighted (by factors Kp, Ki, Kd) and summed to obtain the output OUT of the controller. In this case, the PID governor works in a discrete time domain (dT, Simulink simulation time step) and contains also saturation and anti wind-up functionalities. The pre-control IMEP is then corrected with the PID output:

$$IMEP_{corrected} = IMEP \cdot (1 + OUT)$$

The decision to not put the PID controller directly on the injected amount of fuel has been taken also because several engine parameters, including the air path ones, are directly dependent on the cylinder IMEP. Then, if only the fuel injection was corrected, not only the EGR, boost, rail pressure would be chosen starting from a "wrong" IMEP, but the PID would also bypass the smoke limitation, resulting in higher soot than permitted from the engine calibration.

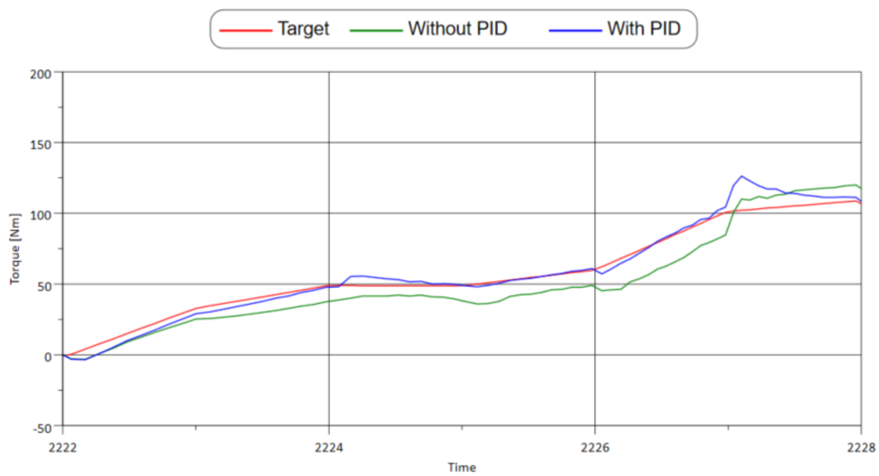

Figure 6.5: Engine torque comparison of a part of simulation (in particular, iEGR) without PID and with PID (closed-loop)

The several testings made with the modified model showed how the PID controller is capable of improving the torque matching, which however isn't still 100% accurate, because of transient phenomena and needed limitations such as the smoke one.

6.4 Implementing of VVA strategies

Each cylinder of the case engine has two intake valves and two exhaust valves, but in GT Power each couple has been modeled as a single equivalent valve. The lift profiles of the two valves are imposed as a function of the local crank angle of the corresponding cylinder.

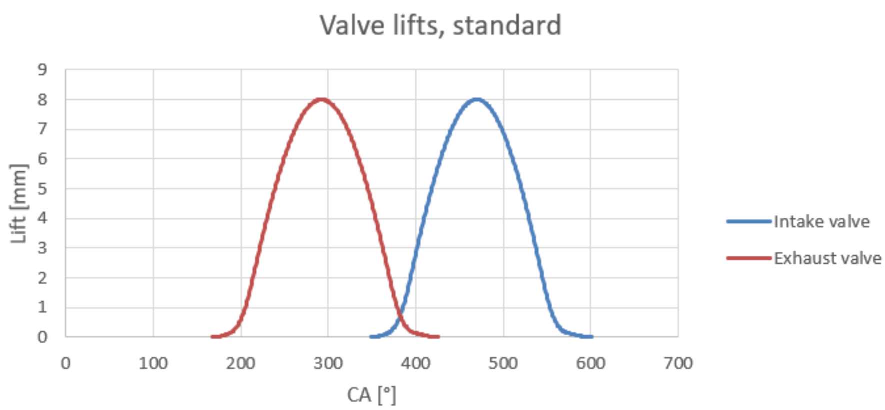

Figure 6.6: Standard lift profiles of intake and exhaust valves

The first VVA strategy to be implemented is Internal EGR, through Exhaust Valve reOpening (EVrO). When iEGR is active, the exhaust lift profile has an additional portion, where the exhaust valve is opened again during the intake stroke, but with lower maximum lift ($<$ 2mm) and a duration of 113° CA (crank-angle).

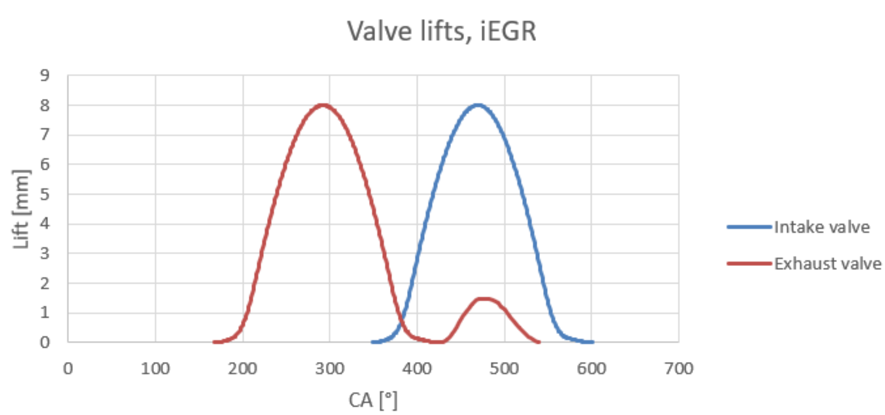

Figure 6.7: Internal EGR lift profiles of intake and exhaust valves

iEGR is activated only when the temperature of DPF mass flow is below 250 °C, with an hysteresis of 25 °C (for example, when iEGR is not active, it is switched on only when

$T \leq 225$ °C, and stays activated until $T = 250$ °C), and BMEP (break mean effective pressure) is below 4 bar. Also, the re-opening is done only on one of the two exhaust valves; having in the model only one equivalent exhaust valve, the implementation of iEGR was more complicated and manually required to halve the mass flow through the valve, during the re-opening lift, by tracking the local crank angle of each cylinder and check if it is within the re-opening angles range.

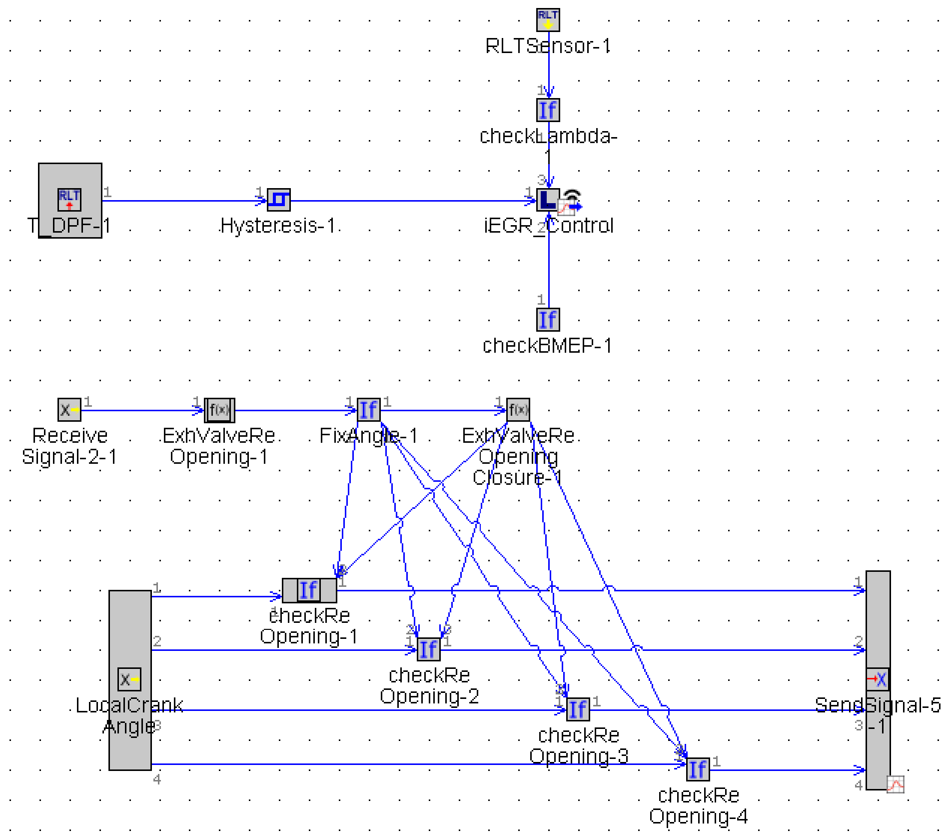

Figure 6.8: Implementation of iEGR control in GT Power: on the top, it is decided whether iEGR should be active or not; on the bottom, the module responsible of halving the mass flow through exhaust valve during re-opening, which is equivalent to open only one of the two real valves

As for LIVC (Late Intake Valve Closure), the implementation was simpler because it is performed on both the intake valves. Different lift profiles for the intake valve were generated, with closure delay (° CA) and secondary lift (mm) as parameters. Also in this case, LIVC has been decided to be activated only when BMEP $<$ 4 bar.

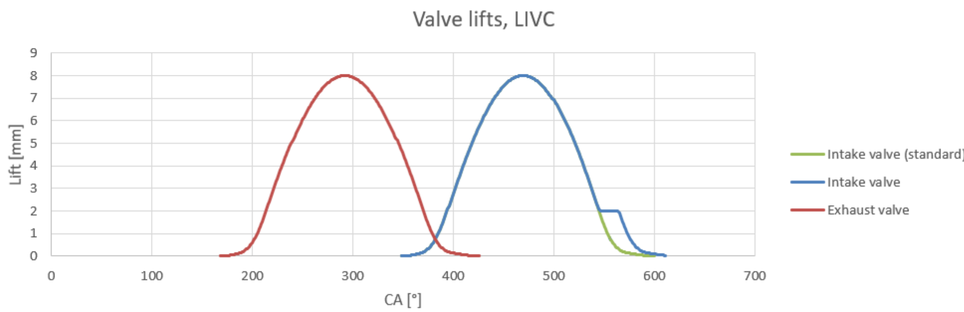

Figure 6.9: LIVC (2 mm, 20° delay) lift profiles of intake and exhaust valves

6.5 Implementing of cylinder deactivation

When a cylinder is deactivated, not only no fuel has to be injected, but also intake and exhaust valves should be closed.

In a single engine cycle, firing density FD is defined as the number of active cylinders, divided by the total. For FD = 1, 4 out of 4 cylinders are active; for FD = 0.75, 3 out of 4; for FD = 0.5, 2 out of 4; for FD = 0.25, 1 out of 4. For each engine cycle (or time instant), depending on the engine operating point and other conditions, the firing density FD is pre-calculated and put as an input of both the GT Engine and the Simulink ECU models. Also, an engine cycle (or time) signal for each cylinder establishes if it is active or not, in the GT Engine model.

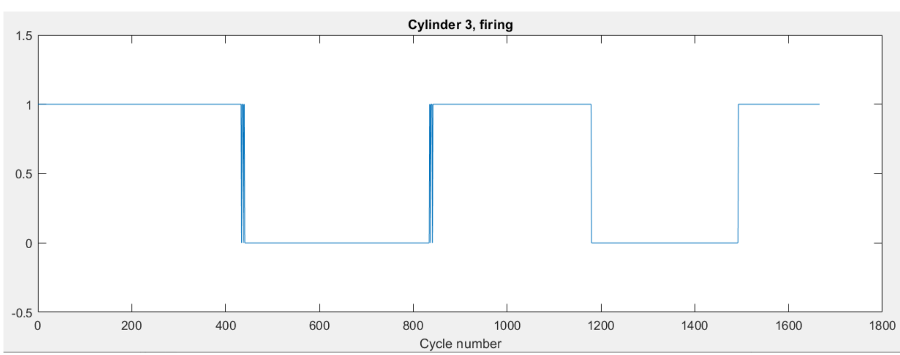

Figure 6.10: Example of Cylinder Firing signal: 1 = firing; 0 = not firing

The cylinder deactivation (CDA) module made in GT Power, depending on engine cycle number or time, checks if each cylinder should be active or not, setting to zero the injected fuel mass and the intake/exhaust valve openings if it is not. Attention was made to correctly synchronize the deactivation with the engine cycle, which starts from each cylinder 1 intake;

in order to do so, the local crank-angle of each cylinder is tracked, holding the signals in the window of intake-injection-exhaust for each cylinder.

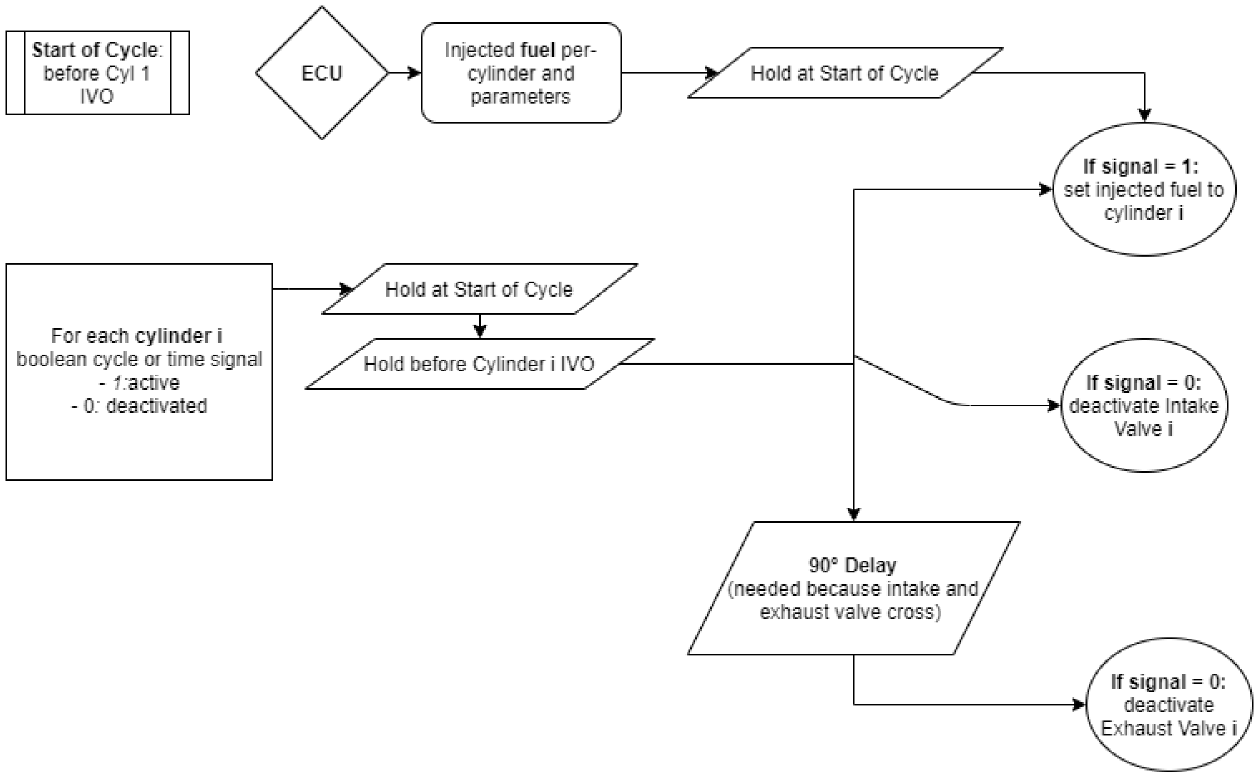

Figure 6.11: Cylinder Deactivation implementation scheme

As for the ECU Simulink model, at the start of this thesis work, the original model had a different approach for handling the Cylinder Deactivation. In fact, inside the Simulink ECU model, the firing density was only used for correcting the IMEP value at the start of the air path controls, where the NO_x set-point is calculated; for example, in two-cylinders mode, the entry point of the NO_x SP map was twice the computed IMEP. All the rest of the control part didn't take into account the different number of active cylinders.
Then, as a new control approach, considering that the Simulink ECU is a Mean Value Model and quantities are computed as averages for a single cylinder, it was decided to use the original calibration, but adapt it with a dynamic number of cylinders. So, wherever the controls makes calculations using the number of cylinders, this is set to a number depending on the firing density ($N_{cyl} = 4/FD$). This happens for example in the BMEP calculation, which starts from the desired torque. Also, the PID Controller, realized for the loop closure, has been key for obtaining good results in terms of torque matching during CDA phases.

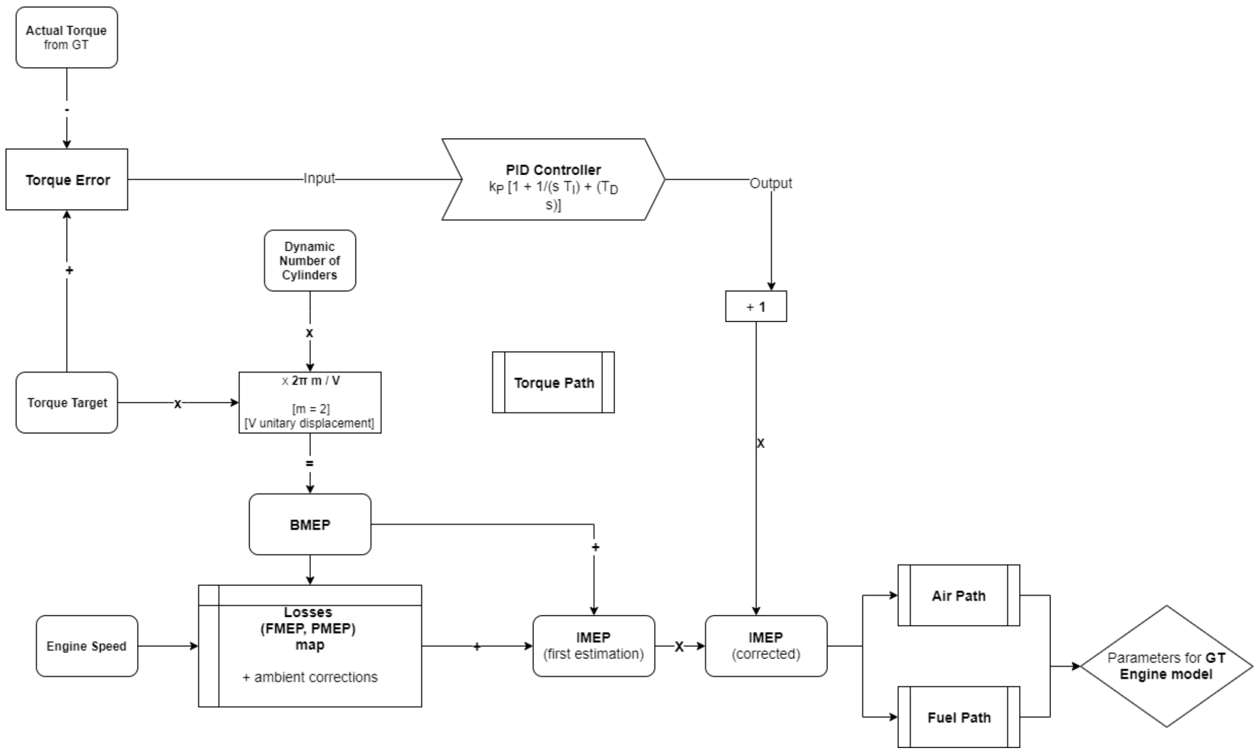

Figure 6.12: Desired BMEP calculation and PID controller: $BMEP = \frac{2\pi \cdot m \cdot Torque}{n_{cyls} \cdot V}$, with m = 2 number of strokes in a cycle, V unitary displacement, n_{cyls} dynamic number of cylinders

The problem of the starting model was that all the calculations, besides the NO_x set-point, worked with an IMEP that was the equivalent for a four-cylinder engine and several engine parameters didn't take into account of that in CDA mode cylinders should work with higher IMEP. The new model approach, using a dynamic number of cylinders, is more physically correct and proved to be more robust.

For example, a comparison between the old model and the new model results for a constant torque (32 Nm) and speed (2000 RPM) maneuver are shown. In this case, it is the firing density to change, from four active cylinders to two active cylinders and vice versa. With the old model, for instance, the injection parameters (rail pressure and start of injection angle) had almost the same value for the two firing densities (1 and 0.5). With the new model, that takes into account the higher IMEP needed for a single cylinder, more optimal parameters are selected, giving a lower BSFC (Brake Specific Fuel Consumption) during CDA mode.

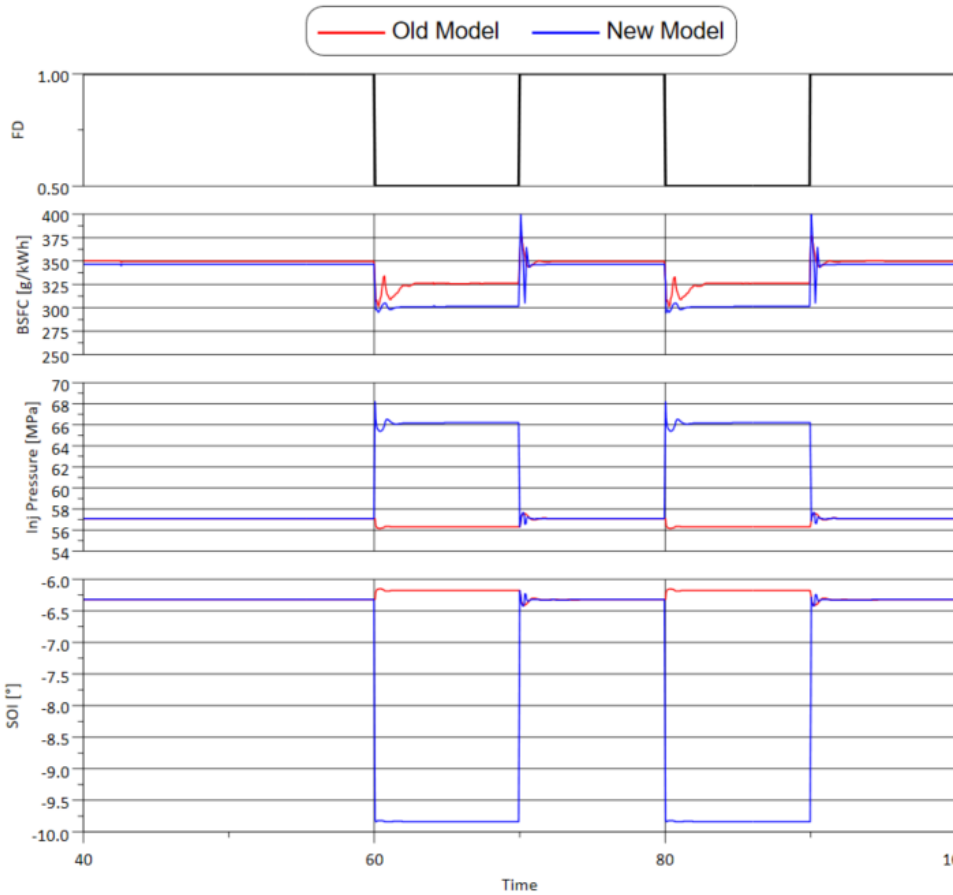

Figure 6.13: Comparison between old and new (dynamic number of cylinders) control models: Firing Density, BSFC, Injection Pressure, Start of Injection

7 Simulations and analysis

7.1 Case study and baseline simulation

The object of this study is a 2-liter turbocharged Diesel engine, for light-duty applications, having four inline cylinders, equipped with Variable Geometry Turbocharger (VGT), High Pressure (HP) and Low Pressure (LP) EGR, DOC, SDPF and passive SCR. The transient simulations focused on the urban phase of a TFL driving cycle, that has been chosen in order to evaluate the part load benefits of different strategies, in terms of fuel consumption, NO_x emissions, and warm-up.

After each GT/Simulink engine simulation, before passing the outputs to the after-treatment model, a post processing of the results is performed in order to implement the Cut-Off and Start & Stop features. Both strategies reduce fuel consumption and emissions. Fuel cut-off consists of interrupting the fuel injection when the accelerator pedal is released, in order to allow a fluid deceleration of the vehicle. The start-stop system automatically shuts down and restarts the engine to reduce the amount of time the engine spends idling.

For the purpose of this thesis, the same engine calibration and ECU logic has been used for all the strategies. As a reference, the baseline simulation, with no heat-up strategies, has been performed. All the simulations have in common the engine speed, which is actuated, and the target brake torque, with the operating points obtained through a previous vehicle simulation of the TFL cycle phase considered.

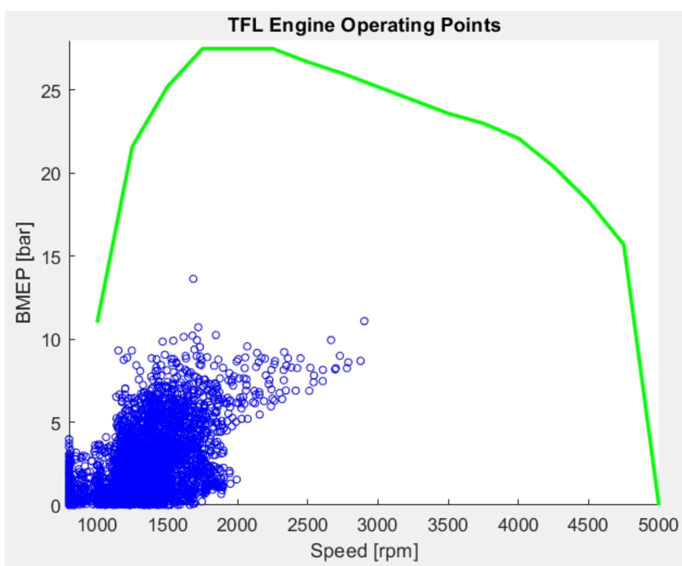

Figure 7.1: Engine Operating Points (EOP) of the cycle in the BMEP [bar] - RPM plane

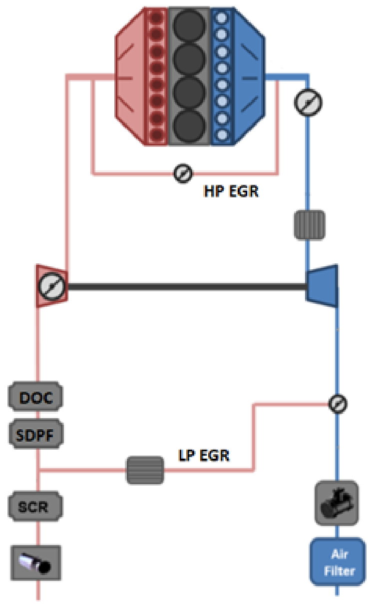

Figure 7.2: Engine scheme

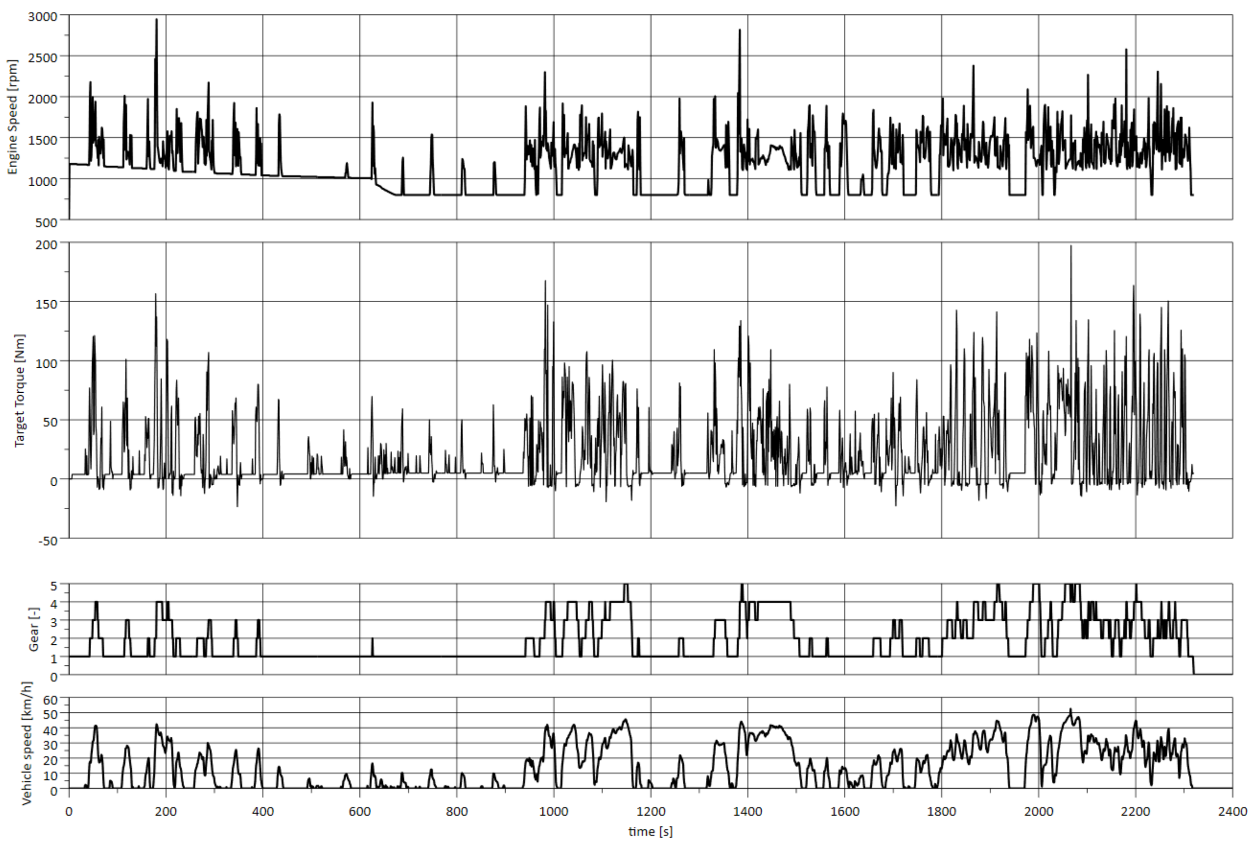

Figure 7.3: Engine speed, target torque, gear profile, and vehicle speed

7.2 iEGR simulation

Internal EGR is achieved by Exhaust Valve reOpening, activated when BMEP < 4 bar, and $T_{DPF} < 250\ ^{\circ}C$ (with $25\ ^{\circ}C$ hysteresis). The valve profiles have been shown in the modelling chapter (figure 6.7).

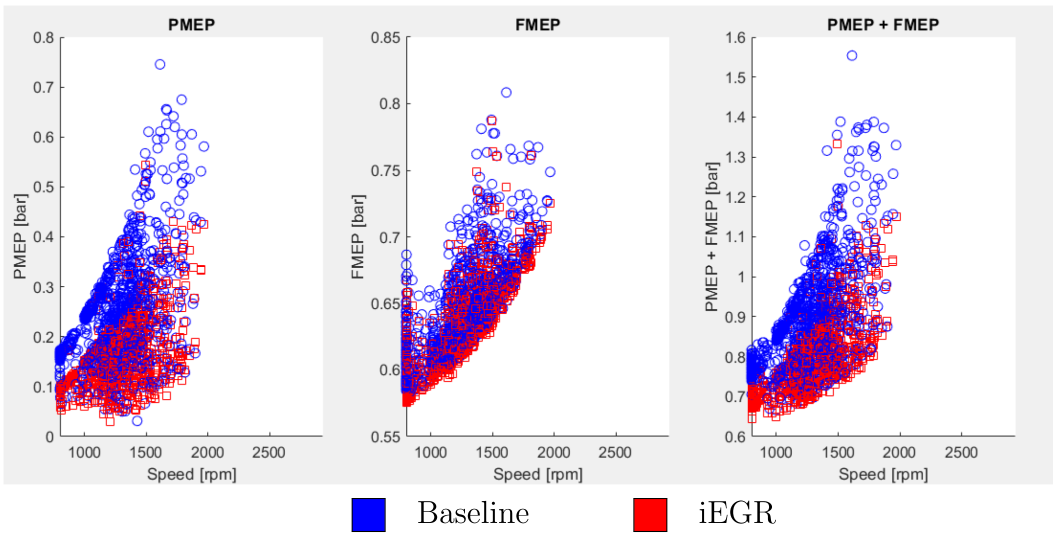

Figure 7.4: PMEP and FMEP corresponding to points where iEGR is activated (red), compared to the corresponding values for baseline (blue)

iEGR strategy showed fuel benefits in respect to the baseline simulation, and this is due to the lower losses, in particular pumping. The lower pumping work can be explained due to the ECU trying to achieve an in-cylinder O_2 concentration target, which, during iEGR phases, results to requesting a lower external EGR from HP and LP valves, because a fraction of exhaust gas recirculation already occurs "internally".

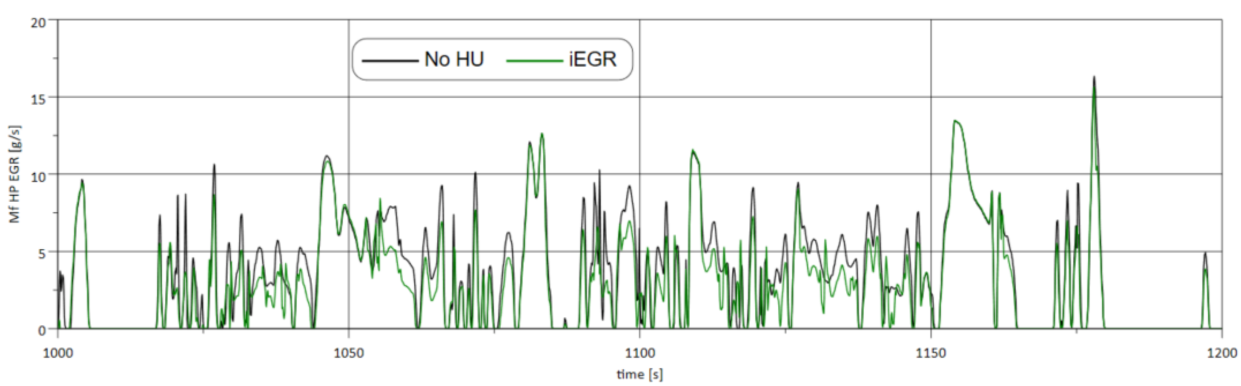

Figure 7.5: Zoom of HP EGR mass flow during iEGR simulation, compared to baseline

7.3 LIVC simulation

LIVC is activated when BMEP < 4 bar, in order to enable the Atkinson cycle, delaying the intake valve closure of 15° CA, with secondary lift of 2 mm. In the case of LIVC, as expected, the low load engine operating points, where Atkinson cycle was enabled, show lower friction, with consequent benefits on fuel consumption.

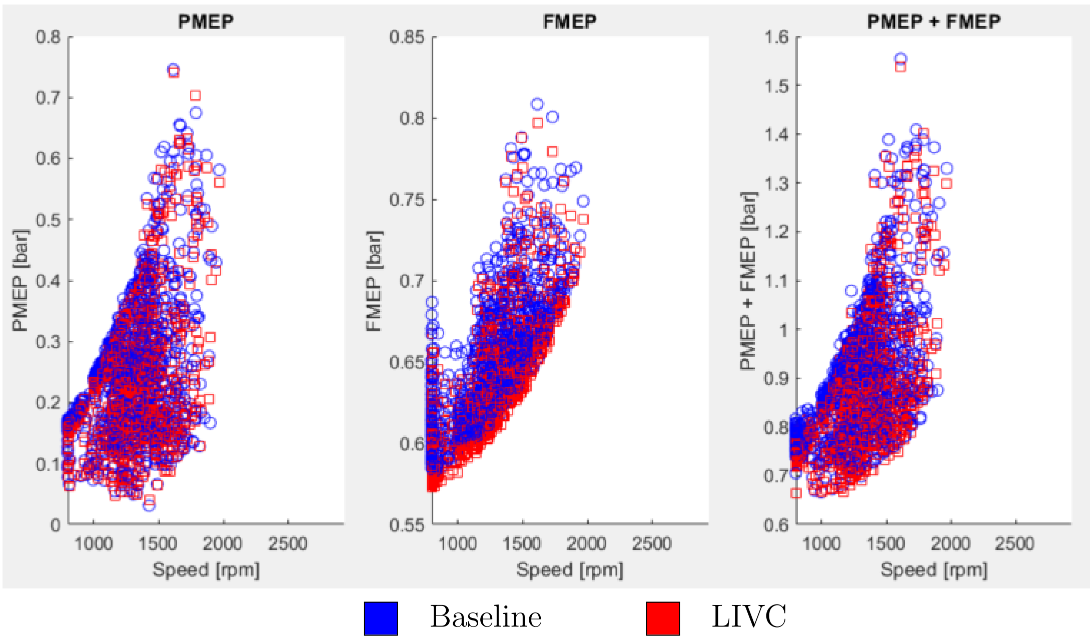

Figure 7.6: PMEP and FMEP corresponding to points where LIVC is activated (red), compared to the corresponding values for baseline (blue)

A DoE (Design of Experiments) study has been performed to establish the optimal secondary lift in terms of fuel consumption; in particular, 1.5 mm, 2 mm and 3 mm secondary lifts, with the same closure delay, have been simulated over the entire cycle. Closure delays higher than 15° have been tried, but the original engine calibration didn't prove to be suitable for this purpose, and misfire points were detected in the cycle. Better results in terms of fuel savings and warm-up could possibly be obtained by simulating a more aggressive LIVC, but adapting the calibration when this strategy is applied, for example by increasing the boost target and reducing the requested EGR.

7.4 CDA simulation

When BMEP < 3.5 bar, two cylinders out of four are deactivated, obtaining in each engine cycle: cyl1 fire, cyl3 skip, cyl4 fire, cyl2 skip (firing density FD = 0.5). So, the deactivated cylinders are always the same (cylinders 3 and 2).

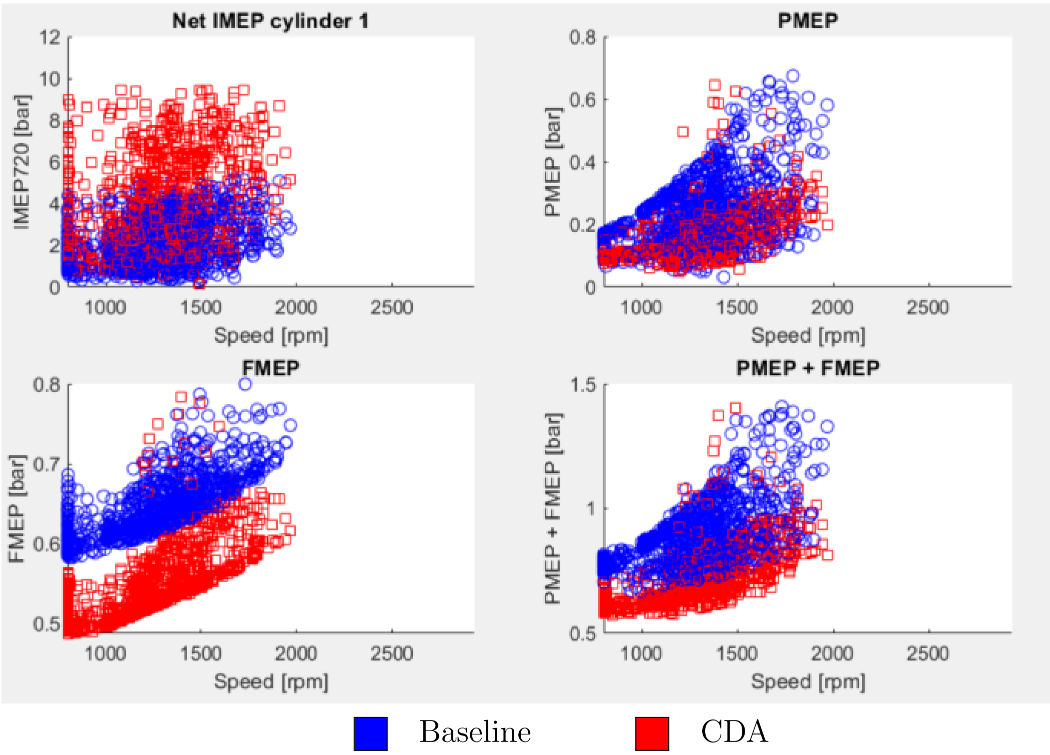

Figure 7.7: Cylinder 1 Net IMEP, PMEP and FMEP corresponding to points where CDA is activated (red), compared to the corresponding values for baseline (blue)

During CDA phases, lower pumping work is observed, due to the non-contribution of the deactivated cylinders. Also, the active cylinders work at higher load (IMEP) than normally, with increased thermal efficiency and thus lower losses.

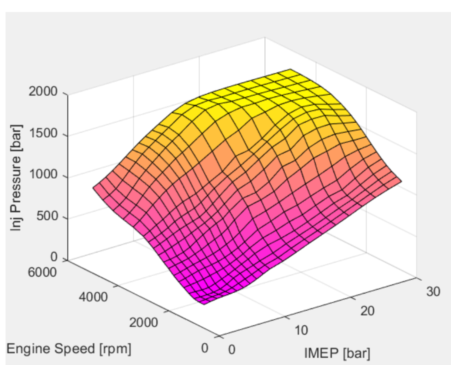

Figure 7.8: Example of injection pressure calibration map, function of cylinder IMEP and engine speed

The ECU works with cylinder mean values, so, during cylinder deactivation (CDA) phases, the control accounts for the higher IMEP that is needed by a single cylinder. For instance, as a first approximation, during CDA, a cylinder would need about twice the fuel that is needed normally for the same load. At the same time, parameters like injection pressure, timing, and requested external EGR are largely affected. For example, from its calibration map, it can be seen how the optimal injection rail pressure should increase with increasing load. Thus, the rail injects fuel at higher pressure during CDA phases.

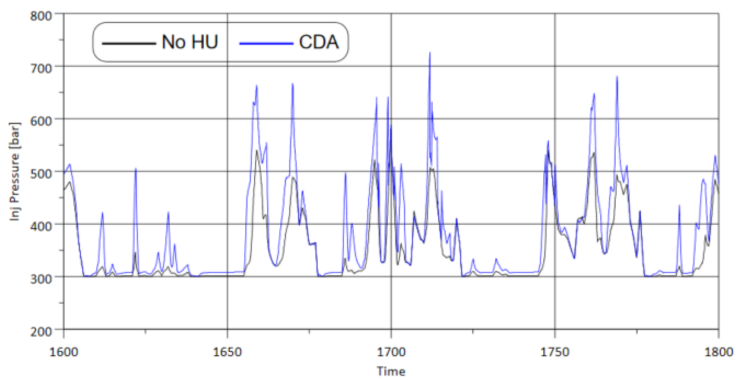

Figure 7.9: Zoom of rail injection pressure during CDA simulation, compared to baseline

In the air path control part, the ECU computes a NO_x concentration set-point, according to engine speed and load, that is used as a starting point for all the air path parameters calculation, in particular the HP and LP EGR desired. At higher IMEP, like in CDA phases, the calibration allows an higher NO_x concentration. Moreover, the NO_x set-point is corrected taking into account the smoke limitation, with a calibration map that gives a minimum relative air-to-fuel ratio (lambda λ), then used to compute the minimum in-cylinder molar fraction of O_2.

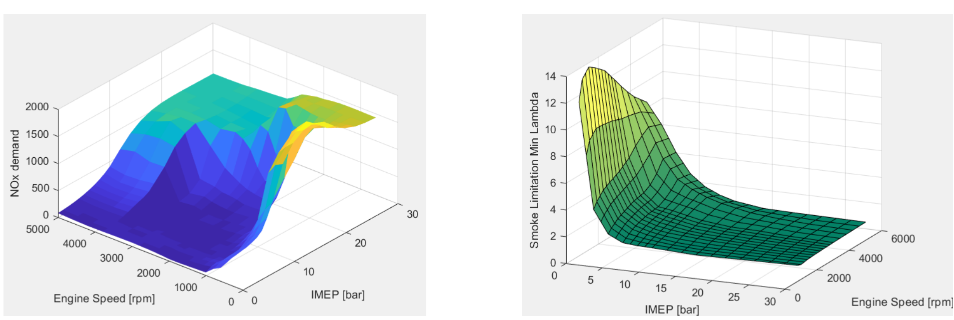

Figure 7.10: Example of NO_x demand (left) and Smoke limitation (right) calibration maps, functions of cylinder IMEP and engine speed

The combined effect of higher NO_x set-point and smoke limitation becoming more stringent (because the fuel to be injected is greater and then the actual λ gets lower, possibly crossing the new limit), makes the ECU request higher in-cylinder oxygen concentration, with the higher IMEP of CDA phases. Then, in CDA simulation, the HP and LP EGR valves are less open, resulting in a lower external EGR rate. For this reason, there are phases of the cycle where CDA gives greater engine-out NO_x emissions than the baseline simulation. However, in very low-load and idle phases, where no EGR or low percentages are requested, CDA NO_x emissions are lower.

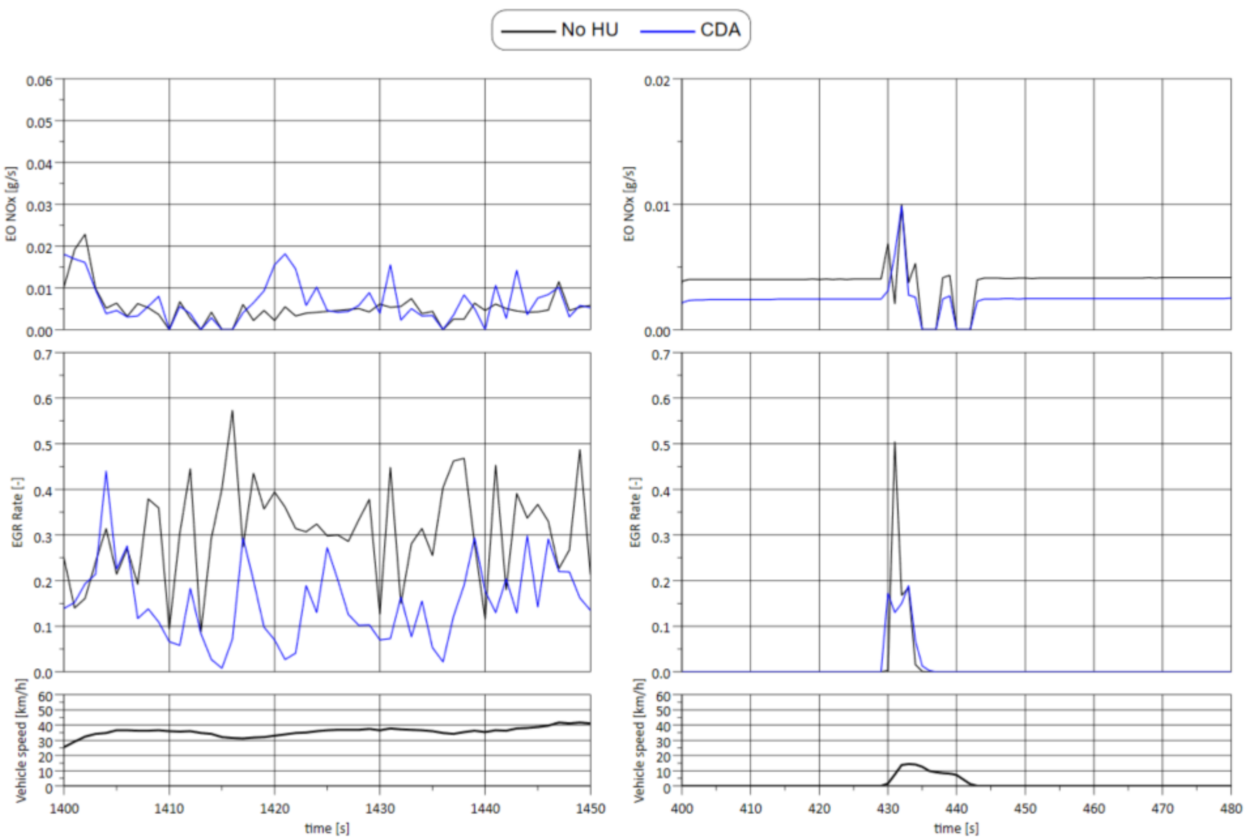

Figure 7.11: Two different zooms of NO_x engine-out emissions and EGR rate of CDA simulation, compared to baseline

The ECU is able to utilize different calibration maps according to the engine combustion mode. Then, in order to exploit better the benefits of CDA, it would be ideal to have a different calibration also for two-active cylinders phases, both in terms of NO_x set-point and smoke limitation. Also, re-calibration may be required for reaching the new target HP and LP EGR mass flow rates, in CDA phases.

7.5 Analysis and comparison between the different strategies

	Fuel	*NO_x EO*	*NO_x TP*	*NO_x conversion eff.*
No HU	ref	ref	ref	25.89%
iEGR	-1.55%	-27.33%	-36.88%	35.63%
LIVC	-0.90%	-2.56%	-4.35%	27.25%
CDA	-8.29%	+4.56%	-14.54%	39.43%

Table 7.1: TFL fuel consumption and emissions results in respect to baseline engine-out

Table 7.1 sums up the differences of the different strategies in terms of fuel consumption, engine-out NO_x and tail-pipe NO_x (after the action of DOC, SDPF and passive SCR). The values from the baseline (No Heat-Up) simulation are taken as reference. Also, NO_x conversion efficiency is defined as the relative difference between engine-out and tail-pipe NO_x. The results can be visually compared through the bar plots, in figures 7.14, 7.16 and 7.17.

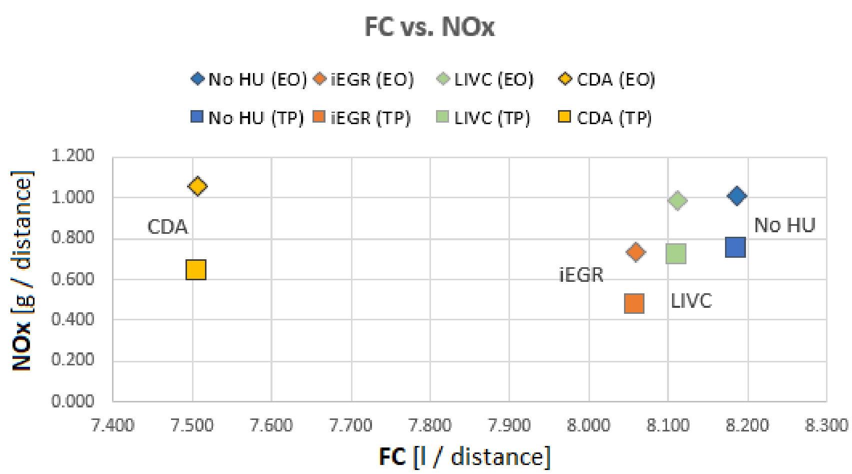

Figure 7.12: Diagram of normalized fuel consumption vs. NO_x emissions (engine-out and tail-pipe)

As for fuel consumption, all the strategies showed benefits. The most effective strategy for saving fuel was CDA, as expected. Also, iEGR fuel consumption is lower than LIVC one, despite iEGR was chosen for its NO_x emissions and warm-up benefits. As already explained, this is due to the lower pumping work.

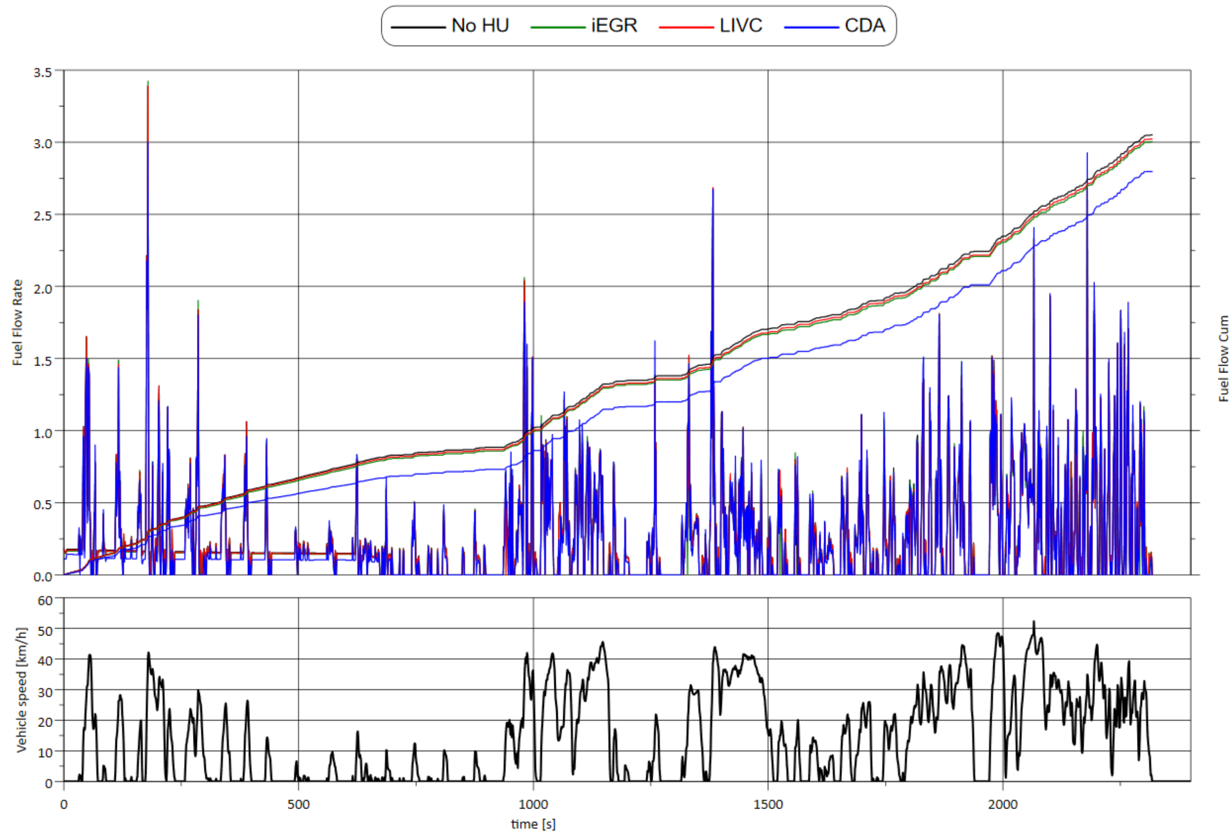

Figure 7.13: Traces of instantaneous and cumulative fuel consumption (only trends, no values)

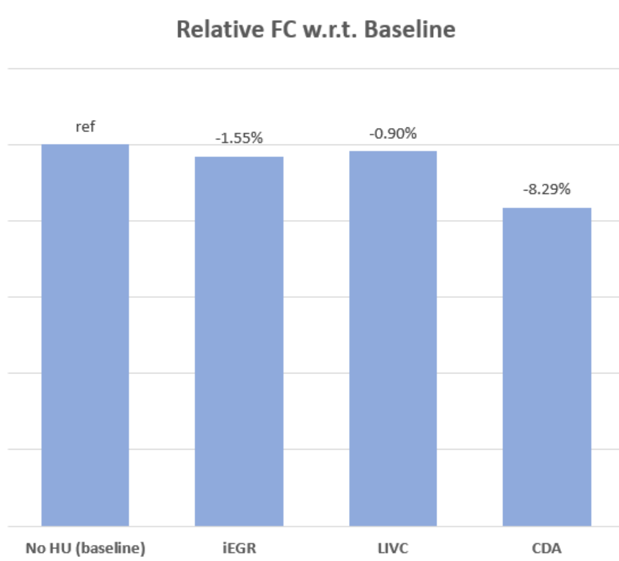

Figure 7.14: Comparison of relative fuel consumption, with respect to baseline

As for reducing engine-out NO_x emissions, iEGR was the most effective, while LIVC still was able to accomplish a NO_x reduction of lower entity. CDA has been the only tested technology to actually be, in the case of these simulations, not beneficial for NO_x reduction, and the reasons have been explained in the previous paragraph.

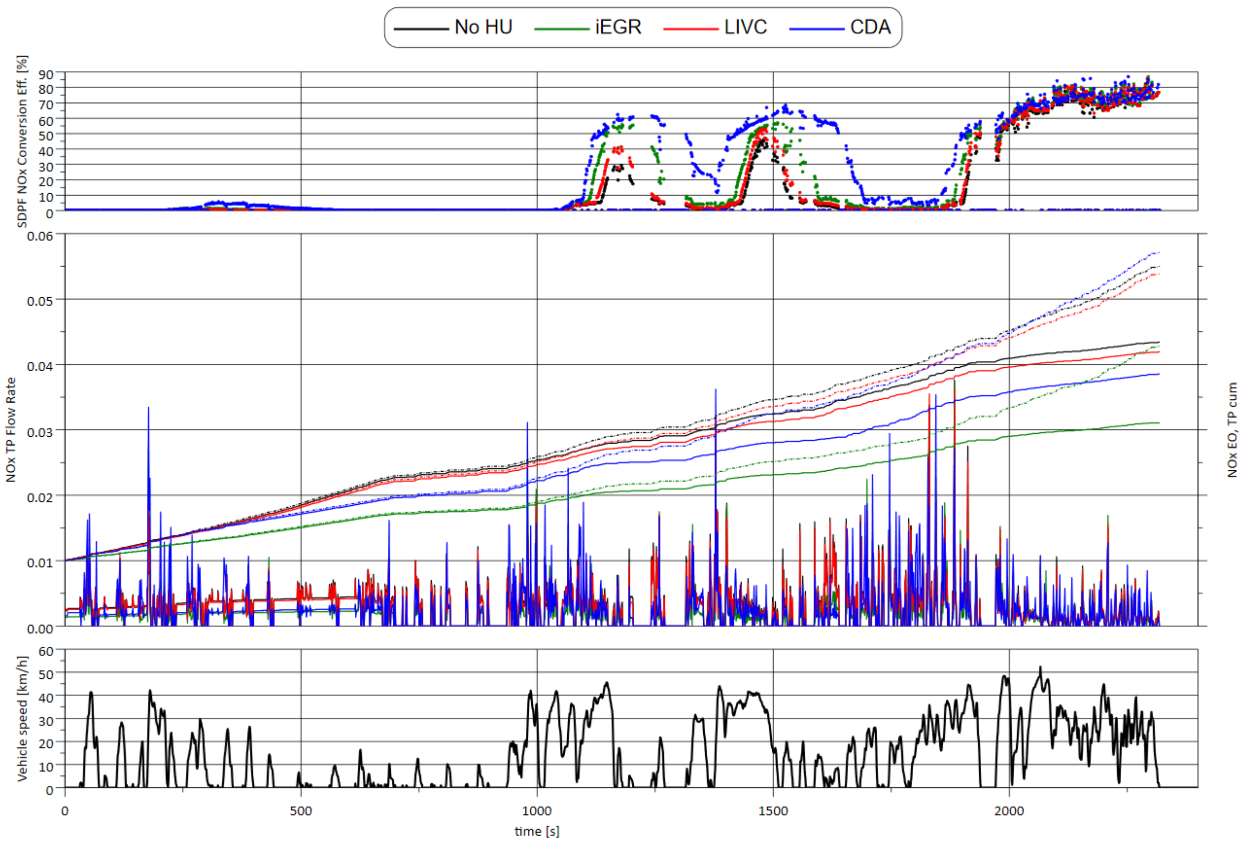

Figure 7.15: Traces of SDPF NO_x conversion efficiency, instantaneous tail-pipe NO_x mass flow, engine-out (dashed lines) and tail-pipe (continuous lines) cumulative NO_x emissions (only trends, no values)

Despite the highest engine-out NO_x, CDA strategy also had the highest NO_x conversion efficiency, resulting in a lower tail-pipe NO_x emissions compared to baseline and LIVC. iEGR was the strategy having the lowest tail-pipe NO_x emissions, with a NO_x conversion efficiency higher than baseline and LIVC.

In case ammonia is not entirely consumed or too much is added, there could be excess NH_3 (the so-called slip) from the SPDF process, and the passive SCR should achieve additional NO_x reduction, utilizing such NH_3 as reducing agent. In the case of these simulations, the NO_x conversion is all related to the SDPF, where Adblue is injected, while the passive SCR is not responsible of NO_x reduction, because there is no NH_3 slip from the previous unit.

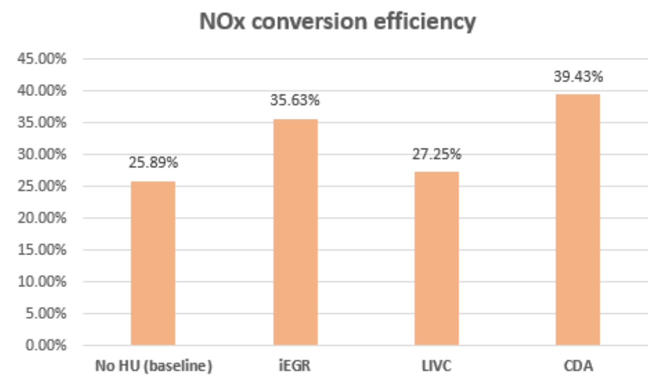

Figure 7.16: Comparison of NO_x conversion efficiency percentage

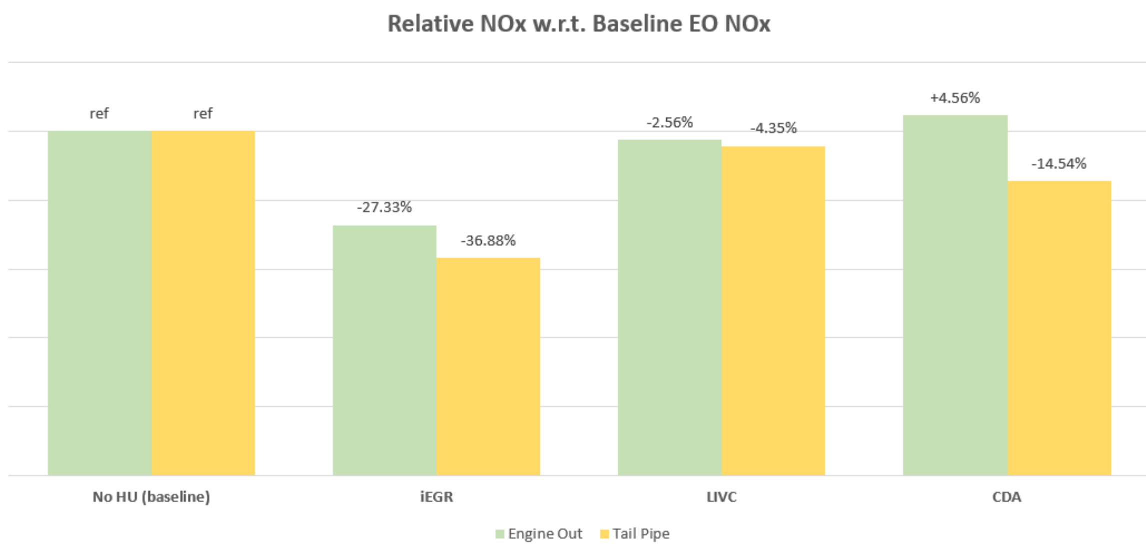

Figure 7.17: Comparison of relative NO_x engine-out (EO) and tail-pipe (TP) emissions, with respect to baseline engine-out

Looking at the temperatures, it can be seen how CDA permits to have higher exhaust temperatures, getting more effective for the warm-up of the after-treatment system. In particular, in the first 200 seconds of the driving cycle, the CDA average SDPF temperature is lower in respect to the other strategies, because the lower exhaust mass flow rate takes more time for the warm-up. Then, the higher exhaust gas temperature heats up the after-treatment to temperatures higher than the ones reached with the other strategies. A similar temperatures behavior, but in lower entity, occurs with iEGR strategy. The higher after-treatment temperature makes it more efficient, thus explaining the higher NO_x conversion efficiencies.

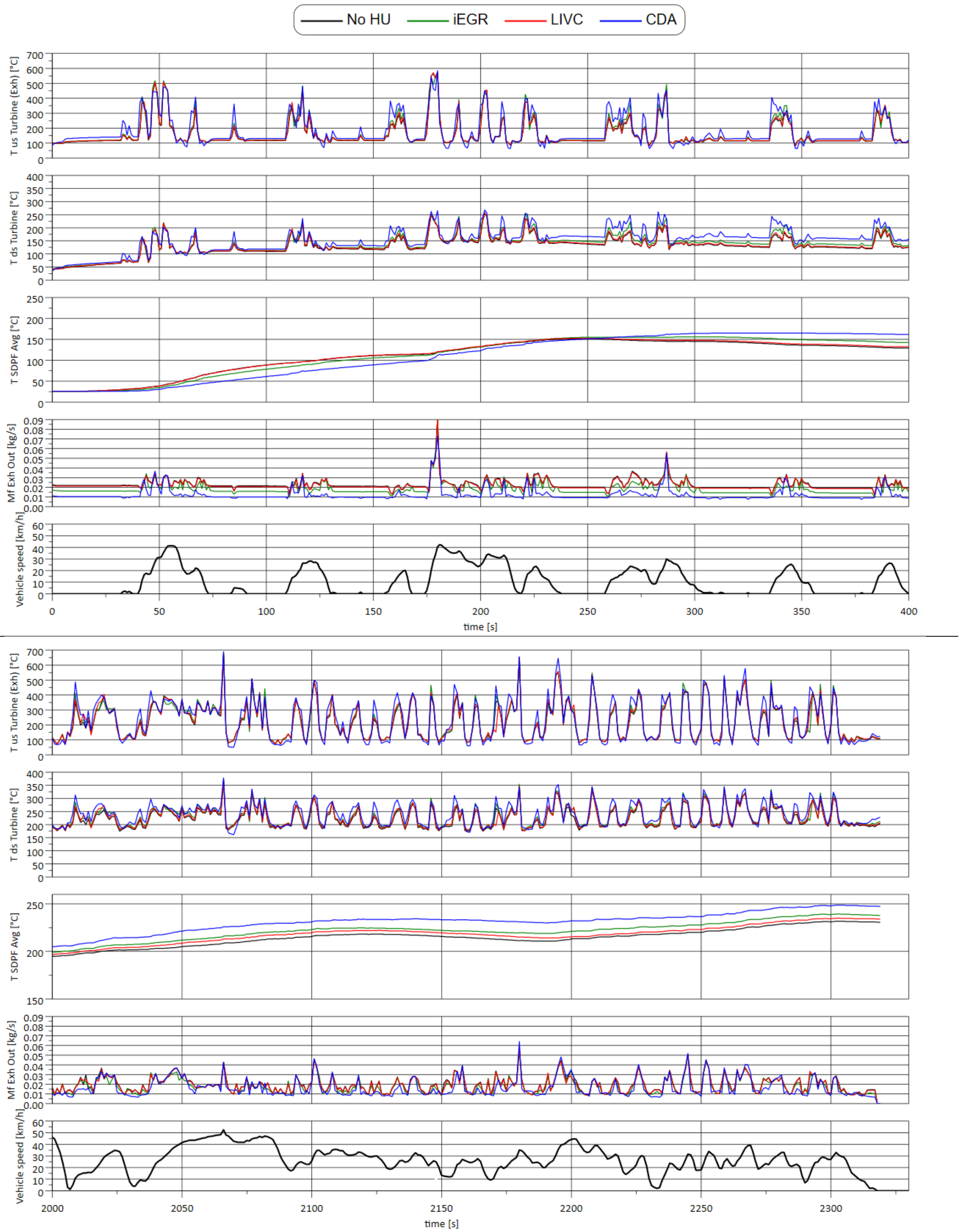

Figure 7.18: Traces of temperature upstream turbine (exhaust), temperature downstream turbine, SDPF average temperature, and exhaust mass flow rate: start of cycle (up) and end of cycle (down)

7.6 DSF simulations

The Cylinder Deactivation part of the model was built so that any pattern of deactivated cylinders could be easily reproducing on a cycle-to-cycle basis. Thus, the model was used to test some Dynamic Skip Fire (DSF) concepts. However, the model was not able to achieve a smooth torque profile during DSF transitions, and so this chapter will focus on investigating the causes of such simulation difficulties and developing ideas for a future work.

All the DSF simulations focused on a simple maneuver at constant engine speed (2000 RPM), and a linear torque transition from 48 Nm to 40 Nm, that takes 25 engine cycles. Before the transition, the engine has a constant torque target and works with four cylinders (Firing Density = 1). After the transition, the engine gets to a different constant torque target, but works with two cylinders (Firing Density = 0.5). Different transition strategies, in terms of Firing Density and active cylinders, were tested, with torque smoothness as objective.

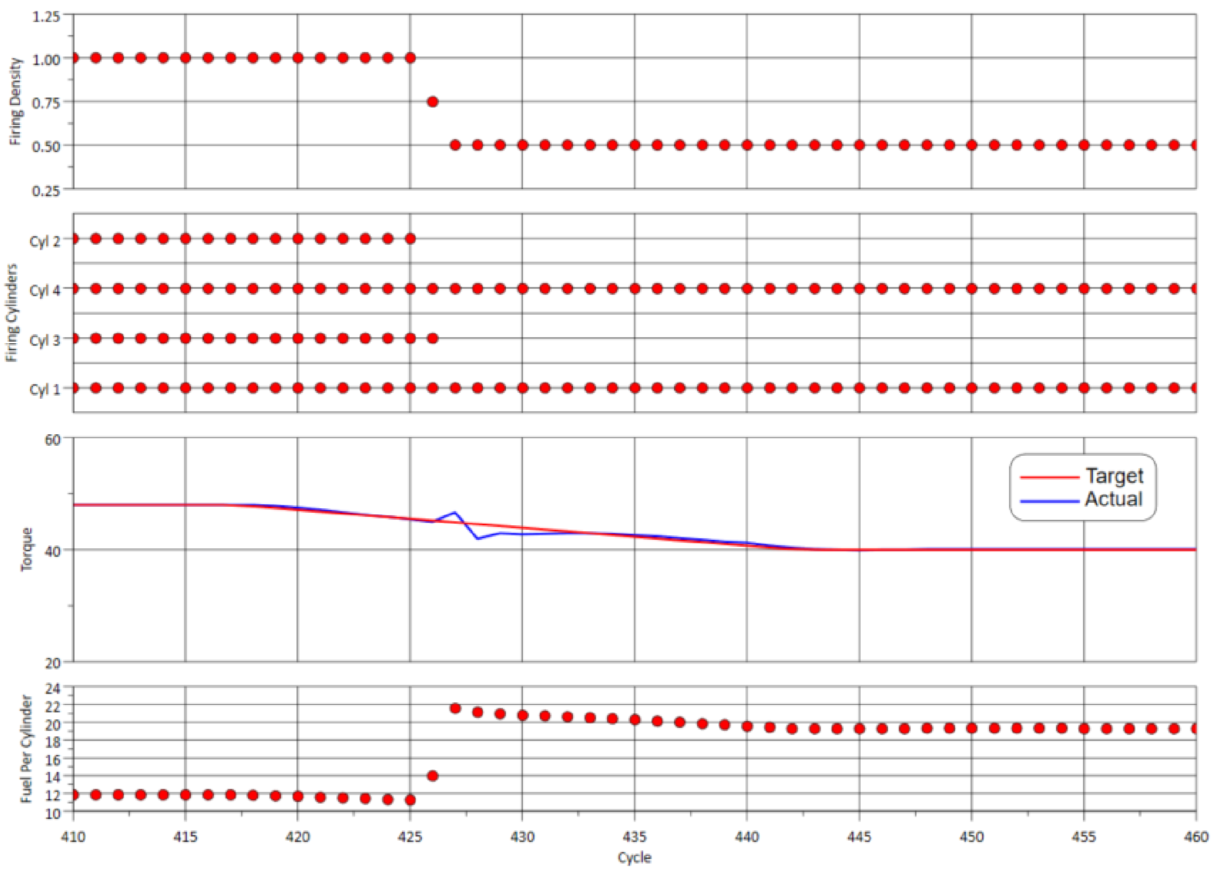

Figure 7.19: Transition 1: Firing Density defined cycle-to-cycle, Firing Cylinders (firing order: 1-3-4-2), Torque [Nm] target and actual value, Fuel injected per cycle in a single active cylinder

The *Transition 1* is a simple transition from 4-cylinder to 3-cylinder to 2-cylinder operation, where the engine works with 3 cylinders only for one cycle. Figure 7.19 shows the firing density cycle-to-cycle, which cylinders are firing and the simulation outputs in terms of torque and injected fuel per cylinder. During the transition, the average actual torque differs from the target of about 2-3 Nm (almost 7% of the target). However, DSF algorithms require more complex and longer transitions of firing cylinders, for NVH reasons.

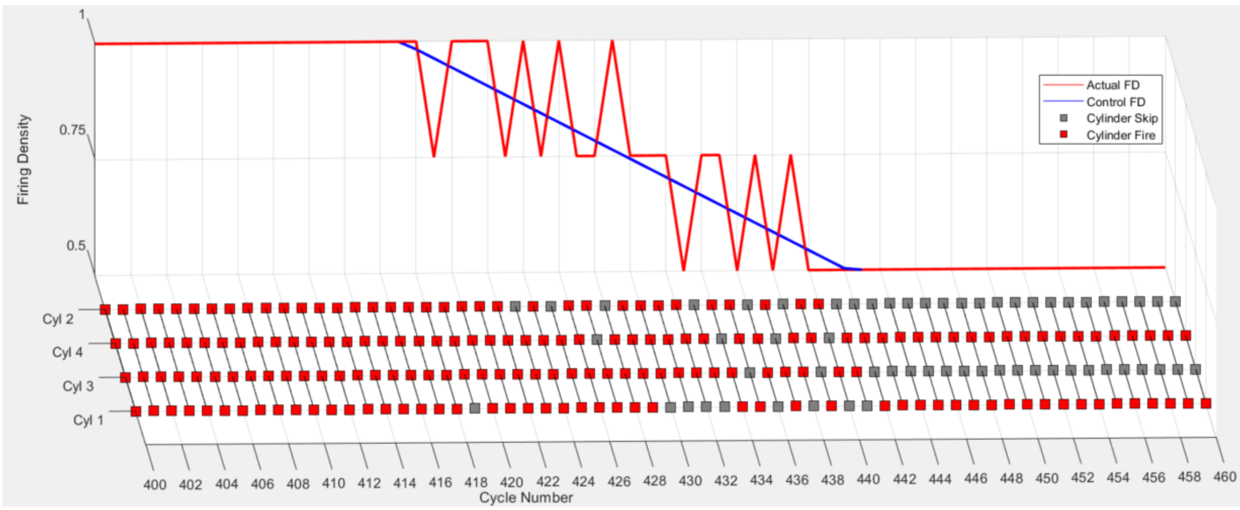

Figure 7.20: Actual and Simulink cycle-to-cyle Firing Density; cylinders fire/skip for DSF Transition 2

The *Transition 2* is a complex 25-cycles (100 cylinder events) transition, taken to represent the possible fire/skip decisions of DSF Tula algorithm, in order to comply with NVH targets. Figure 7.20 shows the firing (red) and skipped (gray) cylinders for each cycle, the actual cycle firing density (number of firing cylinders in the cycle, divided by the total number of cylinders), and a fictitious firing density used in the ECU controls. Operating with four cylinders at the start, and with two cylinders at the end, during the transitions the engine alternates first cycles with four and three active cylinders, and then cycles with three and two active cylinders. It has been chosen to operate the ECU with a linear firing density that goes from 1 to 0.5.

With this approach, the injected fuel in a single cylinders varies almost linearly from the lower value, correct only for four-cylinders operation, to an higher value, correct only for two-cylinders operation. Then, during the transition, where the number of active cylinders has sudden cycle-to-cycle changes, the fueling is not of the correct amount.

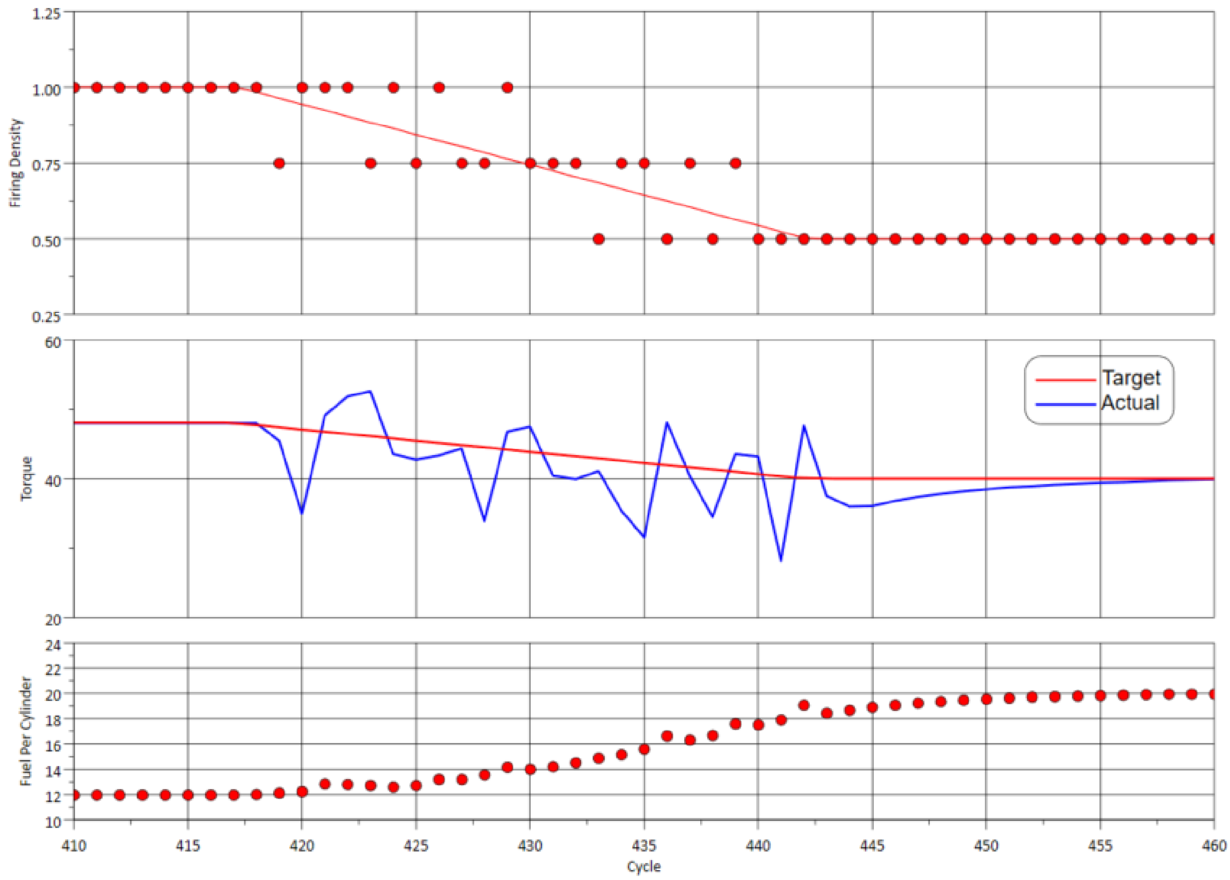

Figure 7.21: Transition 2 simulation results, first approach. Firing Density defined cycle-to-cycle and compared with the one used in controls, Torque [Nm] target and actual value, Fuel injected per cycle in a single active cylinder

As a second approach, it has been chosen to use the actual firing density also in the control part, so varying in a non-continuous way. As for the air-path controls, such as EGR valve position, in the ECU they have an inherent delay modeled, and so, despite the jumps in control firing density, the air-path parameters resulted to vary anyway with an almost-linear trend. Then, the only control parameter (passed from the ECU to the GT Engine model) having "jump" discontinuities was the amount of fuel to be injected in a single cylinder. This approach was then expected to be more correct, because when a single or more cylinder is deactivated, the other ones should compensate it by producing more torque, and so more fuel should be injected.

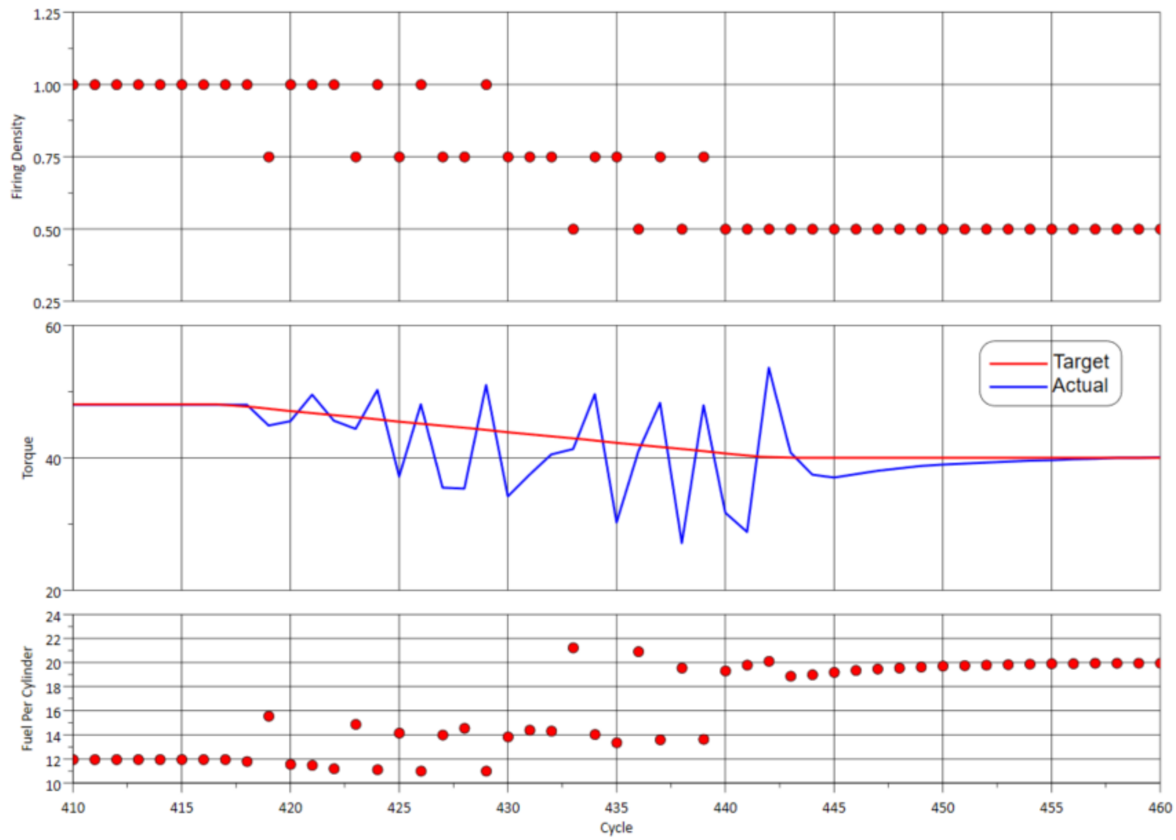

Figure 7.22: Transition 2 simulation results, second approach. Firing Density defined cycle-to-cycle, compared, Torque [Nm] target and actual value, Fuel injected per cycle in a single active cylinder

However, with both approaches, the output torque, given by GT Power results, does not smoothly follow the target profile. Three issues have been found to explain these simulation complications: non precise fueling, choice of averaging window, and transients in the turbocharger behavior.

The control model uses a PID to adjust (indirectly) the amount of fuel injected in a single cylinder. Even if there is a pre-control, that takes into account also the dynamic number of cylinders, the output is not precise enough without the PID to correct it. During DSF transients, where the number of active cylinders changes cycle-to-cycle, the PID does not have time to match the target and the actual torque. Moreover, with the actual model, the amount of fuel to be injected in order to satisfy the torque target has been found to be not only dependent on the firing density, but also on the pattern of active cylinders. A test has been performed by having a constant torque target of 48 Nm, a constant engine speed of 2000 RPM, and changing the patterns of firing cylinders. The quantity of interest was the total fuel to be injected for satisfying the torque demand, normalized by the one needed in all-cylinder operation.

Pattern	*FD*	*Cyl 1*	*Cyl 3*	*Cyl 4*	*Cyl 2*	*Total fuel (normalized)*
1111	1	**F**	**F**	**F**	**F**	1.000 (ref)
1110	0.75	**F**	**F**	**F**	s	0.921
1011	0.75	**F**	s	**F**	**F**	0.922
1101	0.75	**F**	**F**	s	**F**	0.922
0111	0.75	s	**F**	**F**	**F**	0.927
1010	0.5	**F**	s	**F**	s	0.866
1100	0.5	**F**	**F**	s	s	0.883
0011	0.5	s	s	**F**	**F**	0.885
1001	0.5	**F**	s	s	**F**	0.886
0101	0.5	s	**F**	s	**F**	0.875

Table 7.2: Constant torque (48 Nm) and engine speed (2000 RPM) test with different firing patterns (F = Fire, s = Skip)

When considering more than one cycle, all the patterns with three-cylinder operation (1110, 1011, 1101, 0111) are equivalent to three fires followed by one skip, and thus they have almost the same fuel needed. Patterns 1100, 0011, and 1001, with two-cylinders operations, are equivalent to two fires followed by two skips and also in this case their result in terms of fuel is similar. Patterns 1010 and 0101 are equivalent to one fire followed by one skip, being more "balanced" in respect to the other patterns with two-cylinder operation, so they look to be more efficient, but unexpectedly their need in terms of fuel is different of not a negligible quantity.

One of the possible reasons of such differences is how GT Power computes not only the torque result, but also all the engine results, such as pressures and temperatures. The instantaneous quantities are captured through a moving-average filter with a window of one cycle. However, the result of this average during non-stationary behavior is dependent on where the averaging window starts, which in GT Power is approximately at Intake Valve Closure (IVC) of Cylinder 1.

As a future work, advanced fuel compensation algorithms for the firing cylinders transients could be developed, with the support of experimental data and a deeper study of the involved phenomena. In order to avoid averaging issues, it may be better to make an analysis of the torque using the instantaneous one, rather than the cycle-averaged one.

8 Conclusion

According to several publications, Variable Valve Actuation (VVA) has the potential to achieve benefits in terms of fuel consumption, after-treatment warm-up, and emissions reduction also in Diesel engines. In particular, Internal EGR (iEGR), through a secondary opening of the exhaust valve, can lead to NO_x emissions reduction, and Late Intake Valve Closure (LIVC) can enable the "Millerization" of the engine working cycle and lead to efficiency improvements at low load. Cylinder Deactivation (CDA) is a more advanced application of VVA for low load operation and can achieve fuel consumption benefits, while improving the after-treatment warm-up, thanks to the higher exhaust temperatures.
The primary aim of this thesis has been implementing such strategies (iEGR, LIVC, CDA) in a GT Power engine model, and adapting the Simulink control model for correctly matching the actual torque with the desired one. For this purpose, a PID controller is used to close the fuel injection loop, by correcting the estimated engine losses, with torque as target. Moreover, all the calculations in the control model now use a "dynamic" number of cylinder, which changes according to the number of deactivated cylinders.
Then, iEGR, LIVC and CDA have been compared with Baseline (No Heat-Up strategies), by simulating the urban phase of a TFL driving cycle, characterized by low loads, for a passenger car with a 4-cylinder 2-liters engine. As for fuel consumption, all the strategies lead to efficiency benefits, with CDA as the most effective. iEGR was the most effective for reducing engine-out NO_x emissions, while CDA was the only one that actually caused engine-out NO_x increase. However, thanks to the higher exhaust temperatures, CDA was the strategy for which the after-treatment had the best efficiency at converting NO_x, resulting in lower tail-pipe emissions than Baseline and LIVC, but still lower than the iEGR ones.
Such results are influenced by the Engine Control Unit (ECU) calibration, which has been kept the same for all the simulations. As a future work, a model with a different calibration depending on the active strategy could be developed, in order to obtain the best compromise in terms of fuel and emissions. These improvements on the model could also allow to test more aggressive LIVC profiles, considering that the one used for this thesis gave only small benefits (of fuel and emissions) in respect to the baseline.

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