POLITECNICO DI TORINO

Master's Degree in Mechanical Engineering

Master's Degree Thesis

Assessment on the performance of an innovative turbocharged gasoline engine: PHOENICE Project

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Abstract

Global warming and air pollution are the challenges that the automotive industry has to face to guarantee a fast transition towards sustainable mobility. In order to achieve this goal, the best strategy is to follow different paths, from powertrain electrification to innovative high efficiency ICEs. In this framework, the PHOENICE project aims to demonstrate that ICEs can still play an important role for the decarbonization of on-road mobility. To minimize fuel consumption and pollutant emissions, a set of novel technologies has been adopted to maximize the efficiency of the ICE and to fully exploit the synergies with a PHEV hybrid system.
In particular, this project aims to obtain an ICE peak indicated thermal efficiency of 47% using innovative technologies such as E-Turbo and dual dilution combustion (using both lean mixture and cooled EGR) and be compliant with upcoming EURO 7 regulations by means of an exhaust line fit for a lean combustion exhaust gases.

This thesis shows a comparison between the baseline state-of-the-art turbocharged gasoline engine and its modified version of PHOENICE Project.

In this work a Virtual Test rig was modeled using GT-Suite in order to evaluate the improvements of the new solutions implemented in the engine in terms of BSFC and pollutant emissions.

Afterwards, experimental tests at steady state were performed in order to optimize the engine calibration data at the Politecnico di Torino dynamic engine test bench. The results obtained on the Brake Thermal Efficiency (BTE) in the operative key-point 3000 x 13 show an efficiency increase of 5.5%, with a BTE of 40.3% with the Dual dilution approach and a reduction of 70% on NOx emissions. In the medium load key-point 2000 x 5.5 the BTE gain is 4%,with a BTE of 37.5% in the PHOENICE setup, leading to a NOx reduction of 60%.
The $\lambda > 1$ + EGR is the desired condition for a large range of working conditions for the PHOENICE engine.

The best simulation results obtained in terms of Indicated Thermal Efficiency gain can be seen at partial load for the 3500 rpm x 10 bar Brake Mean Effective Pressure (BMEP) key-point, where a strong lean combustion at $\lambda = 1.43$ with EGR rate of 2.0% dilution stated a **net increase of + 4.38%**, with respect to the baseline performances.

Acknowledgements

The completion of this thesis could not been possible without all the support I got from Prof. Federico Millo, Prof. Luciano Rolando and my test-cell tutor Mr. Giuseppe Castellano, that helped me in a very significant way in learning all I needed to obtain satisfactory results.
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I owe a lot to my whole family, to my father Giuseppe, to my mother Cinzia, to my brother Luca and all my relatives that helped me during my hard work, supporting me in each occasion, with a special mention to my grandparents Renato, Lucia, Rocco and my aunt Elena.
A special thanks goes to my roommate Marco, that lived with me for 5 years and grew up with me in this important phase of our lives, and to all my friends, especially to my childhood friends and to the "5th floor kitchen" group, that made me laugh so many times.
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Table of Contents

List of Tables

List of Figures

Acronyms

ASC
: Ammonia Slip Catalyst

aTDCf
: After Top Dead Centre of Firing

AWD
: All Wheel Drive

BMEP
: Brake Mean Effective Pressure

FMEP
: Friction Mean Effective Pressure

BSFC
: Brake Specific Fuel Consumption

BSG
: Belt-Starter Generator

BTE
: Brake Thermal Efficiency

CA
: Crank Angle Degree

CC
: Close Coupled

CD
    Charge Depleting

CoV
    Coefficient of Variation

CR
    Compression Ratio

CS
    Charge Sustaining

C-SUV
    Segment C Sport Utility Vehicle

EGR
    Exhaust Gas Recirculation

EIVC
    Early Intake Valve Closing

EMS
    Energy Management Strategy

EV
    Electric Vehicle

GDI
    Gasoline Direct Injection

GPF
    Gasoline Particulate Filter

ICE
    Internal Combustion Engine

IMEP
    Indicated Mean Effective Pressure

ITE
    Indicated Thermal Efficiency

LDV

Light-Duty Vehicle

LIVC

Late Intake Valve Closing

LS

Lenght Scale

MFB50

Mass Fraction Burned 50Crank Angle

PHEV

Plug-in Hybrid Electric Vehicle

PHOENICE

PHev towards zerO EmissioNs & ultimate ICE efficiency

PMEP

Pumping Mean Effective Pressure

PN

Particle Number

RDE

Real Driving Emissions

SCR

Selective Catalytic Reduction Catalyst

SI

Spark Ignition

TKE

Turbulent Kinetic Energy

TWC

Three-Way Catalytic Converter

UF

Underfloor

VNT

Variable Nozzle Turbine

VVA

Variable Valve Actuation

WCAC

Water-Charge Air Cooler

WG

Waste Gate

WLTC

Worldwide Harmonized Light Duty Vehicles Test Cycle

WLTP

Worldwide Harmonized Light Duty Vehicles Test Procedure

Chapter 1

State-of-the-Art of Spark-Ignition Engines

1.1 Spark-Ignition Engine Overview

An *Internal Combustion Engine* (ICE) is a volumetric device capable of converting chemical energy, contained in the fuel, in thermal and mechanical energy available on a rotating shaft. By mixing gasoline with air by a certain proportion, the mixture obtained can be ignited in a chamber, resulting in fast expansion of the gases with thermal energy release.
The complete process can be performed in an alternative machine, where the work fluid changes its properties during the work cycle. In this machine, the intake, compression, expansion, and exhaust phases are induced by a piston sliding inside a chamber, in which the volume changes due to this movement.
The shaft will rotate actuated by a alternative movement, given by the piston: this conversion is provided by the combination of different mechanical parts in the engine cranktrain.

The main phases of an ICE need two complete rotations of the crankshaft, such as *four strokes*, that is the name of this kind of engine.

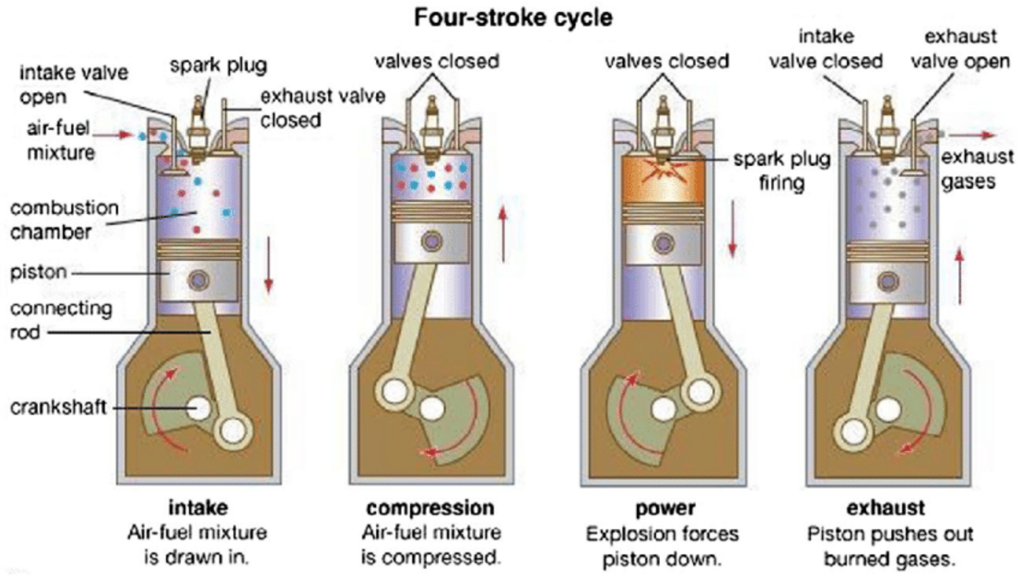

Figure 1.1: Four Stroke Spark-Ignition engine[1]

First the intake phase has to happen: the cylinder has to be filled with air in order mix it with gasoline, if it is not already (as it happens in Port Fuel Injection or carburetor mixing strategies), and start the entire cycle.
Then a compression phase begins, rising pressure in the cylinder and preparing the combustion phase.
Next, the combustion of a compressed mixture is induced by a spark generated by a spark plug at a certain optimal moment: this phase is called *Power stroke* and makes the burned gases push down the piston, generating mechanical energy.
At the end of the process, the new fresh air (or fresh mixture) needs to replace the exhaust gases in order to start again the cycle; the exhaust phase takes place, expelling from the cylinder the exhaust gases. At first it happens by using a pressure delta, given by the chamber pressure, higher than the ambient one, then by pushing them away with the movement of the piston from the bottom (BDC) to the top of the cylinder (TDC).
This fluid exchange is provided by valves, that open and close on the top of the cylinder (placed on the engine head) with a calibrated timing, given by the camshaft, mechanically linked to the crankshaft.

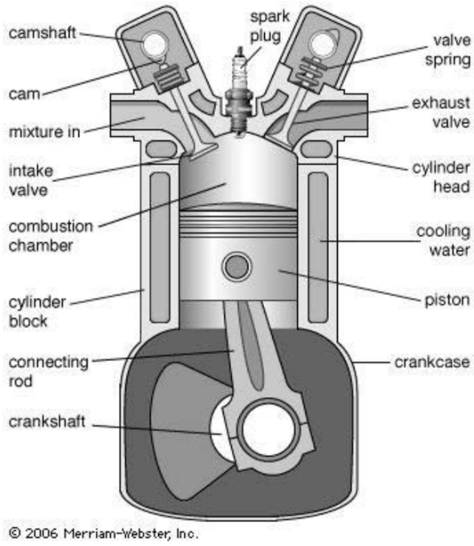

Figure 1.2: Basic Spark-Ignition engine layout[2]

1.2 VVA: effects of Miller cycles

The *Variable Valve Actuation* (VVA) is a system capable of changing the timing of the actuation in both intake and exhaust valves. This goal can be achieved by mechanical, electromagnetic and hydraulic means; it all depends on the constructor preferences.

The mechanical VVA is a solution that bases all its capabilities on a tooth shaped mechanism, that changes the timing by rotating to the designed position for each actuation: the main problem is the durability of this solution due to issues related to low system stiffness and long transients while performing the regulation.

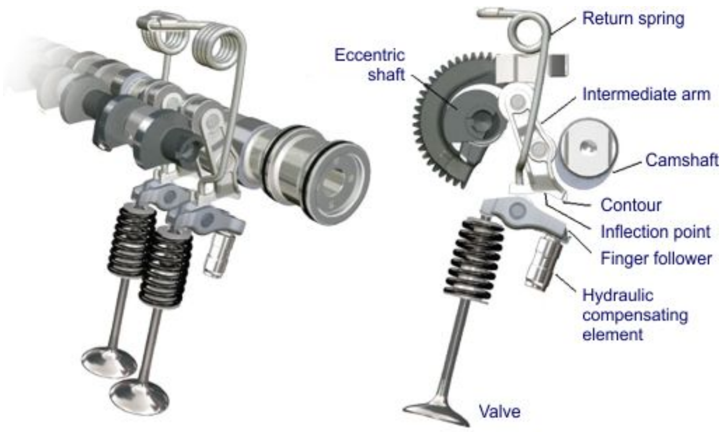

Figure 1.3: Mechanical VVA example

Electromagnetic VVA is the theoretical best solution due to the extremely short transient in valve actuation, but it requires high performance systems that are too heavy and show a very high energy consumption. Moreover, the valve housing may sustain damage from high speed collisions with the valves, without any effective way to reduce this phenomenon.

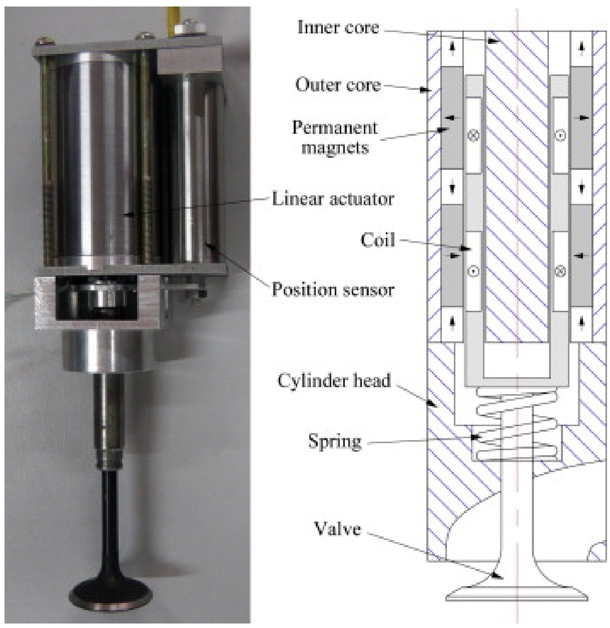

Figure 1.4: Electromagnetic VVA[3]

Hydraulic VVA is the solution that gives the most benefits of the VVA, with very low issues and complications. It uses an hydraulic solution activated by an

electrovalve controlled by the ECU, mainly during partial load conditions, in order to achieve a *dethrottling* result.
The shape of the cams is designed to fit the optimal fluid dynamics performance at full load, but that shape may be not enough efficient during throttled operations.

This system allows to virtually "change" this shape, selecting the correct timing in which the oil lose pressure and make the valve stop following the original shape of the cam. So, the valves will move differently, changing the shape of the original work cycle.

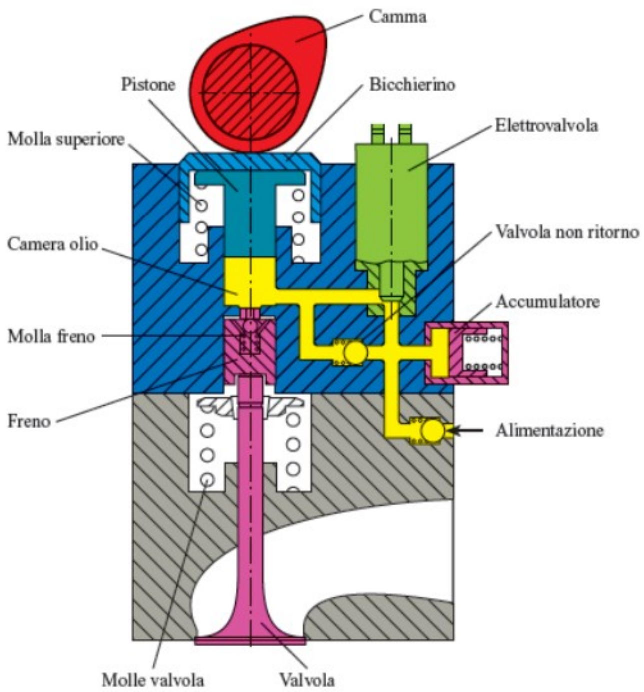

Figure 1.5: Hydraulic VVA working scheme

The engine works with a standard Otto cycle during full throttle operations because the intake flap is fully opened and it does not affect the overall efficiency. The main problem comes at partial loads; the throttle has to be closed as much as the load is low, but this operation causes a lamination of the flow with high dynamic pressure losses, with a significant increase in pumping losses of the engine.

The actual work cycle at full and partial load will be something like this:

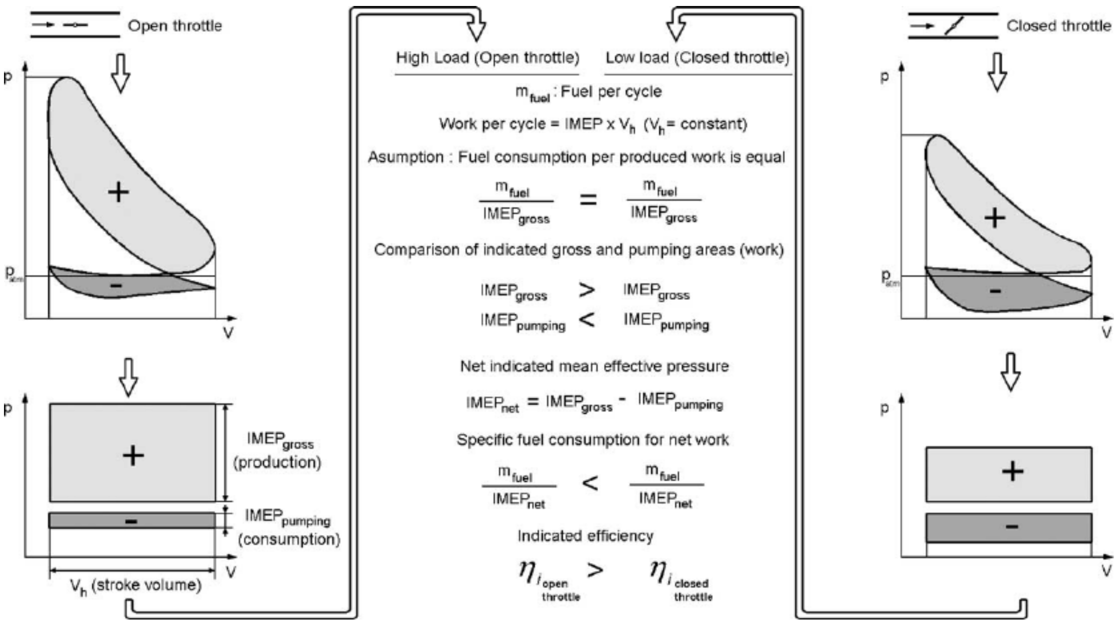

Figure 1.6: Work cycle at full and partial load[4]

The VVA system allows to control the amount of air in the cylinder via valves, keeping the flap open and avoiding these losses.

The Otto cycle modified by this different valve actuation is called Miller cycle: it can be an *Early Miller* or a *Late Miller* referred to the different timing in closing the valves later or sooner than the one of the actual cams, that is geometry based. An example of this calibration is given in the following image:

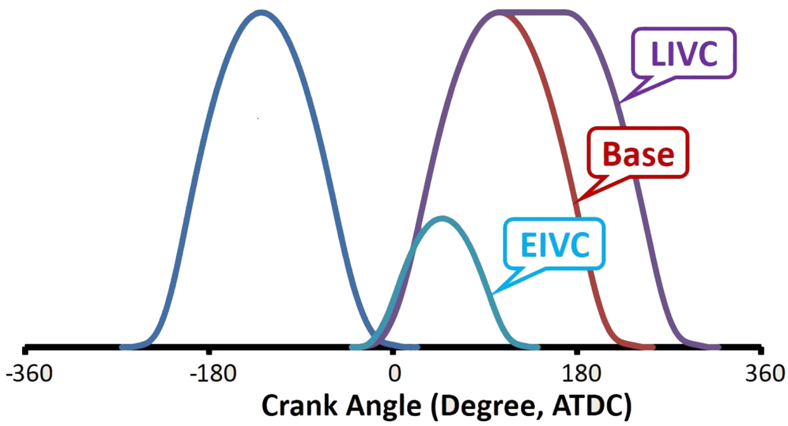

Figure 1.7: Valve actuation in a Miller cycle[5]

The actuation in the Early Miller is named *Early Intake Valve Closure* (EIVC) and in the Late Miller is called *Late Intake Valve Closure* (LIVC).

The effect of the Miller cycle on the efficiency can be easily seen by looking at the area of the pumping losses, where in an EIVC example, obtained by keeping fully opened the throttle, is way more reduced than the exactly same operation done by throttling the flap.

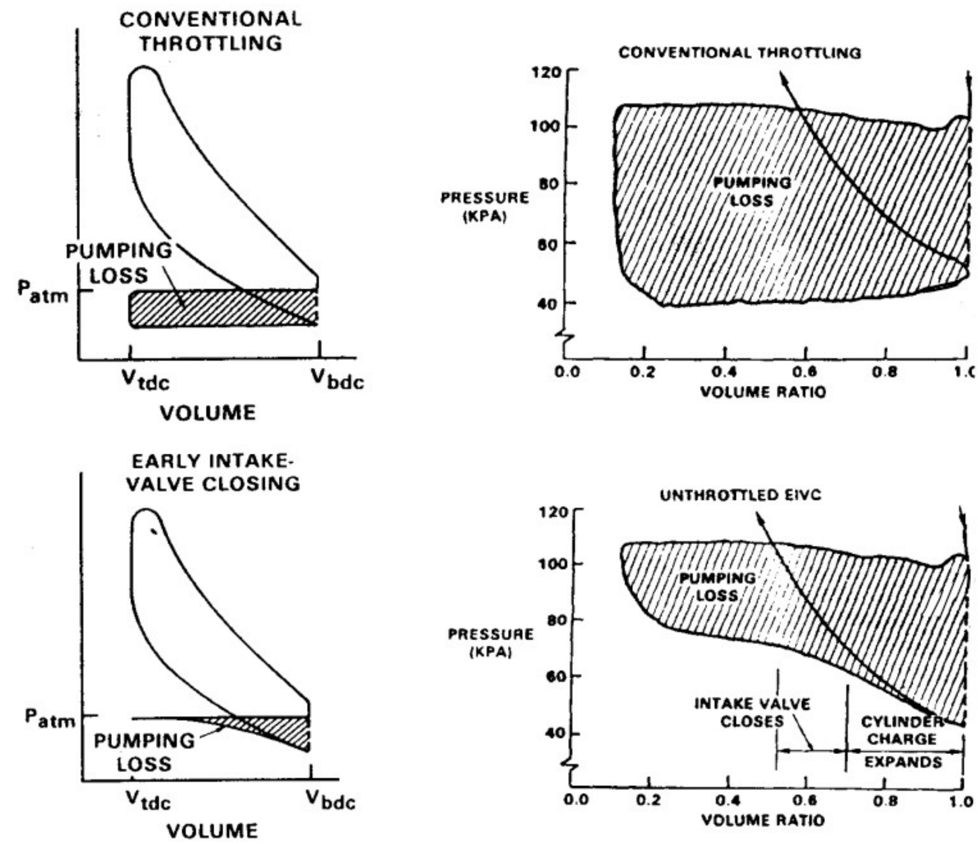

Figure 1.8: EIVC and throttling operations effect on the pumping loss

1.3 GDI: Gasoline Direct Injection

To achieve high performance and efficiency, the only choice is to use a Direct Injection System. This system is capable of high pressure injections, near 200 bar at peak. The injectors are hydraulic actuated by an electrovalve, controlled by the ECU, and they can spray gasoline directly inside the combustion chamber.

The system is a Common Rail that feeds the four injectors, one per cylinder, using a low pressure electric rotary pump and a high pressure alternative pump. These pumps are not synchronized with the injectors due to the presence of this pressurized rail, controlled by the adjustable high pressure pump and by a discharge valve on it.

There are different type of injectors that can be used in a GDI solution; one can be the basic injector, with the atomization performance directly linked to the injection pressure, another one can be the swirler injector, where the atomization of the gasoline is linked mostly to the inside chamber geometry of the injector, reducing its sensibility to injection pressure.

Direct Injection is fit for this use because the vaporization of the gasoline cools down the compressed air in the chamber and raises the volumetric efficiency. The lower charge temperatures have a knock suppression effect too. The vaporization itself is very fast due to the high temperature of the compressed air and it allows even to work with a stratified charge.

A standard Spark-Ignition engine, however, works with an homogeneous stoichiometric charge in order to obtain exhaust gases fit for the Aftertreatment system (CO, HC and NOx reduction via Three Way Catalyst).

The only drawback of the Direct Injection is the formation of an high number of small particles, called Particulate Matter (PM), related to local spots with high gasoline density respects stoichiometric conditions, induced by the injection during the intake/compression phase, that is absent in a Port Fuel Injection (PFI).

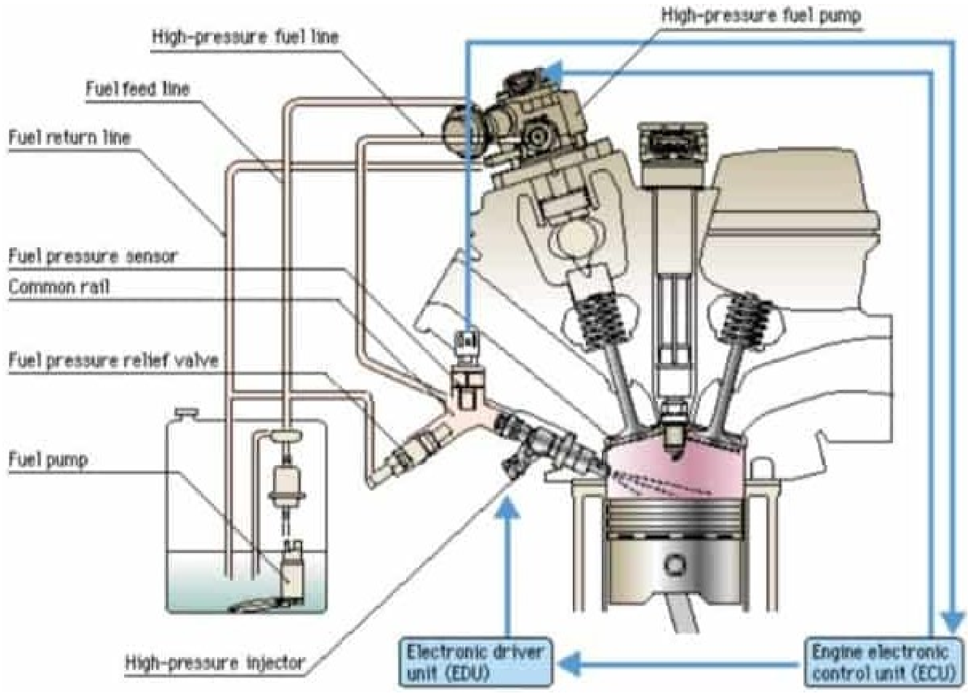

Figure 1.9: GDI System scheme[6]

1.4 Turbocharger with Waste Gate

The Miller cycle used at partial load and the high peak power obtained for a small displacement, typical of a downsized engine, are achievable only by means of turbocharging. A standard Spark-Ignition engine uses a turbocharger with a WG, in order to regulate the maximum pressure of the mixture in the intake ducts during high load operations. In this way the amount of air within the cylinder can be more than the expected quantity of a same displacement NA engine, with a really high torque obtained at low rpm.

Turbocharging is very important for the Miller cycle because it allows to obtain higher intake pressure in order to guarantee a bit of performance during this working condition with very small engines.

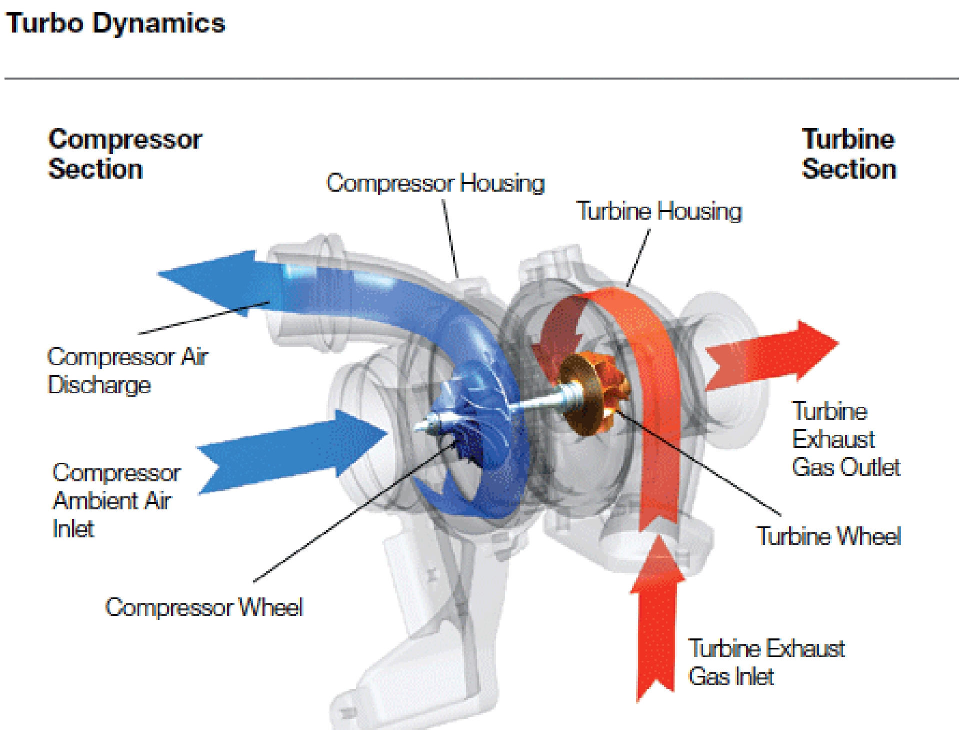

Figure 1.10: Turbocharger working scheme[7]

The turbocharger used is a fixed geometry turbine on the exhaust side; the compressor is linked to the turbine using the same turbo-shaft and it has a target maximum intake pressure to avoid to damage the intake manifold and to suppress knock that may occur at the highest pressures.

To limit that pressure, a Waste Gate is used. The exhaust line, once the four different pipes have converged into a single one, has two manifolds, one on the inlet of the turbine and another that bypass the turbine itself. When the turbine is spinning too fast and consequently the compressor is reaching a pressure higher than limits, the WG is opened progressively in order to decelerate the turbine, allowing the gases to pass inside the bypass and preventing them from passing inside the turbine. The quantity of gases that pass from the bypass is controlled by the opening level of the WG, imposed by the ECU.

This solution is not optimal for efficiency, but it is reliable, simple and low cost, highly used in turbocharged engines.

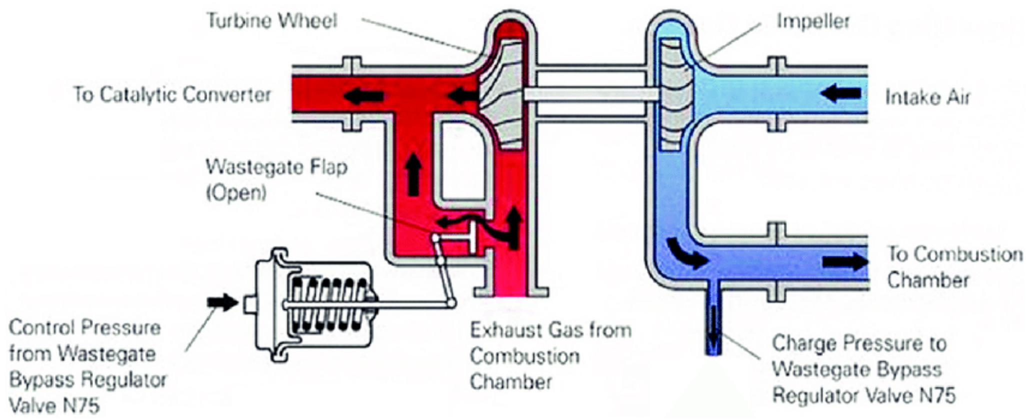

Figure 1.11: Waste Gate System scheme[8]

1.5 EGR System

Miller cycle is very useful to avoid throttling and the losses connected to this process, but there is another way to do dethrottling, while reducing pollutants emissions. The *Exhaust Gas Recirculation* (EGR) is a system that allows the exhaust gases to enter in the combustion chamber and to mix with fresh mixture in order to reduce the combustion temperature and to fill the chamber with non-reactive gases during partial loads, without throttling.

This dilution can reduce the NOx emissions, that are related to the temperature of the gases (by an exponential law) during the combustion, in a extremely significant way.

There are two different ways to achieve this dilution: the first is short-route EGR, using the engine-out gases; the other is long-route EGR using the exhaust gases that have been treated by the ATS and so cleaned.
Typically, a long-route EGR is used in order to keep the combustion chamber clean and to add this gases directly in the compressor inlet at the intake manifold without causing damage.

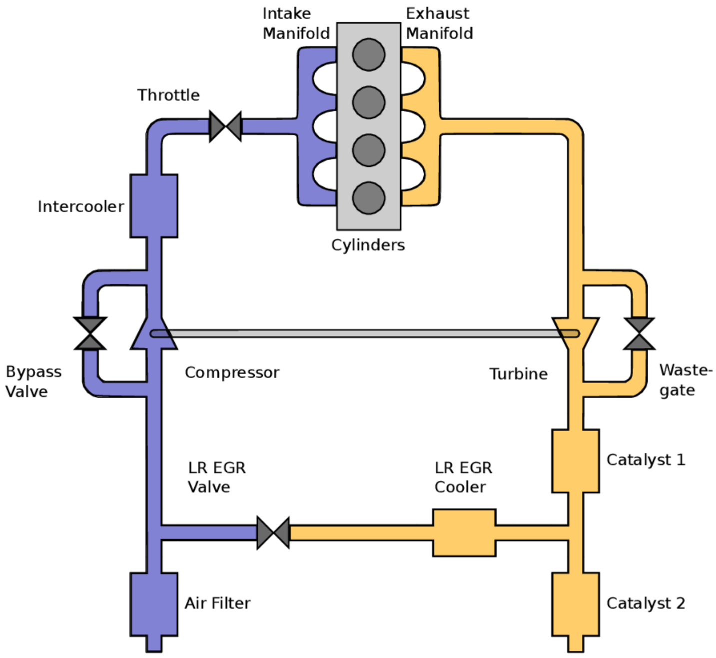

Figure 1.12: Long Route EGR System[9]

1.6 After Treatment System

In order to sell a vehicle, European Regulations established a standard in maximum tail-pipe tolerated emissions; they are classified into pollutant classes such as CO, HC, NOx and Particulate Matter divided in Particle Mass (PM) and Particle Number (PN).

Emission limits	**EURO 6d-final**	**EURO 7 (proposed)**
CO [g/km]	1.0	0.5
HC [g/km]	0.10	0.10
NOx [g/km]	0.06	0.06
NH_3 [g/km]	-	0.02
PM [g/km]	0.005	0.005
PN [#/km]	$6.0 \cdot 10^{11}$	$6.0 \cdot 10^{11}$

Table 1.1: EURO 6 and EURO 7 emission limits[10][11]

To achieve this reduction, the engine-out gases need an *aftertreatment* done by a Three Way Catalyst (TWC), that uses Platinum(Pt) and Palladium(Pd) to *oxidize* HC and CO and Rhodium (Rh) to *reduce* the NOx.
The TWC is built as a cylinder shaped monolith full of small honeycomb passages with a coating of Cerium Oxides, that provides the stoichiometric stabilization of the exhaust gases by means of its *Oxygen Storage Capacity* (OSC).
In this coating there are the active sites of the catalysts (Pt, Pd, Rh) that react with the gases by wall diffusion.

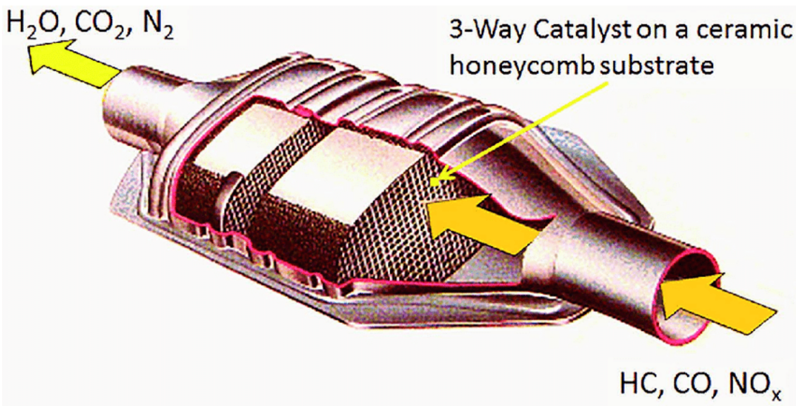

Figure 1.13: Three Way Catalyst (TWC)[12]

Stoichiometric environment is extremely important because the lean mixture can achieve very low HC and CO, but high NOx and in rich mixture there will be high HC and CO, but very low NOx. It is the only way to work with both oxidation and reduction at the same time, in the same component, with high conversion efficiency.

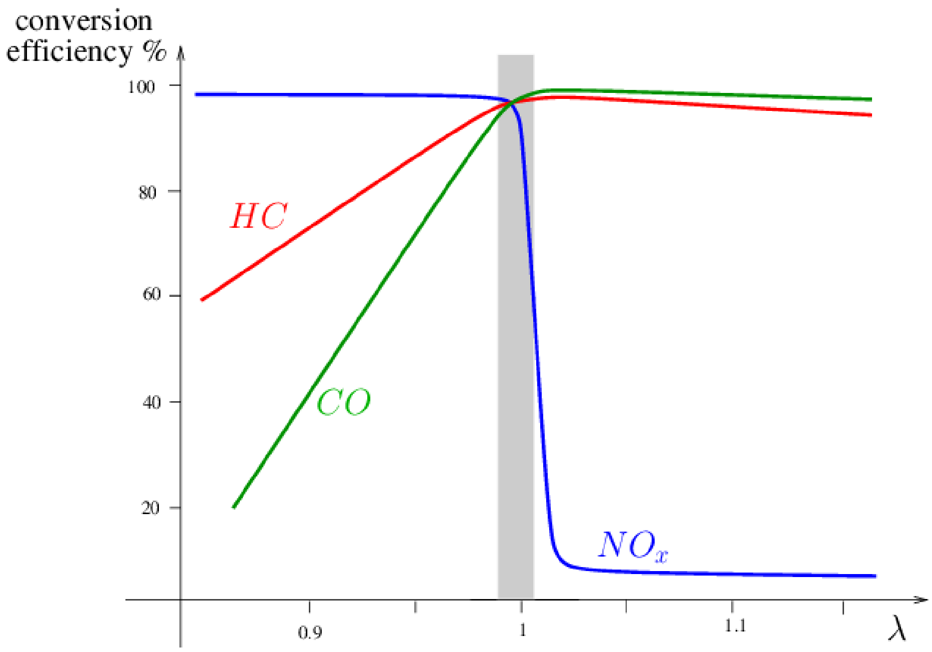

Figure 1.14: Conversion efficiency of CO, HC and NOx peaks at stoichiometric[13]

The pollutants control made by the TWC, however, it is not enough because of the PM limits. A separate system, fit for filtering the smallest PM particles, called Gasoline Particulate Filter (GPF) has to be installed in order to achieve a correct EURO 6-final compliance.
This system has a honeycomb structure too, but with some passages closed at the end in order to get the gases pass through the GPF walls, making a strong filtering, in order to exit from the system itself.

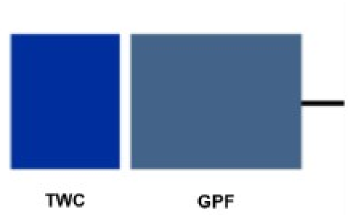

Figure 1.15: EURO 6 exhaust line layout[14]

1.7 Hybrid Solutions

The ICE can be optimized significantly with the help of electric motors, powered by batteries of different size. The layout chosen for this solution helps to classify different hybrid approaches.

The main goal of an hybrid architecture can be the ICE optimization as well as a 4x4 solution without any mechanical differential connecting the front and rear axle; it can be also the possibility to turn off the ICE in urban operations and move the entire vehicle in electric mode only. The size of the battery and the one of the motors are strictly related and they can introduce a significant amount of additional mass to the vehicle, if the motor is pretty big; this parameter can distinguish the level of hybrid architecture as Start and Stop (S&S), mild-hybrid and full-hybrid. But this hybrid solution can use the motor also as a generator, introducing the possibility to recover wasted energy during braking phases.

Using the BOSCH classification, the hybrid layout can be:

- P0, also called BSG, such as Belt Starter Generator, that is an electric motor used in the front part of the engine. It replaces the classic starter and provides the Start & Stop operations, electric assistance during high load conditions and regenerative braking, if used as generator.

- P1, that is an electric motor placed in the back part of the engine, right before the clutch, it is directly connected to the crankshaft. It is limited in the dimensions, power and rotational speed due to this mechanical connection with the engine shaft, but it can provide higher torque and power than the P0 configuration.

- P2, that is very similar to the P1 solution, but it is placed after the clutch. This configuration can drive alone the entire vehicle because it can provide power to the transmission shaft without the engine contribution, that stays isolated from the system by activating the clutch. P2 motors and battery have to be big enough to provide some kilometers of autonomy.

- P3, that has the same function of P2, but it has a different positioning; it is connected to the shaft between the transmission and the differential. This solution can provide a pure electric propulsion with big enough motors and battery.

- P4, that is a big motor and battery, located in the axle that does not provide traction. This solution can move alone the vehicle, but it is designed to provide a 4x4 transmission without mechanical parts, avoiding frictions and so higher fuel consumption.

All these solutions can be combined, obtaining an ICE that benefits of the P0 and provides additional power with P2, to make an example.

The way the energy is introduced in the batteries can be another parameter to make a classification:

- Simple Hybrid (HEV), such as the battery uses and stores energy without any external source, calibrating the medium level of the battery at an optimal percentage to achieve durability. This strategy is also called *Charge Sustaining*, that fits S&S and mild-hybrid vehicles.
- Plug-in Hybrid (PHEV), such as the capacity of the battery is high and it needs to be externally charged and the motor is fairly big in order to provide effective electric traction. Full hybrid vehicles uses this solution and the control on the State of Charge (SoC) of the battery is done by using a *Charge Depleting* strategy, where the energy has to be used intensively from full charge to a minimum level, where a charge sustaining strategy start working instead of the previous one, if the vehicle cannot be charged.

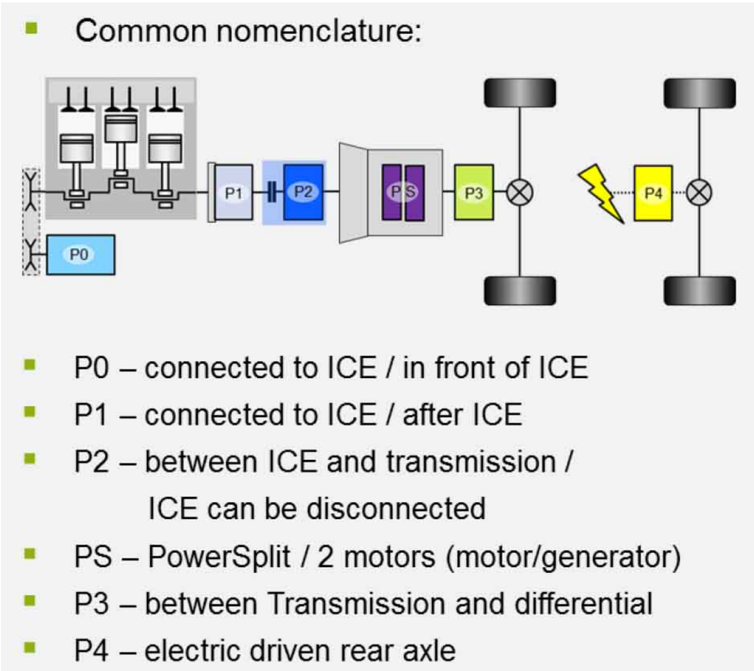

Figure 1.16: Hybrid Layouts[15]

Chapter 2

PHOENICE Project

2.1 Project Overview

The PHOENICE Project is an european research program that aims to demonstrate the full potential of a plug-in hybrid vehicle in reducing fuel consumption and pollutant emissions under real driving conditions[14]. The goal is to obtain a revised and modified state-of-the-art GSE T4 with a peak gross indicated efficiency of 47% and an EURO 7 exhaust line, implementing new technical solutions. These are mainly related to an *E-Turbo* with a VNT pressure control, a new turbulent motion studied by the IFP called *SwumbleTM*, a new *high pressure injection system* up to 350 bar and *redesigned piston head geometry* to increase the compression ratio up to 13.6. The combustion process is modified too, using a *Dual Dilution* approach, such as working in an ultra lean mixture conditions and using high EGR rates, up to 20%, with an enhanced VVA strategy, in order to perform Miller cycles.
However, the consistent modifications done to the baseline engine lead to a reduced peak power for the PHOENICE configuration. This is a trade-off that sees less power for a significant improvement in efficiency and in emissions abatement. Due to the heavy dilution, both in EGR and lean conditions, the engine needs more chamber volume to develop the same power of the baseline engine, that works with a stoichiometric combustion. The displacement of the PHOENICE engine is exactly the same of the baseline one, so the maximum BMEP reachable inside the chamber will be lower. In fact, the full load curve is limited at 20 bar BMEP, a value that is 30% lower than the one of the stock GSE T4.

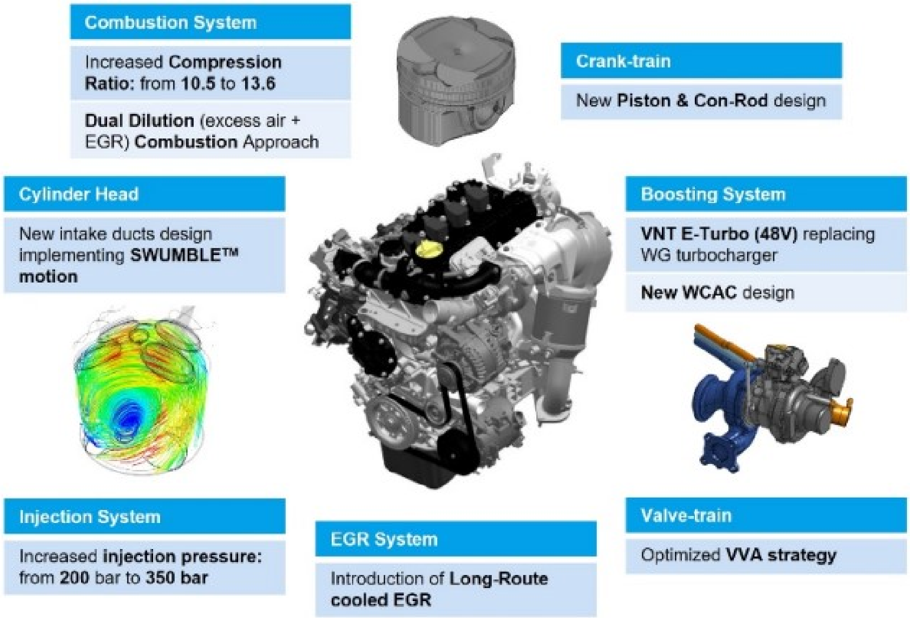

Figure 2.1: PHOENICE overview technologies[14]

2.2 Baseline engine: 1.3 GSE T4 Turbo

The GSE T4 is an engine, produced by Stellantis, used mainly in C-class SUV. It is an inline 4 cylinder 1.3 liter turbocharged gasoline engine equipped with a VVA system called *Multiair*. It has a 10.5:1 compression ratio and a peak power of 132 kW at 5750 rpm. This engine is always coupled with an hybrid P0 unit (capable of regenerative braking and start&stop function) and a P4 electric engine on the rear axle in order to achieve a 4x4 solution without any differential. The exhaust line is designed for stoichiometric combustion gases and in compliance with the *EURO 6-final* homologation.

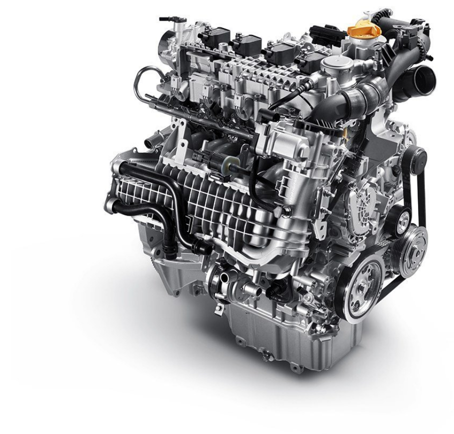

Figure 2.2: Stellantis 1.3L GSE T4[16]

2.2.1 Engine Main Characteristics

The GSE T4 engine is equipped with the most implemented technologies available on the market to achieve high power and efficiency for a downsized gasoline engine, such as a Turbocharger with a WG, a *VVA System* and a *Direct Injection* high pressure system. It is also coupled with a exhaust system in order to achieve an *EURO 6-final* homologation. The combustion process is a standard Otto cycle during full load operations, but it can be optimized, during partial load conditions, by changing the valve timing and lift with the Multiair system. This VVA strategy allows to use internal EGR to perform light Miller cycles, mainly in Early Intake Valve Closure mode. In all the working conditions, in order to achieve a correct pollutants reduction via TWC, the mixture has to be *stoichiometric*.

The main technical information about the GSE are provided in the following tab and they are extracted from *Stellantis official website* .

Engine	**1.3 GSE T4 Turbo 180 CV**
Specs	1.3L 4-cyl in-line 16v
Features	MultiAir VVA, GDI, TC with WG
Bore x Stroke	70 x 86.5 mm
Stroke/Bore	1.24
Compression Ratio	10.5:1
Max. Power	132 kW @ 5750 RPM
Max. Torque	270 Nm @ 1850 RPM
Injection system	GDI- Gasoline Direct Injection
Hybrid system	Belt-Starter Generator (BSG) / P0

Table 2.1: GSE tech specs[17]

2.3 PHOENICE engine features

In this section there will be a brief explanation of the main modified components of the GSE T4 that are implemented in the PHOENICE test engine, in order to achieve the efficiency gain expected and a significantly reduction of pollutants emissions, needed to be compliant to a EURO 7 homologation.

2.3.1 SwumbleTM in-cylinder motion

In a gasoline engine, the main turbulent motion, in order to increase efficiency and combustion performances, is the *tumble* motion. However, in a ultra lean combustion engine, where this dilution can reduce drastically the flame speed propagation, inducing combustion instabilities, it is not enough.
The only way to operate with a certain stability is to use a new turbulent motion solution, that is strong enough to allow good performances. IFP studied the SwumbleTM, a combination between tumble and swirl, the two main turbulent

motions in an ICE, in order to obtain a way to increase efficiency over the top optimization level of a tumble-only motion.

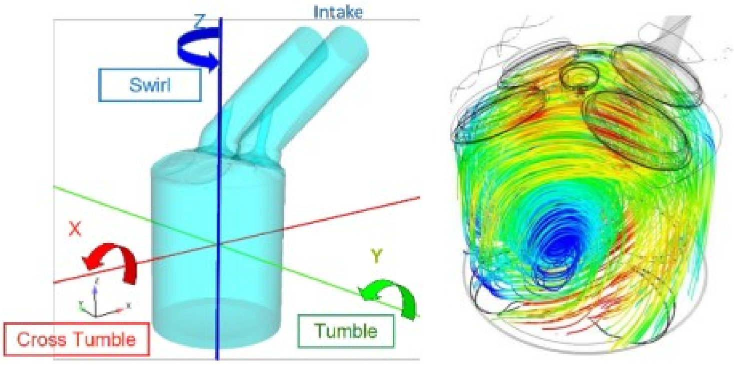

Figure 2.3: Left: turbulence on different axis; Right: SwumbleTM motion[18]

This complex motion has to be generated by re-designed piston head, intake ducts and valve motion at different loads, using the Multiair VVA with an enhanced calibration and actuation logic linked to both Miller cycles in EIVC and LIVC. The Turbulent Kinetic Energy in the cylinder at TDC will be from 0.33% up to 112% in different working conditions, as IFP obtained during its experimental campaign[19].

TKE improvements in [%]	EIVC	LIVC
Average at 690 CA	+53%	+55.8%
Average at 700 CA	+54.4%	+49.7%
Average at 720 CA	+11%	+0.33%
@ Spark plug at 690 CA	+8.4%	+41.8%
@ Spark plug at 700 CA	+24.3%	+32.2%
@ Spark plug at 720 CA	+112%	+46.3%

Table 2.2: SwumbleTM effects with respect to the baseline in Miller cycles[14]

2.3.2 E-Turbocharging

The PHOENICE engine is capable of high power and optimal Miller cycles thanks to its turbocharger. In fact, it is coupled with an internal electric motor, placed on the turbo-shaft, and it has a *Variable Nozzle Turbine* (VNT) instead of the classic Waste Gate, in order to increase the overall efficiency.

This component can dynamically change the geometry of the turbine in order to regulate the quantity of exhaust gases in the turbine itself. At low loads, the VNT is nearly closed in order to accelerate the flux and give enough boost pressure; at high loads, the VNT has to open more to make the gases pass by without significant restrictions. In fact, by opening the vanes, the flux will be less constrained and the turbine will decelerate, regulating the pressure in the compressor, on the other side of the turbocharger.
This method is way more efficient than using a simple bypass with a valve, but it cannot be used in a standard gasoline engine because the high temperature gases damage the regulation rack. However, the ultra lean combustion used in the PHOENICE makes this application possible by reducing the engine-out gases temperature to the allowed structural limits of the rack.
In order to control correctly the system, the PHOENICE engine needs a new ECU program calibrated on the optimal level for the position of the vanes, adding a new variable while generating the engine maps.

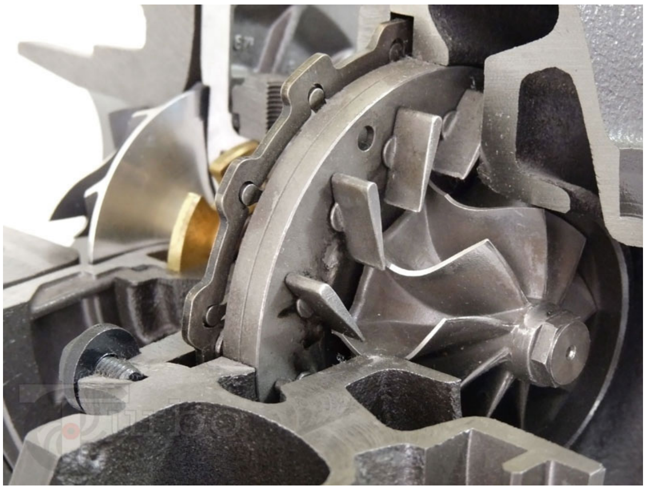

Figure 2.4: Turbocharger with VNT system[20]

The electric motor is installed on the turbo-shaft linking turbine and compressor. This solution allows to make the turbocharger reach the target speed with very short transients, delivering torque when the group has to accelerate and reducing drastically the turbo-lag. It has also to recover kinetic energy from the gases not used by the turbo during low load operation, by making the motor work as a generator.

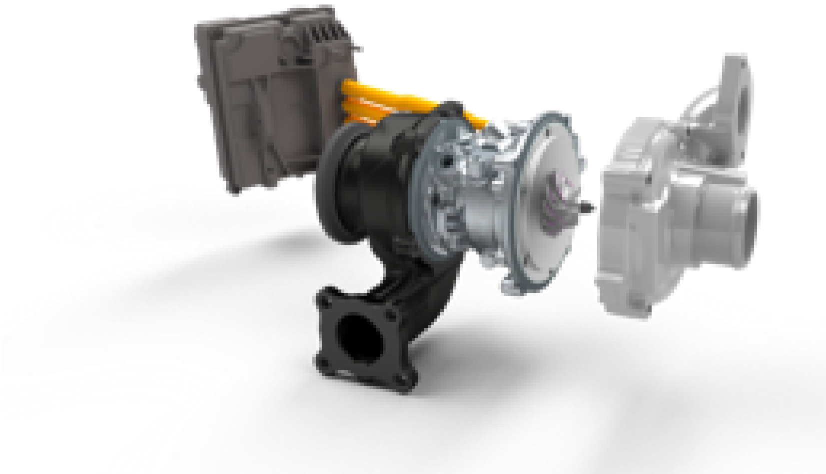

Figure 2.5: E-Turbocharger with dedicated control unit

2.3.3 After Treatment System

In order to anticipate the EURO 7 regulation entry, the PHEONICE engine has a brand new exhaust line, in compliance with the proposed limit by the European Commission[21]. This exhaust system is way more complicated than the one used in the stock GSE T4 because of the ultra lean combustion, that creates an environment inside the TWC not fit for a good NOx reduction efficiency.
The lean exhaust gases have an high oxidation efficiency, but the NOx reduction has to be done in a separate system: the exhaust line will have a *Selective Catalytic Reduction* system (SCR), usually used in Diesel vehicles after EURO 5 regulation. This system has a layout similar to the TWC, but it uses ammonia injections, coming from decomposed urea (that is safe for the health and easy to transport and refill), to catalyze reduction reactions of the NOx in a lean environment.
Coupled with the SCR, there is often an Ammonia Slip Catalyst (ASC), such as a component used to oxidize the ammonia, that makes possible the NOx reduction within the system, without any risk for the human health. The ammonia slip is the excess of ammonia used in SCR that can be found outside it. In fact, ammonia is highly dangerous and it has to be oxidized in NOx outside the system, if there is any residuals coming from the previous SCR reduction reactions.

Another important system is the *Gasoline Particulate Filter* (GPF), that has to block the high number of smallest particles of Particulate Matter that are generated by the Direct Injection in the chamber, that can create harm to health.

The cited systems, in order to operate correctly in pollutant reduction, have to reach a certain temperature. The PHOENICE exhaust line has an *electrically heated catalyst* (EHC) in order to reduce the light-off time of the whole system, limiting drastically the pollutant emissions during the engine startup phase.

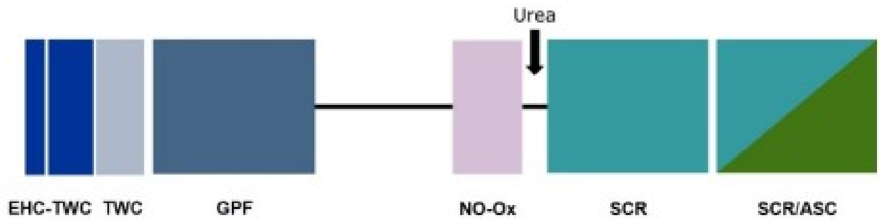

Figure 2.6: EURO 7 exhaust layout [14]

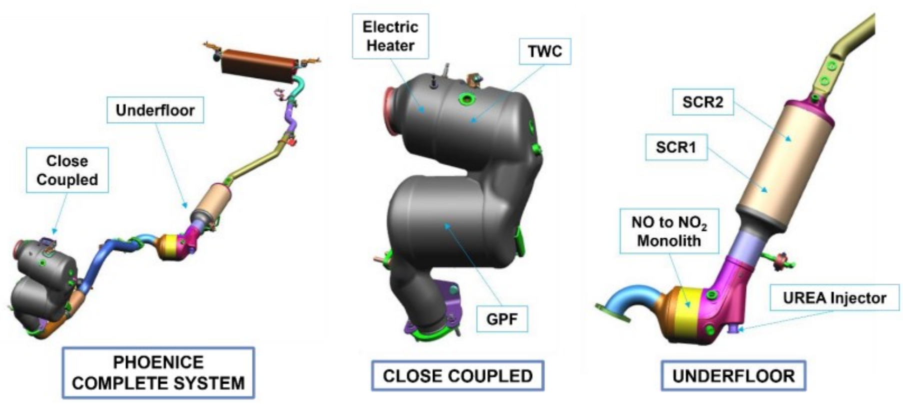

Figure 2.7: PHOENICE detailed exhaust line[14]

2.3.4 Dual Dilution Combustion

The Dual Dilution technique is optimal for the efficiency gain wanted on the modified GSE T4. It consists in performing the combustion in lean conditions ($\lambda > 1$) with EGR dilution.

The advantages of a lean combustion are lower temperatures leading to lower heat losses and to a knock suppression behaviour. The lower knock tendency allows to work with an advanced spark timing nearer the Maximum Brake Torque Timing (MBT Timing), increasing significantly the efficiency. Also, a leaner environment tends to oxidize more the pollutants, reducing the concentrations of HC and CO as emissions in exhaust gases while increasing the NOx emissions (if the lean mixture

is near $\lambda = 1.1$) that need to be chemically reduced and not oxidized. This happens because the temperature drop induced is not enough to compensate the increased production of NOx, due to the excess of air in the hot exhaust gases. If the mixture goes leaner, there will be a crossing point where the temperature is reduced enough (by the higher dilution) to decrease the NOx emissions with respect to the baseline conditions.
By performing a Dual Dilution, the temperatures are lower, so the emissions of NOx are way more limited (if $\lambda > 1.1$ + EGR) and the turbocharger can use a VNT solution to guarantee the best performance; there will be less heat losses too. The dilution given by excess air and EGR works as a really effective knock suppression strategy, so the combustion can be started with a more advanced spark timing in order to gain in efficiency and in a better combustion phasing.

The parameter that has to be checked is the Coefficient of Variation in percentage (CoV%), that is an important indicator of the stability of the combustion process. In both lean and leanest operations this number has to be below the 3% threshold, if it is under the 2%, smooth operating conditions[14] can be achieved.

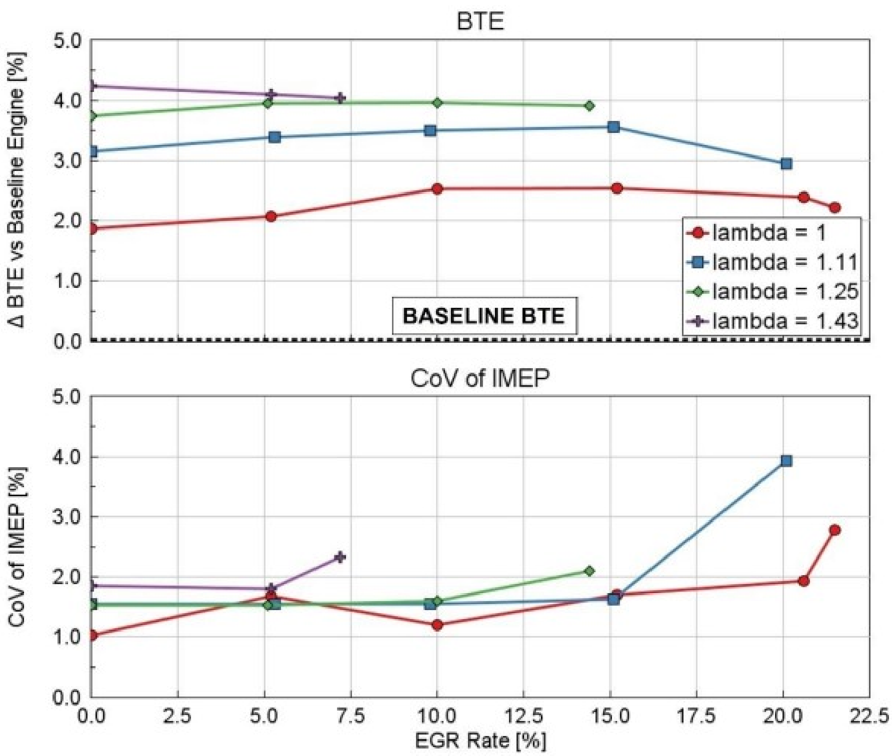

Figure 2.8: Dual Dilution effects on BTE and CoV% at 3000 RPM x 7 bar BMEP

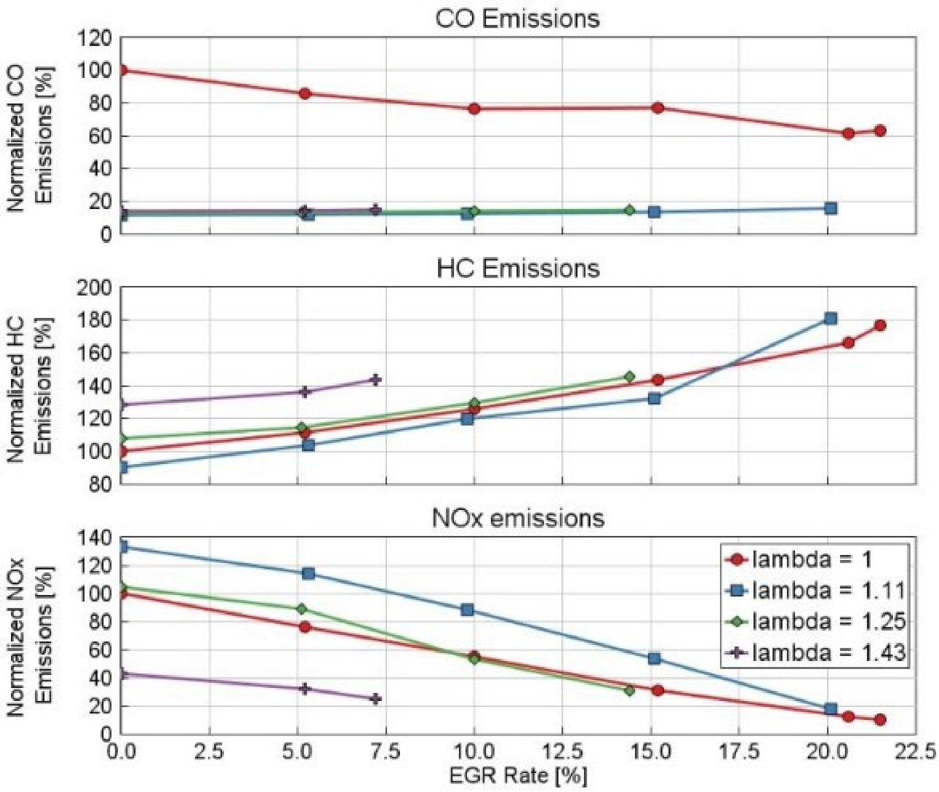

Figure 2.9: Dual Dilution effects on engine-out pollutants at 3000 RPM x 7 bar BMEP

2.3.5 Hybrid System

The GSE T4 engine is coupled with a BSG starter-generator in order to obtain a P0 configuration to increase the overall efficiency; this solution is implemented as well in the PHOENICE system.
The BSG is linked via belt with the engine crank and it replaces the standard starter. This solution cannot drive alone the vehicle due to the small electric motor and the low battery capacity, but it can help to balance the engine load during transient operations and recover energy with a regenerative braking. This energy is all stored and used in a auxiliary battery designed only for this purpose.

The hybrid part that can give an important contribution to the whole traction system is the P4 configuration, where a medium size electric motor (70kW peak power) and a bigger battery (near 11 kWh), mounted in the rear axle, that can achieve a 4x4 solution. The energy managed by the P0 will be transferred directly to the *HV battery* used by the P4 motor, in order to have a single medium-size battery.

It is important to state that the GSE T4 is coupled to a P0 - P4 hybrid system only in the *4xe* configuration of the vehicles made by Stellantis.

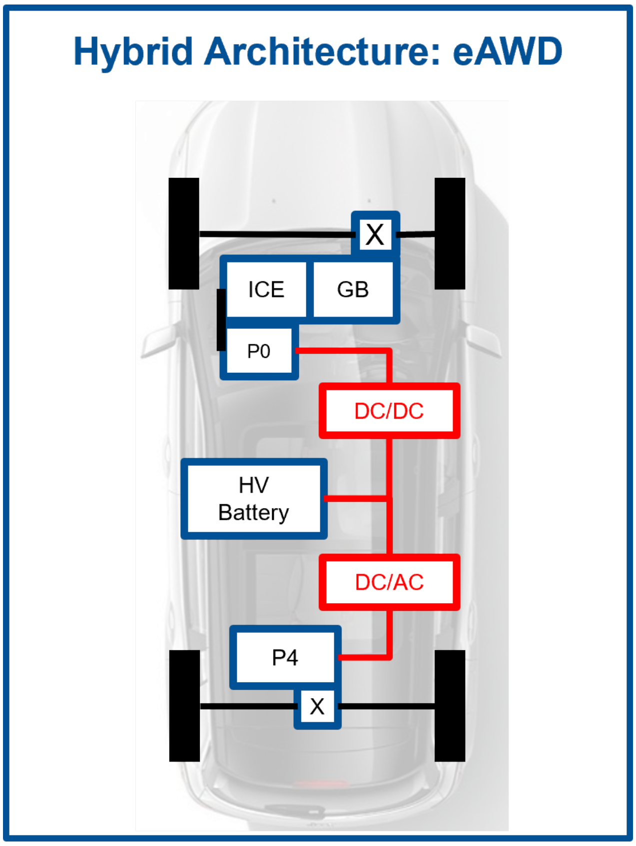

Figure 2.10: P0 and P4 layout of the GSE T4 for the 4xe configuration

Chapter 3

Experimental Setup and Test Bench

3.1 Overview

In order to perform tests and evaluations on the PHOENICE engine, a proper test-bed is needed. The *Dynamic Test-Bed* of the Politecnico di Torino is placed inside the DENERG facility and it has an AVL control unit.
The tests need a proper engine instrumentation, such as thermocouples, pressure sensors, encoders and gas analyzers.
The Dynamic Test-Bed has a control room where all the measuring software and the control panel are checked and used by the test-cell operators. This environment has a security glass window that allows to directly check the engine conditions and the two environment, such as the engine test bench one and the control one are isolated acoustically and thermally. There is only one door to access the engine side from the control room side.
In the other environment there is the engine and all the instruments that provide power, fuel, some pollutants analyzers and the acquisition boards to convert the sensors signals to software readable information.

3.2 Engine room

In this side of the test-bed, the engine is mounted on a *Dyno*, that controls the engine speed and load according to the user requests, depending on the test conditions.

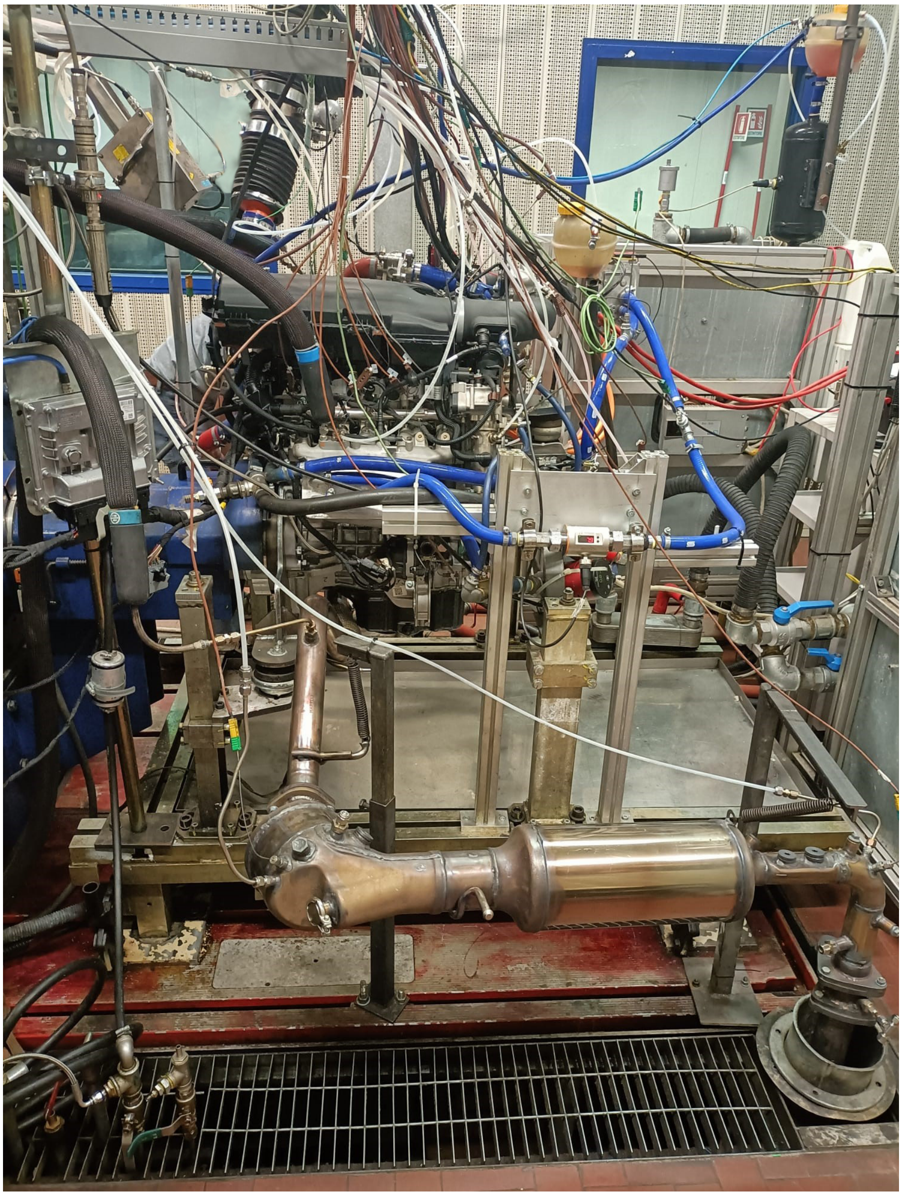

Figure 3.1: PHOENICE mounted on Dyno

The power to all the auxiliary systems, such as FEV control unit for the VNT control, E-Turbocharger 48V supply and the Stellantis ECU, is managed by Low and High Voltage Power Supplies.
There is a High Temperature Cooling Water line that provides water to cool down the engine water and oil, controlling their temperatures in order to guarantee a stable 90°C for the water and a maximum 110°C for the oil.
There is also a Low Temperature Cooling Water line that provides water to the

WCAC heat exchanger, in order to cool the compressed air that comes from the compressor to 30°C, if needed. This strategy can increase the volumetric efficiency in a significant way, because the air is more dense when it enters the cylinder.

The exhaust gases are expelled from the test-bed by means of a fan placed under the exhaust end pipe.
In order to avoid problems due to residuals in the gasoline (E5), the *KMA system* is used: it is capable of filtering, pressurizing at 4 bar and thermal controlling the fuel, while sending all the information to the Control Room.

The acquisition system is placed near the engine and it provides to convert and acquire all the different data coming from the sensors of the PHOENICE: it has an AVL X-ion board for the encoder and the pressure signals.

Inside the Engine room there is a Smokemeter too, useful to evaluate the PM concentration.
The pollutants evaluation is done by using the AVL AMA i60, that manage three acquisition lines with:

- a NDIR (Non-Dispersive Infrared Detectors) to measure the CO and CO_2 emissions, that can easily track the CO_2 concentration in the intake manifold in order to measure the EGR rate too;
- a FID (Flame Ionization Detector) in order to measure the HC concentration and type of HC (it may be Methane HC or Non-methane HC);
- a CLD (Chemi-Luminescence Detector) that can measure the NO_x, a very important parameter to evaluate the effect of the EGR rate on the PHOENICE engine emissions.

3.3 Control room

The conditions required by the engine during a test are imposed in the Control room. The main engine functions, for example the pedal %, the requested speed, requested torque, can be controlled by means of the control panel in the Control Room. Using this panel, the engine can start up, stay on idle or it can be fully controlled by the user. The cell preparation too can be started and monitored by the panel in order to achieve the controlled temperature and air quality, controlling humidity too.

The software used in the Control room are AVL Puma, that can manage the same parameters available in the control panel and it can communicate with other software in order to generate a single output file with all the acquired data. It can also acquire low frequency signals.
The Post-Processing of the data acquired in Puma can be done by means of AVL

Concerto, that is a software integrated within Puma that provides a lot of tools to plot properly all the results.

Another software is AVL IndiCom, suited to acquire high frequency signals, such as the pressure signals, and to calculate useful parameters, for example knock indexes. Pressure signals and work cycles per each cylinder can be displayed. LogP-LogV cycles can be plotted too, in order to graphically visualize if the Miller cycle is an Early or Late one.

Figure 3.2: Control Room layout

The software ETAS INCA is necessary to control and acquire the ECU signals. All the ECU parameters can be really important to check what actuation was performed during a certain test. With this software every ECU parameter can be read and modified according to the calibration used. To make an example, the turbine rack actuation, the EGR valve position and the VVA parameters (eIVO and eIVC) can also be imposed by user, granting full control on the engine features.

Chapter 4

Experimental Data Analysis

4.1 Engine #1 measurements

Tests on the PHOENICE engine are necessary to check and analyze the true capabilities of this configuration, discussed in Chapter 2.

In order to achieve the optimal combination between BTE and emissions, many sweeps have to be tested. The main experimental activity is to perform EGR sweeps in order to understand which EGR rate can possibly give the best solution.

The data that IFP has shared are related to a close-coupled-only exhaust line, that will produce a lower back-pressure than the entire EURO 7 exhaust line configuration, and to a reference configuration of fixed engine parameters per test working point.

The only parameter that has to change is the EGR rate. Its effects are related mostly to BTE, engine-out emissions and to the spark advance, that is the only working parameter that can be calibrated in order to reach the target BMEP of the test and achieve a stable combustion.

4.2 Data Analysis

EGR sweeps have been tested for a set of 11 key-points, in stoichiometric conditions ($\lambda = 1$) and in lean conditions ($\lambda = 1.1$).
This key-points are divided into Low, Medium, High and Full load groups in order to visualize a trend of the results in the engine for different load conditions. It is important to state that the KP are defined as **rpm** x **bar BMEP** for the whole analysis performed in this section. The points, from now on, will be called only with their values of rotational speed and BMEP for sake of simplicity, to make an example 2000x2 (2000 rpm x 2 bar BMEP).
In the following bar graphs the quantity of interest analyzed, such as BTE, NOx

and CoV can be read on the left side. Also, the line and dot plot is referred to the spark advance and it has to be read on the right side of the graphs.

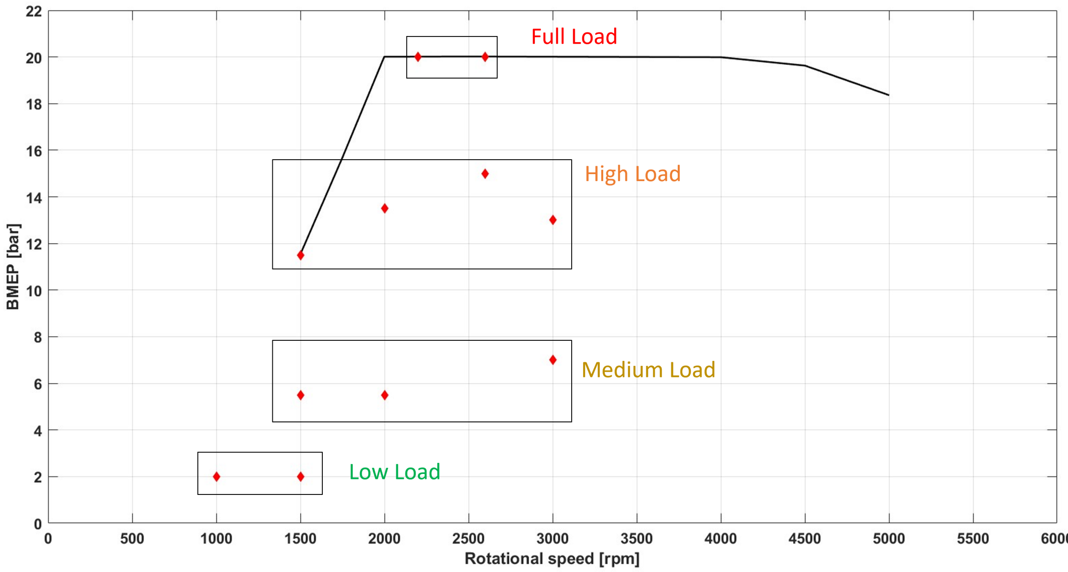

Figure 4.1: 11 key-points tested with Load intensity

4.2.1 Stoichiometric conditions

Low Load

The Low Load conditions analyzed are referred to two engine working point, such as the 1000 x 2 and the 1500 x 2. Each bar graph shows the values of BTE, NOx and CoV information, related to different EGR sweeps tested and the Spark Advance trend, plotted over each bar graph. The accepted maximum level of CoV in order to achieve a smooth operation of the engine is set to 3%, with an ideal target 2% at most.

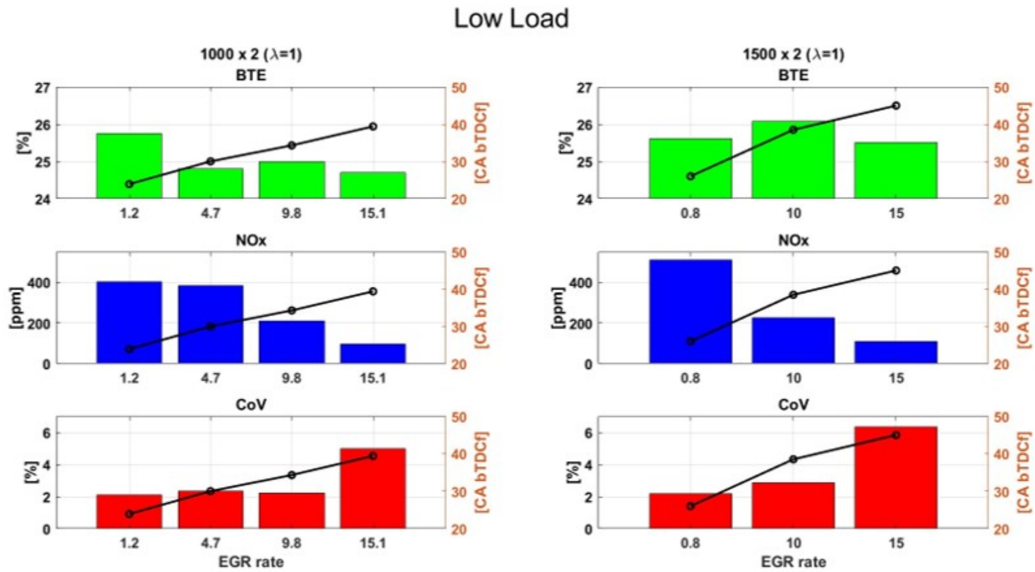

Figure 4.2: Low Load Stoichiometric EGR sweep

In the 1000 x 2 the best BTE result is achieved at 1.2 % EGR, but the NOx level is at its peak. The CoV shows decent values, it is not a concerning condition for the engine smoothness. An interesting compromise may be the 9.8% EGR point, but its BTE is too low and it cannot justify this solution, selected only to achieve almost an half of NOx emissions.

However, the 1500 x 2 shows the best solution at 10% EGR; the BTE is at its peak and the NOx emissions are an half of the 0.8% EGR point, but with a CoV lower than 3%. The best NOx emission condition is obviously at 15% EGR, but the lower BTE and the very high CoV make this choice inconvenient.

Medium Load

In Medium Load condition, three key-points are analyzed; they are the 1500 x 5.5, the 2000 x 5.5 and the 3000 x 7. During Medium Loads, the effect of the EGR can be very strong in enhancing BTE while reducing NOx emissions, the only problem may be the high CoV that an high dilution can generate, slowing down the combustion flame speed.
There are a lot of EGR sweeps for the 3000 x 7 because it is the most studied key-point of the whole PHOENICE Project so far.

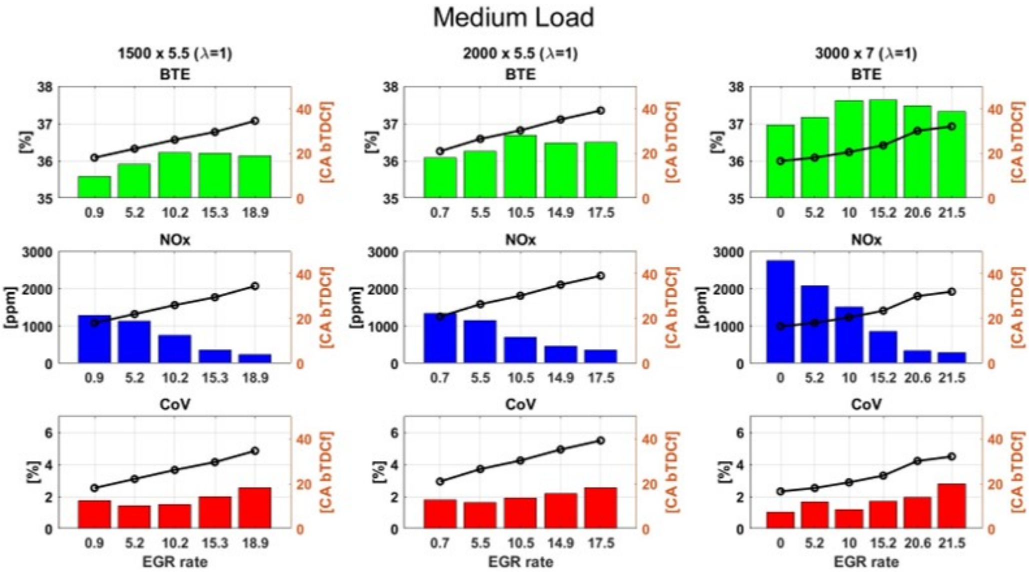

Figure 4.3: Medium Load Stoichiometric EGR sweep

The trend is clear in this load condition: the EGR rate near the 10% is the most suitable for the BTE-NOx compromise, providing acceptable CoV values. Low EGR rate values cannot work as a dethrottling strategy and they cannot reduce the NOx emissions aswell. Also, High EGR values cannot produce a BTE peak because the flame speed is too low, generating an high CoV value, even showing the lowest NOx emissions of the entire sweep. The knock mitigation property of the EGR is clear in the Spark Advance trend. In order to speed up the flame speed, it is necessary to increase the SA, operation allowed thanks to the lower temperature in the chamber due to the EGR dilution. The same effect can be seen in Low Load.

The key-point 3000 x 7, due to the high number of tests in $\lambda = 1$, $\lambda = 1.1$, $\lambda = 1.25$ and $\lambda = 1.43$, can be used as reference in the study of the BTE - NOx trade off. Each degree of dilution, starting from the stoichiometric operation, presents a different NOx behaviour respects the EGR rate, but the almost same BTE trend.

The leaner is the mixture, the lower will be the temperature, resulting in a NOx reduction. The $\lambda = 1.1$, however, has more air to oxidize the mixture into NOx, while the temperature are still high, so the NOx emissions will be higher than the $\lambda = 1$ conditions.

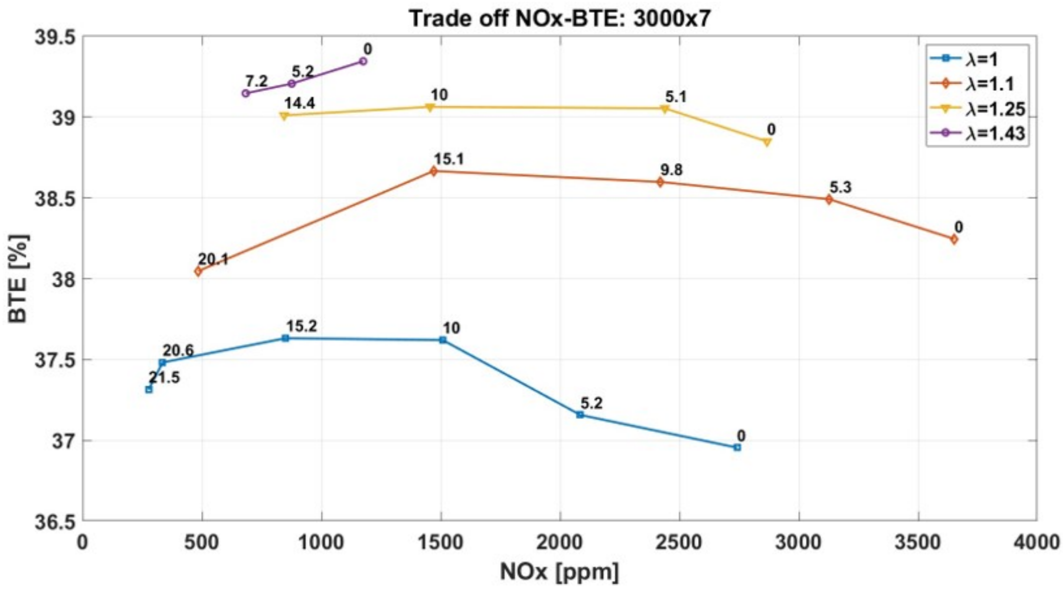

Figure 4.4: NOx-BTE trade off: 3000 x 7; tagged points represent EGR rates

High Load

Four key-points are related to the High Load category. This condition is linked to a fairly high value of BMEP. The EGR rate can be raised due to the faster flame speed and the higher combustion stability.

The key-points are the 1500 x 11.5, the 2000 x 13.5, the 2600 x 15 and the 3000 x 13.

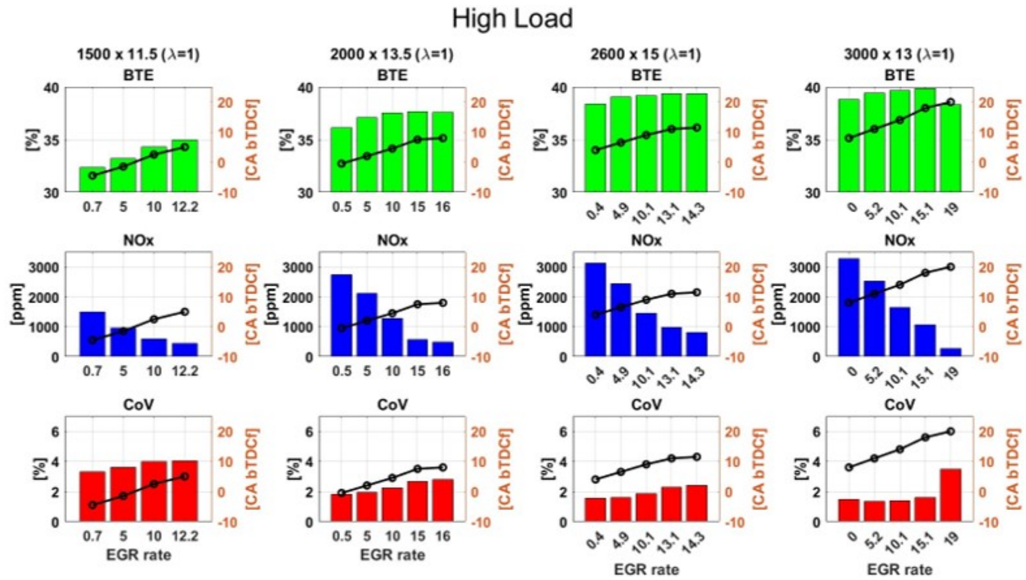

Figure 4.5: High Load Stoichiometric EGR sweep

The best EGR value is over 10%, near 15% in the 2000 x 13.5 and in the 3000 x 13, as expected. In this conditions the NOx reduction is very consistent and the CoV values are all below the 3% limit. The only exception is the 1500 x 11.5, that is a difficult point for combustion stability, showing an high CoV even at 0% EGR. As mentioned before, losing some BTE for a NOx optimization, in this particular condition, is not an acceptable trade-off.

The maximum value of the BTE is near 40%, that is an impressive result for this layout, considering that it is a stoichiometric combustion.

Full Load

During Full Load operations, in order to achieve the maximum power, it is expected a low EGR dilution because the chamber volume has to be almost fully filled with the mixture to achieve an high BMEP value.

However, the PHOENICE engine is limited at 20 bar maximum BMEP by design, but the baseline engine is capable of an higher peak value. So, there is room for a good EGR dilution.

The full load key-points are the 2200 x 20 and the 2600 x 20.

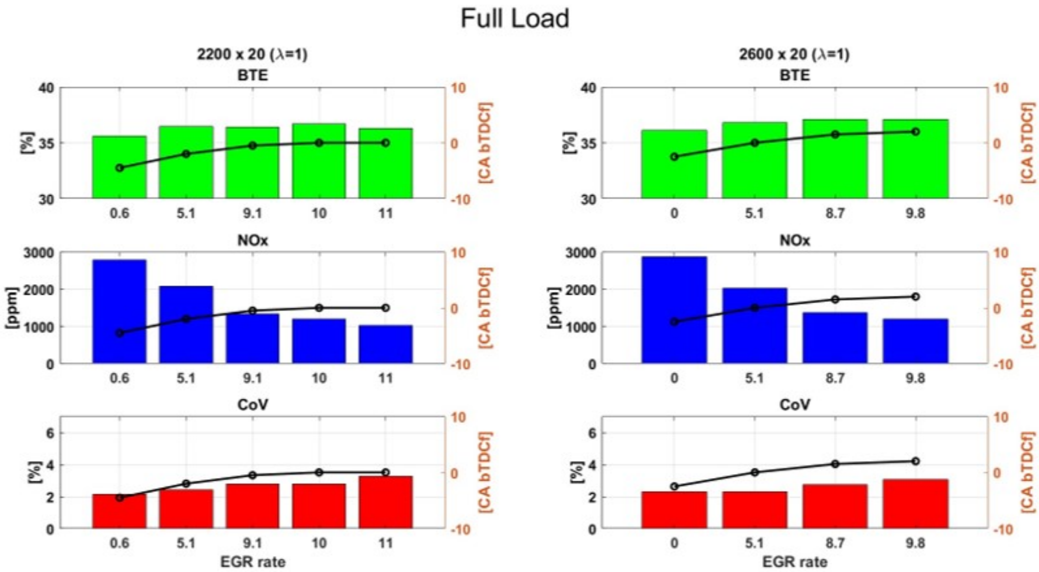

Figure 4.6: Full Load Stoichiometric EGR sweep

For the 2200 x 20, the peak BTE is at 10% EGR rate, with a NOx emission less than an half of the 0.6% EGR condition and a CoV lower than 3%. As expected, the EGR dilution rate is not as high as the Medium Load case. The Spark Advance trend is the same, in fact the SA will increase when the EGR rate raises, but its optimal value is near 0 because the high pressure in the chamber can easily generate knock; the SA is negative in 0.6% EGR, this means that the spark happens after the TDCf.

For the 2600 x 20, the best solution for the BTE can be found at 8.7% EGR rate, for the exactly same reasons of the previous key-point. Also 9.8% is good and it is different from the 8.7% by a net 0.15% ΔBTE: this difference is so small that is convenient to work at 9,8%, in order to achieve the better NOx reduction. The CoV is pretty the same in both conditions.

4.2.2 Lean conditions

All the tests are made with a lean mixture with $\lambda = 1.1$; it is the only mixture sweep value available for the each key-point tested. Other lean grades were tested as well for different KP, but in this analysis, in order to understand the differences between stoichiometric and lean combustion, they are not considered, even if the BTE, NOx reduction and CoV for leaner mixture may be far better than the cases analyzed in the following part of the work.
In fact, a key-point that has a great number of sweeps is the 3000 x 7, that was tested with lean $\lambda = 1.1$, $\lambda = 1.25$ and $\lambda = 1.43$, as showed in Figure 4.4. Similar

results can be observed for the other KP tested in lean condition sweeps (from $\lambda = 1.1$ to $\lambda = 1.43$), where possible in terms of combustion stability. The key-points related to each Load category are the same of the stoichiometric analysis, in order to compare different working conditions.
NOx emissions are expected to be higher than the stoichiometric conditions, in the same key-point at the nearly same EGR rate, because there is more air with fairly high temperature in chamber. It is a characteristic NOx behaviour linked only to the $\lambda = 1.1$ condition, as explained before. In fact, by increasing the lean grade for $\lambda > 1.1$, the NOx emissions abatement is stronger.

Low Load

The 1000 x 2 key-point shows a BTE peak at 10% EGR rate, very low NOx emission, but the CoV value is far higher than the 3% limit. In this conditions, by selecting the 1.3% EGR rate, BTE maximization can be achieved, but by selecting 5.1% EGR rate, a better NOx reduction can be achieved. Analyzing the benefits on NOx reduction, it can be seen that from 616 ppm the NOx emission drops to 447 ppm, but there is a net 0.68% ΔBTE loss. The BTE, in this case, has to be chosen as primary parameter because the SCR can reduce by itself this low difference in NOx quantity.

The 1500 x 2, however, shows a trend easy to analyze because at 4.9% EGR there is the highest BTE, the lowest NOx emission with a CoV of 3%.

As expected, the dilution given by the air, in lean conditions, helps with the BTE enhancement. In fact, these maximum values are achieved using a reduced EGR rate respects the stoichiometric conditions. The dual dilution, however, slows down the combustion process, increasing the overall values of CoV and works as knock suppression strategy, allowing an advanced Spark Advance.

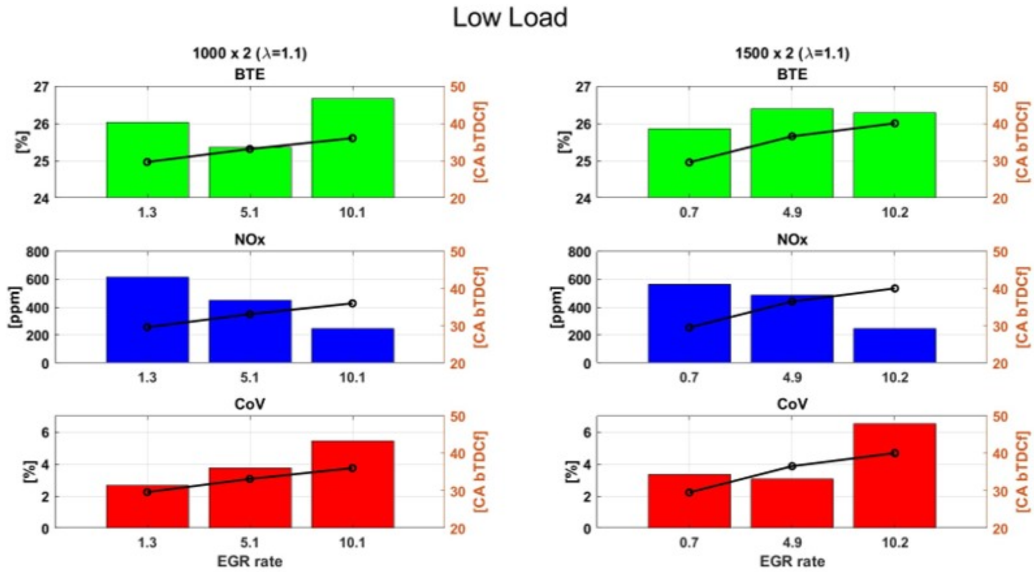

Figure 4.7: Low Load Lean EGR sweep

Medium Load

The Medium Load condition shows an overall higher BTE maximum value, as expected.

The 1500 x 5.5 key-point shows an optimal EGR rate at 5.3%, with a peak BTE of 37.37%, a NOx emission of 1161 ppm and a CoV of 2%. The only other interesting EGR rate value may be the 15% EGR, with a reduced BTE of 37.26%, but half of the NOx emissions; the CoV, however, has a value of 3.6%, that may be too far from the 3% limit.

The 3000 x 7 key-point shows an optimal EGR rate of 15.1%, achieving a maximum BTE of 38.66%, a very low NOx emission (less than a half of the 0% EGR rate conditions) with a CoV of 1.63%.

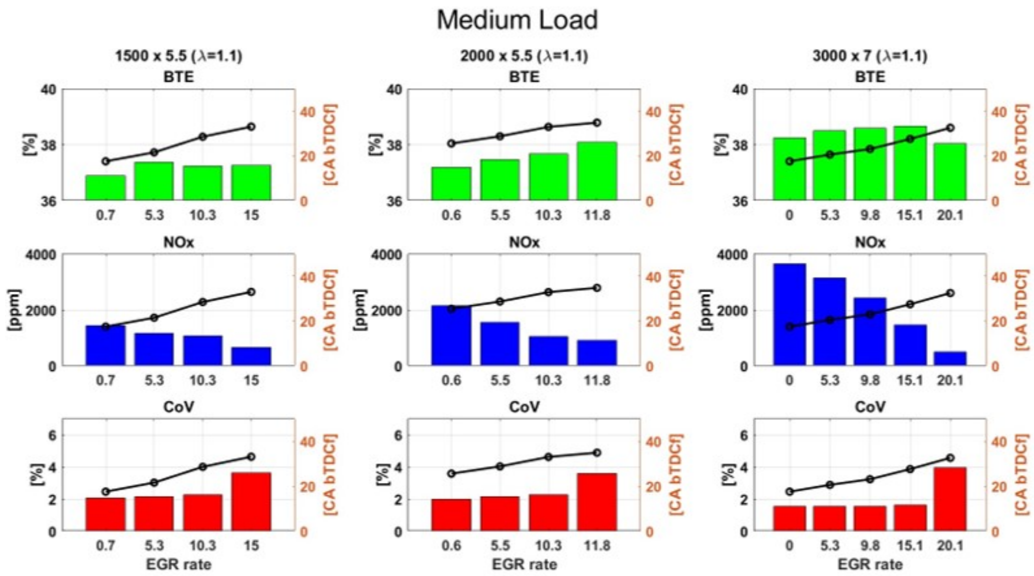

Figure 4.8: Medium Load Lean EGR sweep

High Load

The key-points analyzed in the High Load condition are, as shown before, the 1500 x 11.5, the 2000 x 13.5, the 2600 x 15 and the 3000 x 13.

The 1500 x 11.5 is a complex load point, as it has already in stoichiometric conditions strong limits due to the high CoV. For this reason, in Lean conditions too it can run only with a 0.7% EGR rate. Increasing the EGR rate will only raise the CoV value to unacceptable levels, making this solution unusable, no matter what the BTE or NOx results are.

The 2000 x 13.5 has two EGR sweeps, that are very close to each other; the net BTE difference is only of 0.16%, but the NOx reduction achieved at 5% EGR rate is remarkable. The CoV in both 0.5% and 5% EGR rate in below the 3% limits, so it may be the best choice to lose a very little amount of BTE to enhance the NOx reduction by working at 5% EGR.

The 2600 x 15 too has two BTE results very close, but this time the 5% EGR rate shows the best results, with the highest BTE and the lowest NOx emission. The CoV in both 0% and 5% EGR is lower than 3% and the difference between this two value is slight, with a value of 0.4%.

The 3000 x 13 is an interesting point because the BTE reaches values over 40%, with EGR rate higher than 5%. In this situation, it is optimal to chose the 5.3% EGR rate in order to achieve high BTE, low NOx and a CoV lower than 3%, very near to the optimal 2% target for optimal stability. The best results are at 9.4% EGR rate, but the high CoV makes this combination not fit for the engine map building.

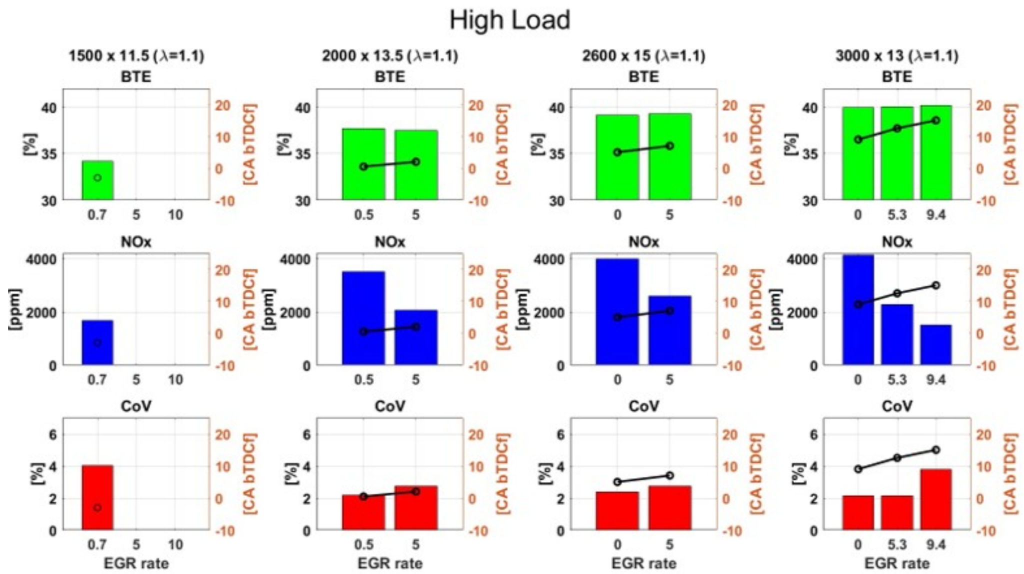

Figure 4.9: High Load Lean EGR sweep

Full Load

The Full Load operations are related to the key-points 2200 x 20 and 2600 x 20. This conditions are very difficult to stabilize, with both EGR and Lean mixture, because of the flame speed instabilities that the dual dilution can introduce.
The two configuration shown in the graph are linked to one sweep only at nearly 0% EGR rate for this reason. The CoV obtained, higher than the 3% limit, is enough to justify the limit that an higher EGR dilution cannot overcome, making other sweeps unnecessary.

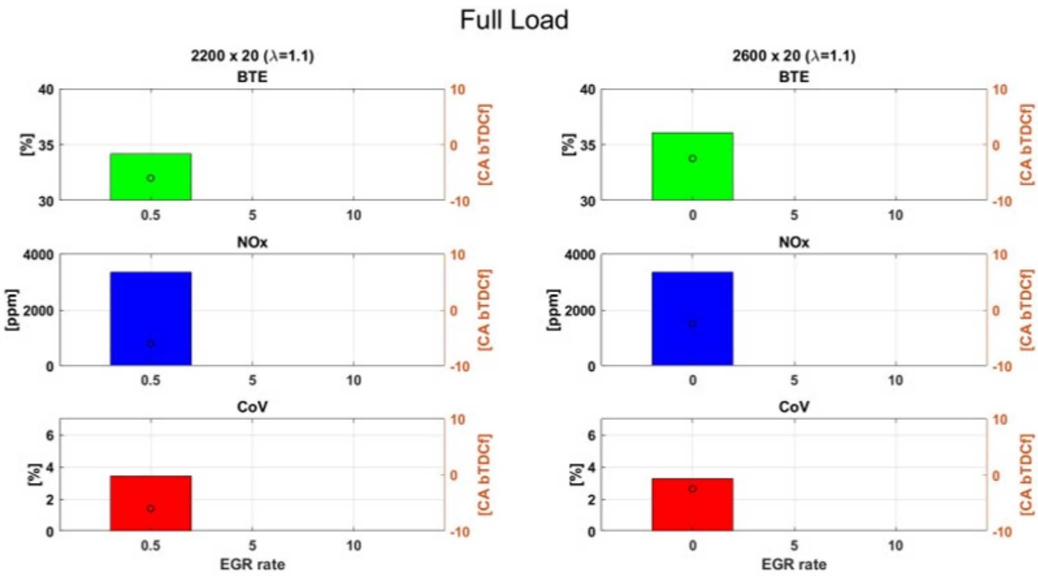

Figure 4.10: Full Load Lean EGR sweep

4.2.3 Key-Points Optimization

The analysis done before is crucial to evaluate the importance of the EGR dilution at different mixture conditions, both stoichiometric and lean, for different categories of Load type. In order to understand how each single key-point can be optimized, the following discussion is based on the assessment of the best BTE, NOx and CoV combination for one tested working point at a time. The CoV limitations are exactly the same of the previous analysis, with a preferred maximum CoV of 2% and a limit CoV of 3% tolerated. As stated before, the analysis is performed only for $\lambda = 1$ and $\lambda = 1.1$, because the information linked to these conditions are available for each KP tested, allowing to better assess the influence of each parameter on different optimization combinations. However, the overall best performance achievable may be reached for different lean conditions than the one studied.

1000 x 2

This key-point is tested in both $\lambda = 1$ and $\lambda = 1.1$ conditions, as the majority of the points analyzed, as stated before. It can be seen that the peak efficiency at near 1% EGR is reached in lean conditions. In order to minimize the emissions an higher percentage of EGR can be adopted, in fact this point has a tested condition of 5% and 10%. The overall maximum efficiency it achieved at lean condition with 10.1% EGR, but the very high CoV is a huge limitation. The 5% EGR conditions are the worst, the efficiency is low in both $\lambda = 1$ and $\lambda = 1.1$, and there is a really small gain in the emissions reduction, so the best solution can be seen at 1% EGR. The highest efficiency achievable with a good CoV is 26.03% in lean conditions, but the NOx increment with respect to the stoichiometric conditions, due to the high concentration of the oxygen in chamber, may not be acceptable. The baseline stoichiometric 1.2% EGR is the best trade-off between all the three parameters to optimize.

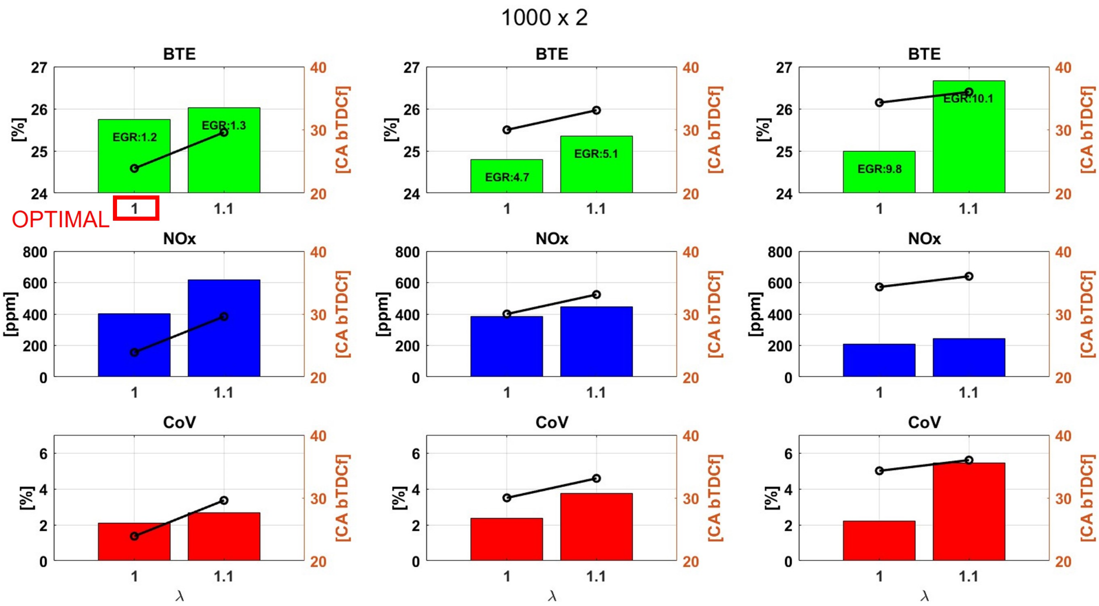

Figure 4.11: Analyzed sweeps for KP 1000 x 2

1500 x 2

This KP is tested in stoichiometric and lean conditions with an EGR rate in this case approximately of 0% and 10%. The best results for BTE, NOx and CoV can be seen in stoichiometric conditions with 10% EGR, in fact the dilution given by the EGR can enhance the efficiency and reduce the NOx emissions. The lean condition choice, at 10% EGR, leads to a very high CoV level, but with better BTE and the same NOx reduction of the $\lambda = 1$. So, even if the efficiency is lower than the $\lambda = 1.1$, the lower CoV in stoichiometric conditions is the main parameter to set the optimum results for this key-point.

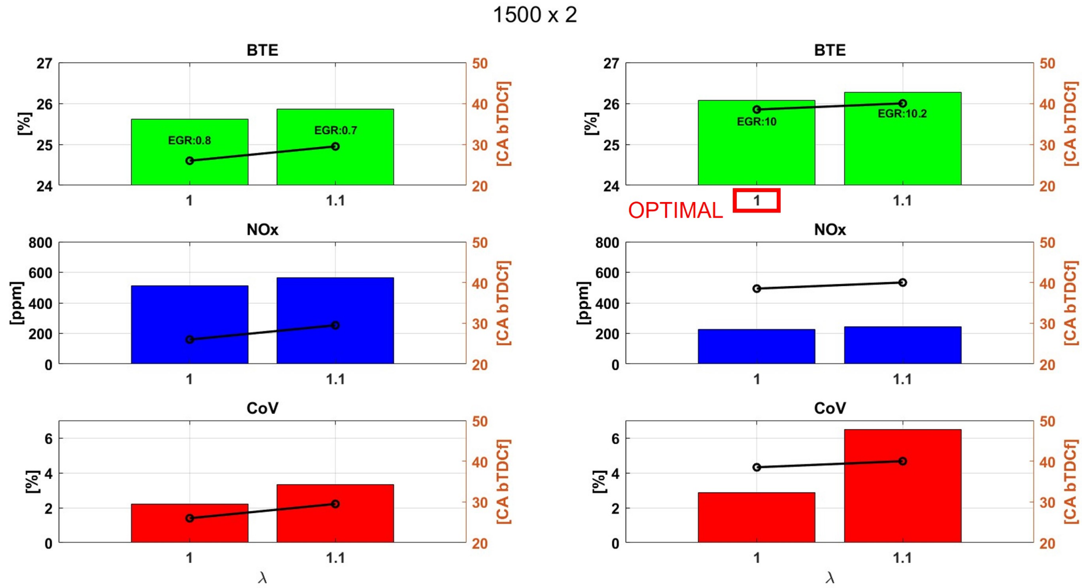

Figure 4.12: Analyzed sweeps for KP 1500 x 2

1500 x 5.5

The 1500x5.5 is a very interesting point because there are a lot of combinations to achieve the best result. This case is run with 0%, 5.2%, 10.2% and 15.3% EGR rate, so the EGR rate effects on the main parameters to optimize are clear. The highest efficiency of 37.37% can be seen at 5% EGR rate in lean conditions, but very high values of BTE are in the 10.3% and 15.3% too. So, looking only at the BTE, the optimal choice is to set the lean 5% EGR rate condition as the best operating combination, but the NOx emissions are still very similar to the baseline conditions. To reduce the NOx, keeping a CoV acceptable, it may be good to chose the 10.3% EGR in lean conditions, with a relative efficiency loss of 4.6% and a NOx reduction of 25%, with a CoV of 2.25%.

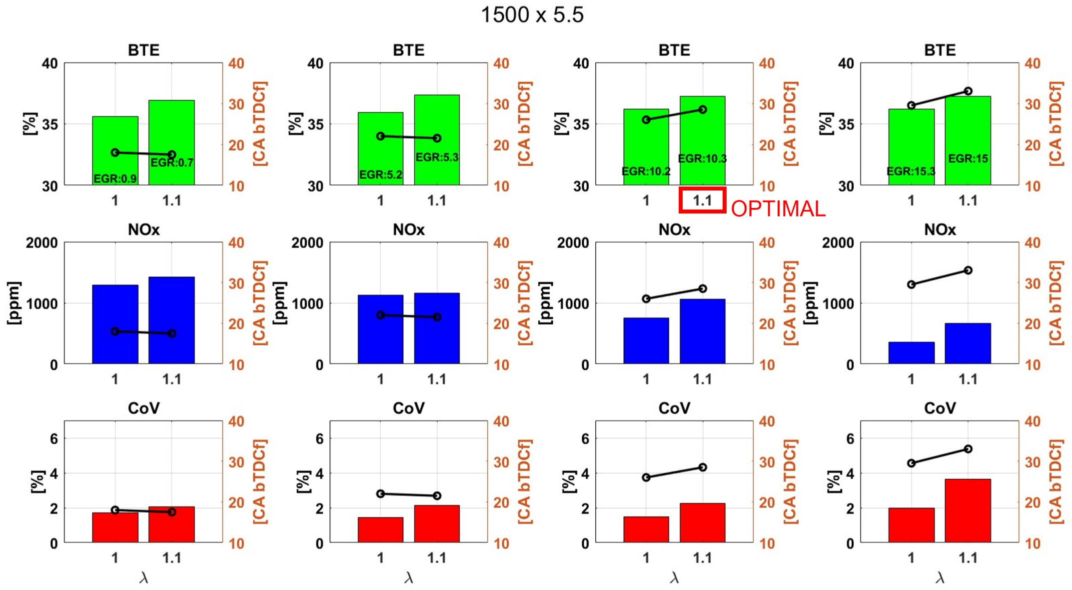

Figure 4.13: Analyzed sweeps for KP 1500 x 5.5

2000x5.5

The 2000x5.5 is very similar to the previous KP, in fact it has four different EGR rate levels. The best combination in this case can be seen at 10% EGR in lean conditions, with a CoV of 2.275% and a BTE of 37.66% (relative +4.4% respects the baseline 0% EGR at $\lambda = 1$) with a NOx reduction of 27.6%. The 5% EGR and the 10% EGR are very close in terms of BTE and CoV, but the NOx reduction achieved with an higher residuals rate is significant and justifies the choice made for the optimization.

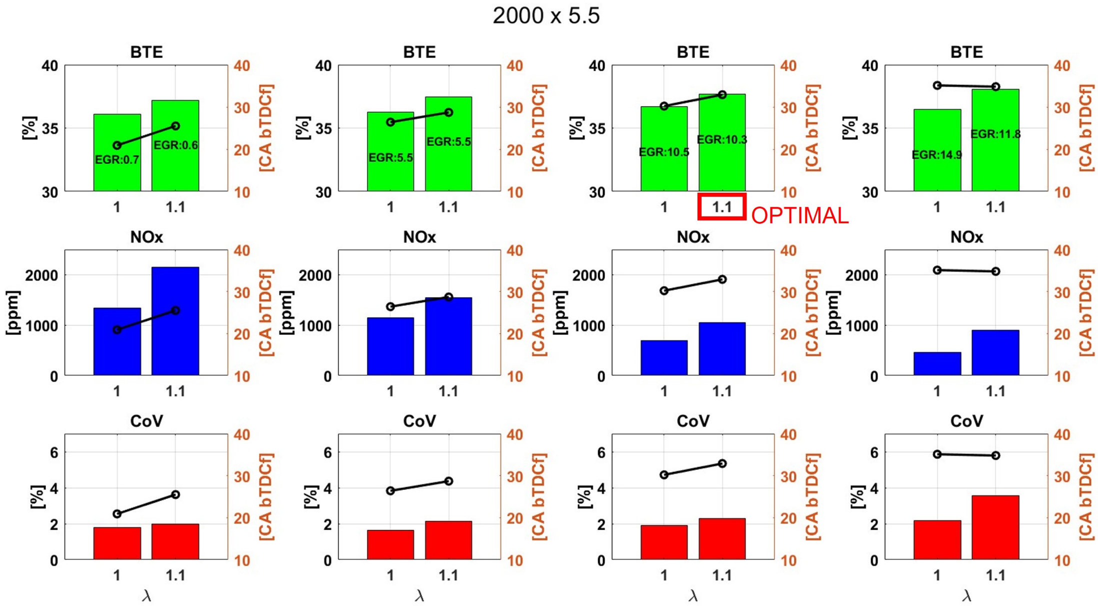

Figure 4.14: Analyzed sweeps for KP 2000 x 5.5

3000x7

In the tests performed to evaluate the potentiality of the PHOENICE, this KP is the most studied, with different lean conditions sweep (from $\lambda = 1.1$ up to $\lambda = 1.43$) and with very high EGR rates, up to 20%. The evaluation of the best combinations to achieve the optimum is performed considering only the $\lambda = 1$ and $\lambda = 1.1$ sweeps, because the EGR rate obtained at higher lean levels is much different from the EGR rate examined in this section, making a comparison very hard. However, the effects on the BTE and NOx can be seen in Figure 4.4. The best combination can be seen in lean conditions at 15% EGR rate, in fact the CoV level is good, with its value of 1.625%, the BTE is at its highest value of 38.6% with a 47,4% NOx reduction. The lean combinations at 5% and 10% are really good too in terms of BTE and CoV, but the lower residuals in chamber increase the NOx levels.

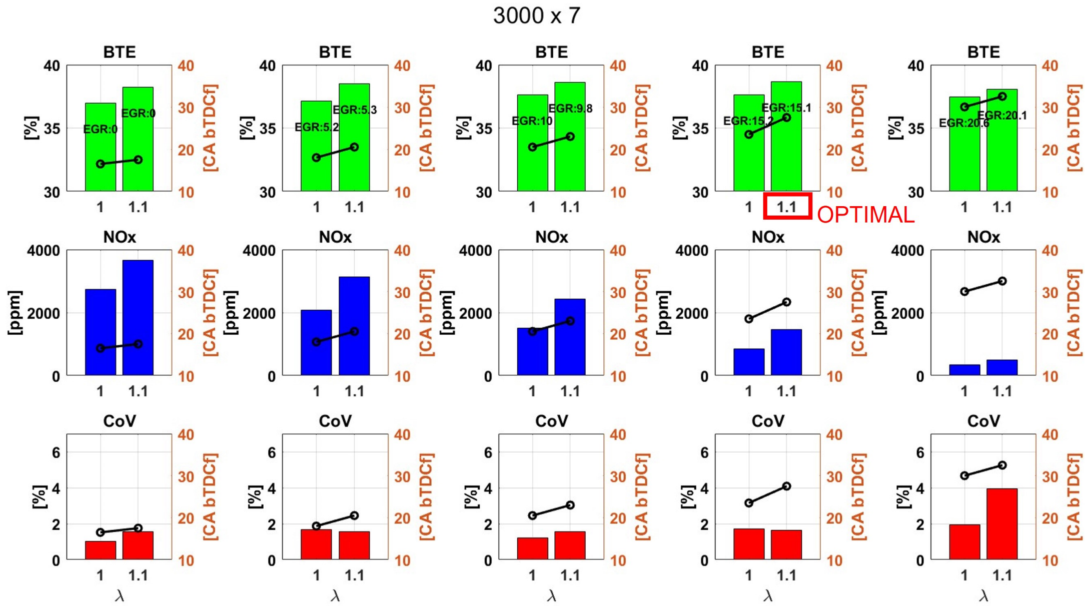

Figure 4.15: Analyzed sweeps for KP 3000 x 7

1500x11.5

The 1500x11.5 is a really unstable key-point, in fact the EGR rate tested is nearly zero (0.7%). The very high CoV at zero EGR makes the dilution very difficult to be performed with a stable combustion, in both stoichiometric and lean conditions. In lean conditions the CoV is still very high, over the 3% threshold, even if the BTE is the highest, this combination cannot be implemented. So, the combustion stability, represented by the CoV, is the main choice parameter to optimize a KP.

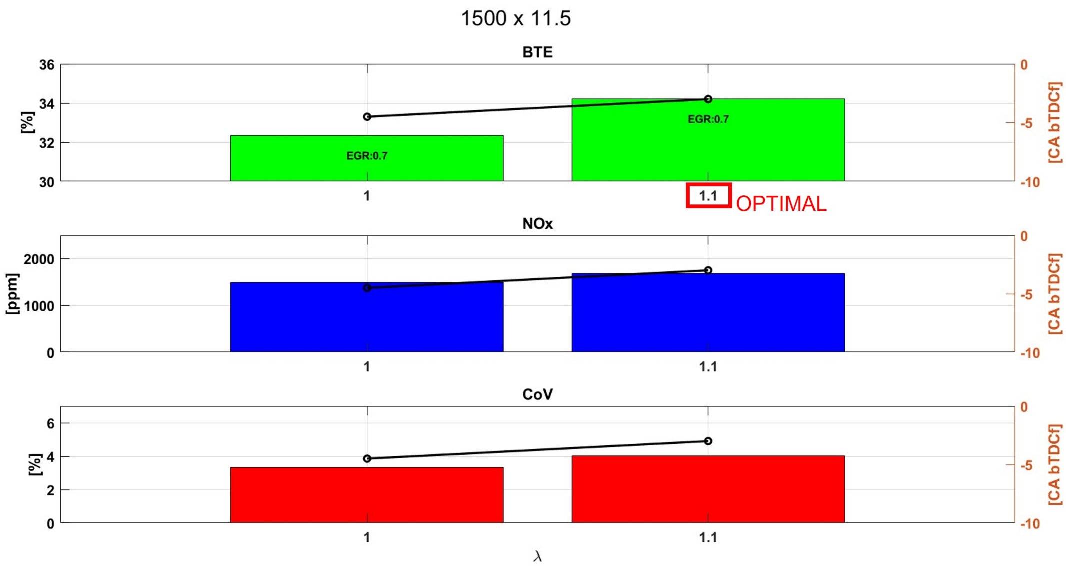

Figure 4.16: Analyzed sweeps for KP 1500 x 11.5

2000x13.5

In high load conditions, this KP is very important because it is frequently tested in homologation test cycles. The higher loads need more air to obtain a certain BMEP, so the volume available to perform EGR dilution is lower. In fact, the EGR rates tested are 0% and 5%. The best efficiency can be seen at 0% EGR in lean conditions, but the higher NOx produced in this conditions are not tolerable, leading to an emission increase of 29%. The best solution is in lean conditions at 5% EGR rate, with a good CoV, lower than 3%, with a BTE of 37.5% and a NOx reduction of 25%.

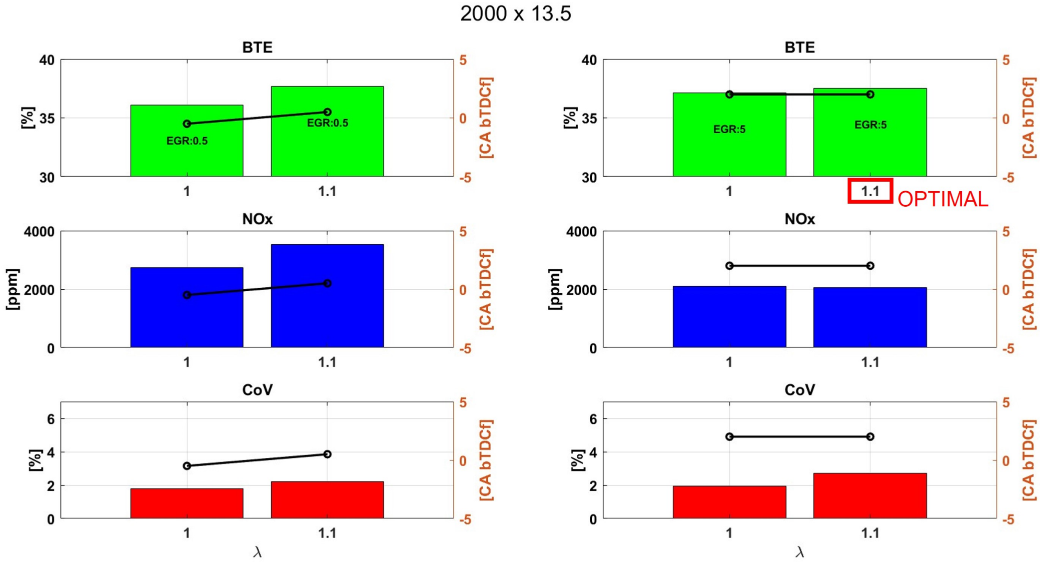

Figure 4.17: Analyzed sweeps for KP 2000 x 13.5

2600x15

For the same reasons of the 2000x13.5, in this KP the EGR rate is up to 5%. In lean conditions with 5% EGR, the BTE has its maximum at 39.29%, with a tolerable CoV (2.73%) and a NOx reduction of 16.7% respects the baseline stoichiometric with 0% EGR rate. If a lower CoV is preferred, the 5% EGR in stoichiometric conditions too can be a good choice because the NOx emissions are nearly the same, but the BTE is slightly lower, with a value of 39.06%.

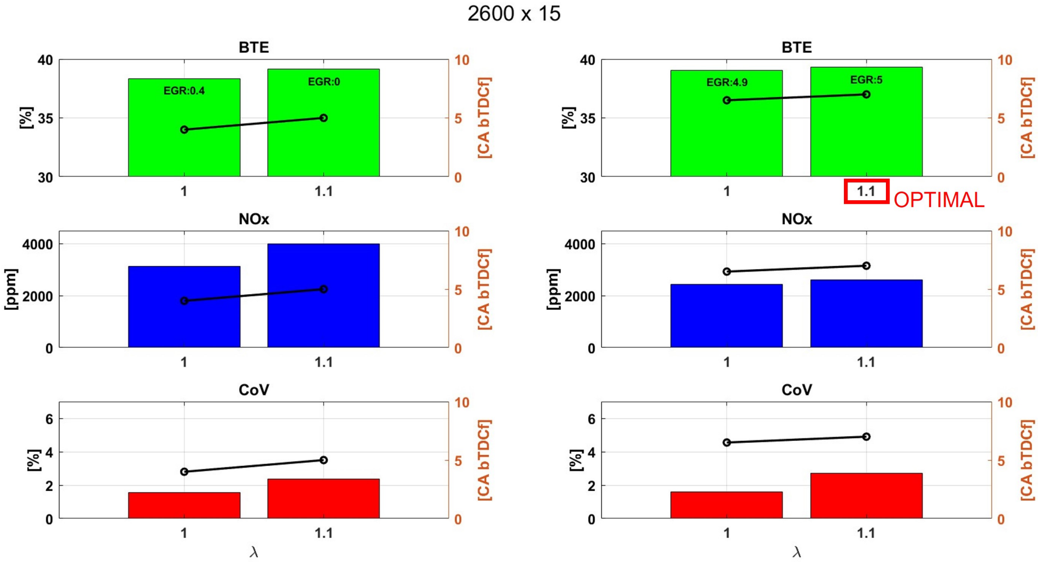

Figure 4.18: Analyzed sweeps for KP 2600 x 15

3000x13

The last high load analyzed point is the 3000x13; in this case the EGR rate is tested up to 10% in order to analyze the maximum tolerable dilution in this conditions. In fact, the lean 10% EGR has a CoV of 3.775%, preventing this solution to be implemented due to combustion stability issues, even if the BTE and the NOx are at their best conditions (40.14% peak BTE and 54% NOx reduction achieved). It can be seen that at 5% EGR rate, in lean conditions, the BTE is nearly the same of the 10% EGR, with a value of 40.06% and a NOx reduction of 30%, with stable combustion at 2.15% CoV. This is the best combination achievable, but if the NOx is the main optimization parameter, another solution that can be implemented is the stoichiometric 10% EGR, with a good BTE (39.68%), a very low CoV of 1.4% and a NOx reduction of 49.6%.

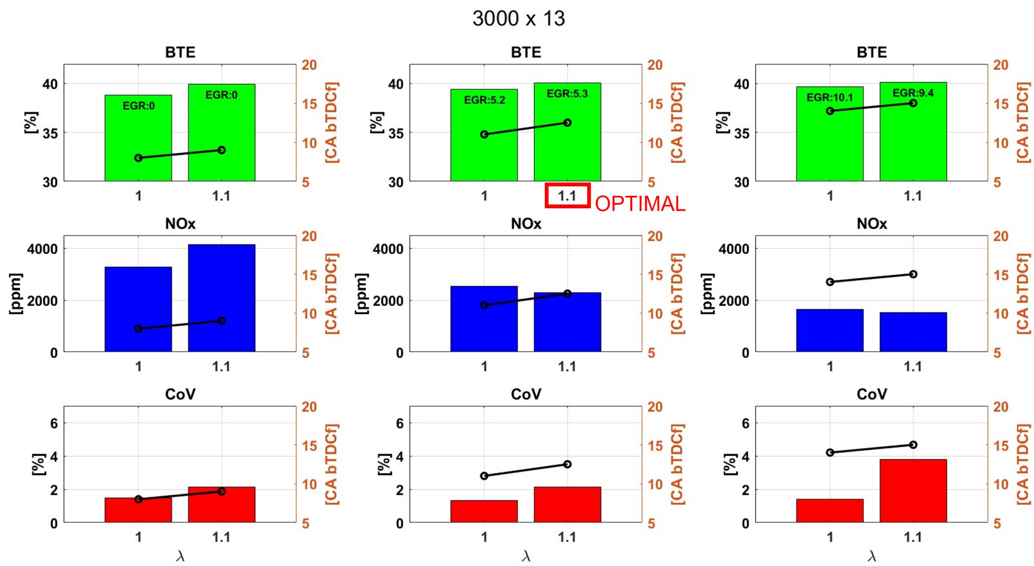

Figure 4.19: Analyzed sweeps for KP 3000 x 13

2200x20

At full load conditions, the expected EGR rate of the engine is nearly zero, because the large amount of air and the Miller cycle require a lot of available chamber volume in order to reach 20 bar of BMEP. In fact, the λ sweep tested is at 0.5% EGR rate, highlighting the benefits of a stoichiometric combustion that requires less air and grant a more stable combustion, peaking in all BTE, NOx reduction and CoV values. So, it is clear that the best combination is 0.5% EGR at stoichiometric conditions.

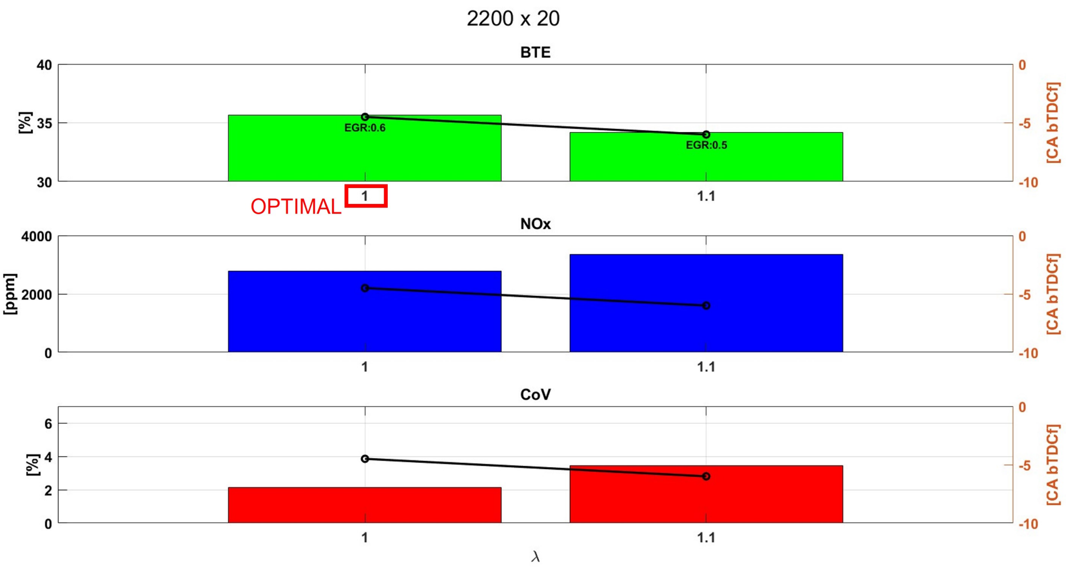

Figure 4.20: Analyzed sweeps for KP 2200 x 20

2600x20

The last full load KP is the 2600x20 that shows the same trend of the previous one. The necessity to stay in stoichiometric conditions without EGR dilution, in order to reach the 20 bar BMEP target, is evident for this case. In this conditions the BTE shows a value of 36.12% with a CoV of 2.3% and a NOx emission of 2874 ppm.

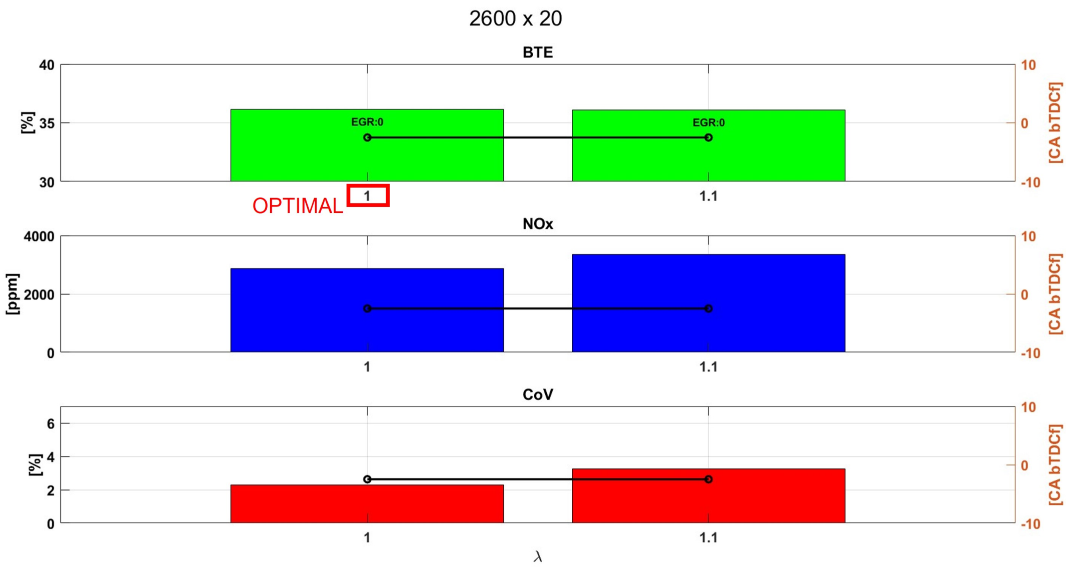

Figure 4.21: Analyzed sweeps for KP 2600 x 20

Chapter 5

Numerical Simulation

5.1 Motivations and Objectives

The analyzed data, acquired by experimental means, can be used to set up a *Digital Twin* in order to perform a complete analysis, obtaining information on parameters that cannot be measured in a test-bed.

Numerical Simulation is the only way to achieve an extremely detailed analysis of a complex system, using all the available physics equations to solve with accuracy the pressure and thermal field of each component.
The model setup, however, needs a calibration, once it is modeled, to correctly simulate the behaviour of the system. This calibration can only comes from experimental data acquired with different tests, taking into account only the most representative working conditions.

The goal of the Numerical Simulation is to compare the performance of the PHOENICE engine respects the baseline configuration, highlighting the importance of each solution implemented in the final setup. In order to correctly perform this evaluation, these solutions have to be modeled and calibrated properly and then introduced in the Complete PHOENICE GT model.
In particular, the EURO 7 exhaust and the turbulence model have to be coherently modeled and tuned.

The GT model used in this work was built and verified by previous students. The SITurb model was calibrated using both diluted and non-diluted stoichiometric combustion, with pretty good results in lean conditions too.
However, this old model has a previous version of the final EURO 7 exhaust line, which is missing all the underfloor part of the ATS, such as NO-Ox, SCR and ASC catalysts. So, it has to be validated again after the implementation of all the new calibrated features.

5.2 Exhaust Line Tuning

In order to perform a correct simulation, the actual back-pressures have to be predicted coherently in the exhaust line. If this condition is not verified, all the data obtained, such as pumping losses of the engine, efficiency and the combustion behaviour too, will not be representative of the actual engine conditions.

In the test-bed, the correct back-pressures at the turbine exit were calibrated experimentally starting from the flow tests made by Marelli, that designed and manufactured the complete exhaust line.
Using the flow tests of the bricks, a preliminary calibration on the pressure losses (using a discharge coefficient and a friction multiplier) was performed for each brick. Then all the bricks were added in the complete exhaust line and simulated to check the numerical overall pressure drop. The comparison between the experimental data and the simulation data shows substantial differences, implying that a fine tuning is required, in order to achieve the same experimental results .
The calibration process was performed using the *Integrated Design Optimizer* of GT-Suite. The optimization algorithm chosen is the Genetic Algorithm with 20 generations in a user-imposed lower and upper parameters sweep boundaries, with a Target type Objective Function set on the objective back-pressure for each calibration point.

The model implemented in GT to perform a fine tuning is a flux model where the fluid used is the air in order to reduce its complexity, because the results obtained are very similar to the ones simulated with exhaust gases as reference fluid.

The PHOENICE exhaust line is a new system in compliance with the EURO 7 regulations, so its GT model needs a fine tuning. The objective of this tuning is also to obtain at least information on the order of magnitude of the pressure losses induced by the additional bricks, such as the NO-Ox, SCR and ASC, that are not present in the production GSE T4 EURO 6 exhaust.

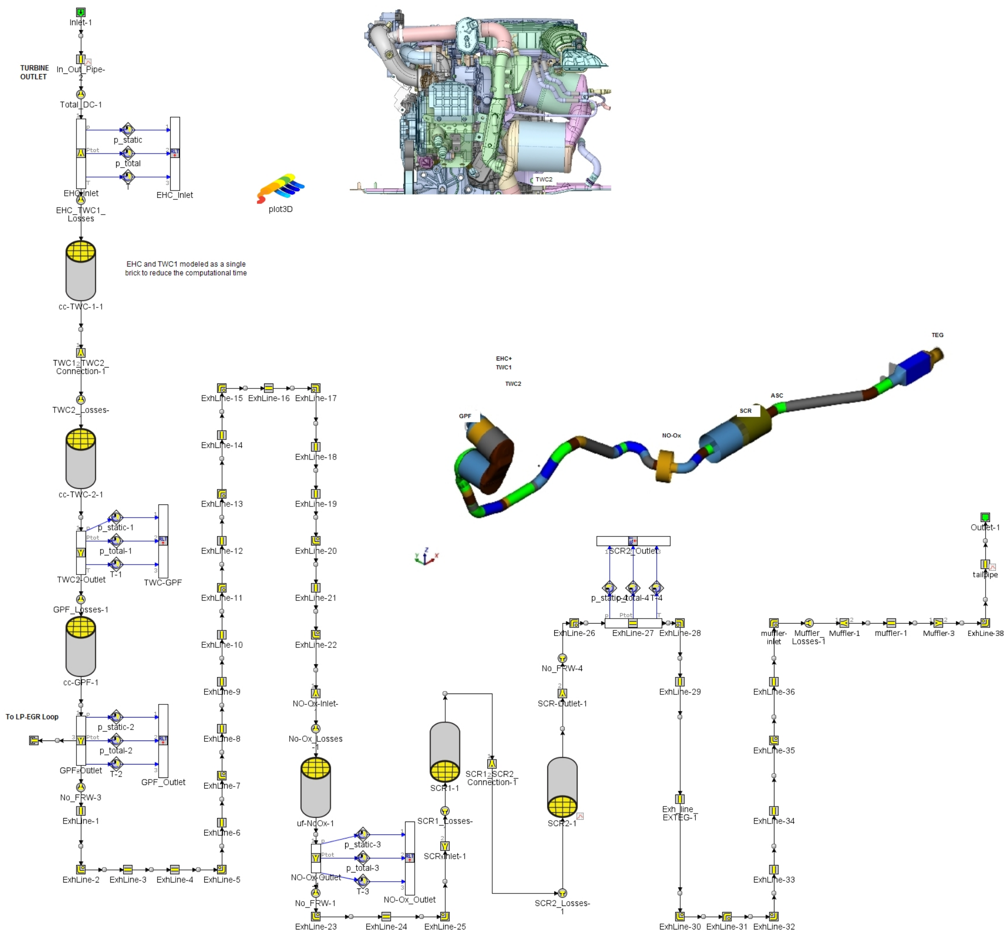

Figure 5.1: Complete EURO 7 Exhaust Model

The *fine tuning* means that for every exhaust brick, it can be calibrated a precise Discharge Coefficient and Friction Multiplier to exactly fit the experimental data acquired in a test-bed.
The actual exhaust is mounted on the engine and the muffler, that is not present in the PoliTO test cell setup, is modeled as a partially closed valve, in order to match the correct back-pressure at SCR 2 exit given by Marelli, that tested the complete exhaust line.

The fine tuning is performed using six calibration points and it is verified using the remaining forty seven point on an available group of fifty three acquired points.

Total Calibration points	**53**
Fine Tuning points	6
Test points	47

Table 5.1: Setup of the fine tuning

Tuning Boundaries	**Minimum**	**Maximum**
Mass Flow Rate [kg/h]	14	260
Pressure [bar abs]	1.00	1.73

Table 5.2: Calibration boundaries

So, the calibration of the complete exhaust model will fit exactly the back-pressures of the actual exhaust line, in each pressure sensor position. The sensors used in the tuning are placed at:

- Turbine Outlet;
- TWC Outlet;
- GPF Outlet;
- NO-Ox Outlet;
- SCR Outlet;
- Muffler Outlet;

To perform a correct tuning on the back-pressures, the actual temperatures in the exhaust line have to be matched. If this condition is not achieved, the density difference of the fluid will completely ruin the pressure calibration. So, the simulation time is set to 2000s because the starting temperature of the exhaust parts is set to 300°C to get a faster convergence to the equilibrium temperatures of the actual exhaust.
The convergence check for the temperatures at each case of the simulation is performed in the SCR 2 brick, that is the part where the warm-up is the most

difficult and slow. If they reach steady state in this brick, all the parts of the exhaust line will be at equilibrium temperature. The GT flux test model has not a Chemical Model implemented, so the temperature at the outlet of the bricks, that in actual conditions may have a higher value than at the inlet, cannot be perfectly matched. A satisfactory calibration will be reached if there is a maximum error of 50°C in the simulated temperature.

Once the temperatures inside the pipes in the model are verified, it is possible to analyze what happens to all the back-pressures in the five different sensors with a good precision.

The final objective of the pressure fine tuning is to match the pressure in the Turbine outlet, but it is not the only purpose. It is important to work on the exhaust system to correctly match not only the pressure at the turbine outlet, but either the pressure at TWC, GPF , NO-Ox and SCR sensors.
To tune in a proper way all the exhaust bricks, both close coupled and underfloor, the analysis has to start from the SCR Outlet and then it has to move up to the Turbine outlet.

The actual exhaust line has a muffler and a silencer after the SCR Outlet, so it has to be accounted for the pressure loss that it may produce.

Muffler fine tuning	**Optimized Value**
Discharge Coefficient	0.385

Table 5.3: Muffler Optimized Parameters

Once the Muffler is fine tuned, upstream of it, there is the SCR, such as the combination between SCR 1 and SCR 2: these are the bricks that need a fine tuning. This optimization is based on matching the acquired pressures at NO-Ox Outlet.

SCRs fine tuning		Optimized Value
SCR 1	Discharge Coefficient	0.804
	Friction Multiplier	1.195
SCR 2	Discharge Coefficient	0.893
	Friction Multiplier	1.611

Table 5.4: SCRs Optimized Parameters (fine tuned)

Upstream the NO-Ox there is the GPF sensor, placed at the GPF Outlet. The pressure acquired at this sensor can be used to be targeted by the Optimizer to fine tune the NO-Ox brick.

NO-Ox fine tuning	Optimized Value
Discharge Coefficient	0.898
Friction Multiplier	0.067

Table 5.5: NO-Ox Optimized Parameters (fine tuned)

The same procedure is followed in order to fine tune the GPF, using the acquired pressure at the TWC sensor at TWC Outlet.

GPF fine tuning	Optimized Value
Discharge Coefficient	0.240
Friction Multiplier	1.445

Table 5.6: GPF Optimized Parameters

At last, the pressure at Turbine Outlet has to be matched by calibrating the EHC and the TWC. The pressure information is acquired by the Turbine out sensor.

EHC and TWC fine tuning		**Optimized Value**
EHC	Discharge Coefficient	0.973
	Friction Multiplier	2.897
TWC	Discharge Coefficient	0.810
	Friction Multiplier	3.516

Table 5.7: EHC and TWC Optimized Parameters

The final fine tuned model matches all the pressure target for each pressure sensor in the test-bed layout.
The main result, however, remains the turbine outlet back-pressure, called P4, that is perfectly matched by the calibrated model.

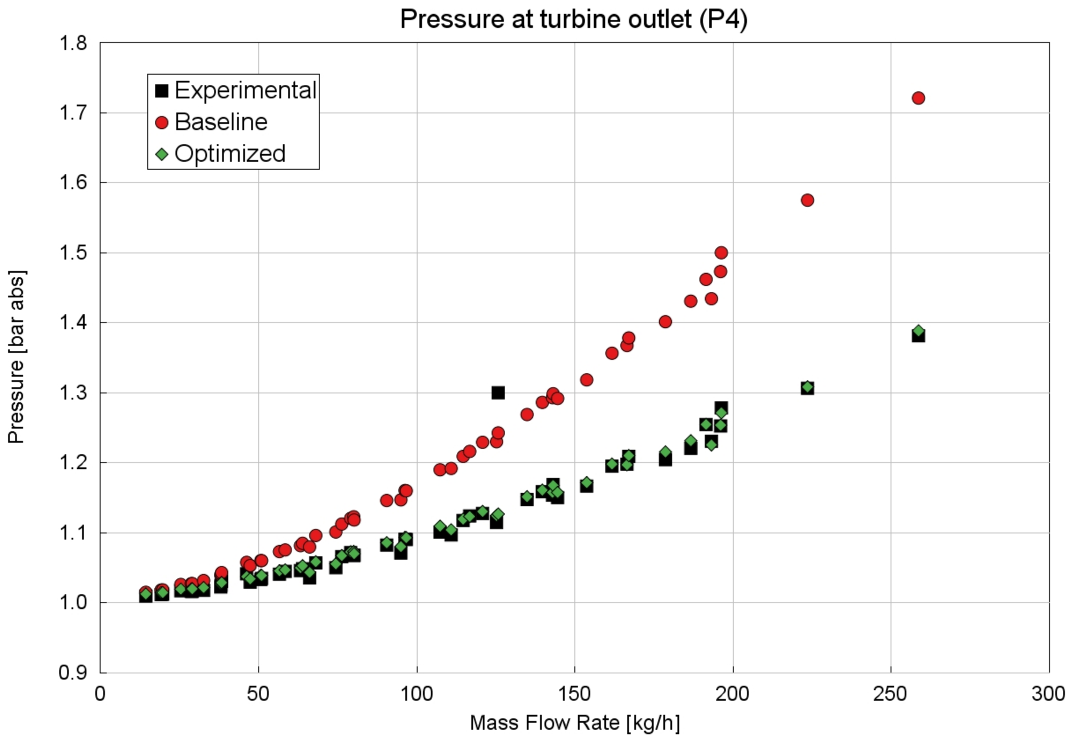

Figure 5.2: Results of the fine tuning procedure

5.3 Turbulence Calibration

The calibration of the Turbulence Model in GT is very important because it affects the behaviour of the combustion process.

Even if GT-Suite is a 1D CFD software, it is important to calibrate the turbulence model via some parameters that take into account the effect of turbulence on the mixture conditions at the start of the combustion phase, in particular for a predictive model. The 3D CFD model can simulate all the time and space behaviour of the flux during intake and compression phases, but the quantities involved in the current calibration are three global parameters used to describe the whole analyzed fluid domain. So, by working on the Turbulence Kinetic Energy, Length Scale and Tumble Number, it can be tuned a set of parameters that are able to reproduce the final state of the fluid at the spark firing time, crucial for the combustion development. For this reason, the objective of the tuning procedure is to correctly calibrate the turbulence model in the angular window that goes from -40° to 10° aTDCf, that is the expected spark advance window used in the PHOENICE engine.

In order to perform a correct calibration, a simple tuning model has to be obtained from the complete PHOENICE engine model, already existing. By isolating the cylinder, cranktrain and valves with their intake and exhaust pipes, the model can be used to set up the tuning procedure.

The GT model has to be geometrically built as a perfect twin of the CFD 3D model on which IFP worked in order to acquire data on the Kinetic Turbulent Energy (TKE), on the Length Scale and on the Tumble Number of the flux. In fact, the data used to calibrate the Turbulence model in GT come from this CFD.

The tuning procedure is performed in a motored engine condition (removing from the model the Combustion Object from GT) at 2000 rpm, with the valve profile actuation used in the CFD and with a gas and components temperature set to 293 K. The available PHOENICE model has to be update with the new intake ducts that are modeled on the basis of the actual ones, that enhance the tumble at the intake phase and with the corrected valve lift for both electric Intake Valve Opening (eIVO) and electric Intake Valve Closure (eIVC).

Valve control Strategy	**eIVO**	**eIVC**
Early Intake Valve Closure (EIVC)	360°	498°
Late Intake Valve Closure (LIVC)	360°	591°

Table 5.8: Valve strategy for the Turbulence model calibration

The calibration is based on three main parameters, such as the Production Term Multiplier (PTM), the Length Scale Multiplier (LSM) and the Intake Term Multiplier (ITM), using the Integrated Design Optimizer of GT. The algorithm chosen is the Genetic Algorithm with 20 generations in order to evaluate a number of combination big enough. The strategy used is a Pareto Multi-Objective based on three relative error functions, integrated from 540° to 720° for the TKE, from 480° to 720° for the Length Scale and from 520° to 720° for the Tumble Number, in order to acquire a single result per simulation cycle to minimize.

This choice is made because it is really difficult to calibrate the Turbulence model in order to fit the entire phases of Intake and Compression perfectly, excluding the Power and the Exhaust phases that are influenced by the combustion process, not implemented and not important for this kind of model tuning.
So, the results that can be considered satisfactory are the ones concerning the last part of the Compression phase, near the TDCf, where it is very important to predict the mixture conditions before the spark firing.

Angular window of interest	**Start**	**End**
Engine Crank Angle (0° at TDCf)	-40°	10°

Table 5.9: Angular window for the Turbulence optimization

The data used in the calibration are referred to Early and Late conditions with a sweep evaluation of this cases in the Optimizer.

There are four additional parameters, such as the Tumble Term Multiplier, the Tumble Destruction Coefficient 1, the Outflow Tumble Decay Multiplier and the Reverse Tumble Decay Multiplier, that can be used to refine the main calibration, in order to get a perfect data fitting of the simulated results on the reference CFD ones.
In order to better understand the Turbulence Model plots, the legend meaning has to be clarified:

- Experimental, such as the data obtained from the CFD 3D based on the actual engine layout;
- Default, such as the turbulence model implemented in GT-Suite, all the calibration parameters set to ’def’;
- Baseline, such as the turbulence model implemented in the original GSE T4 simulation;
- Optimized, such as the result of the calibration based on the CFD 3D data.

The tuning results for the Turbulence model are shown below.

Main Turbulence Parameters	**Optimized Value**
Production Term Multiplier	0.7
Length Scale Multiplier	0.9
Intake Term Multiplier	2.0

Table 5.10: Main Turbulence model tuning parameters

Refinement Turbulence Parameters	**Optimized Value**
Tumble Term Multiplier	1.5
Tumble Destruction Coefficient 1	0.3
Outflow Tumble Decay Multiplier	0.5
Reverse Tumble Decay Multiplier	0.1

Table 5.11: Refinement Turbulence model tuning parameters

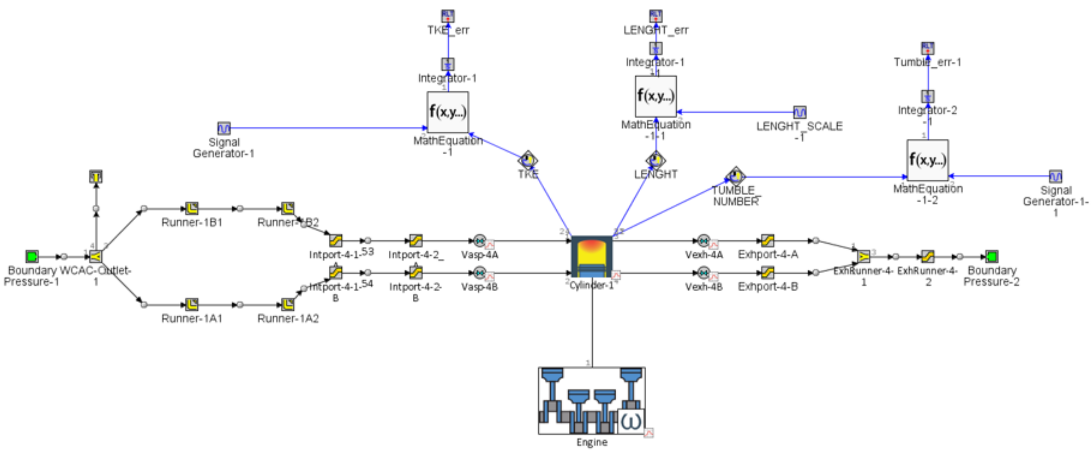

Figure 5.3: GT Turbulence Model

Early Intake Valve Closing Results

In the following figures, the results obtained are shown for Turbulent Kinetic Energy (TKE), Length Scale and Tumble Number for the EIVC valve actuation in the conditions described before.
The simulated TKE has a very good trend. In fact, there is a sort of offset that not lead to a big error in the combustion simulation, that requires the correct TKE and LS at the *combustion start*.

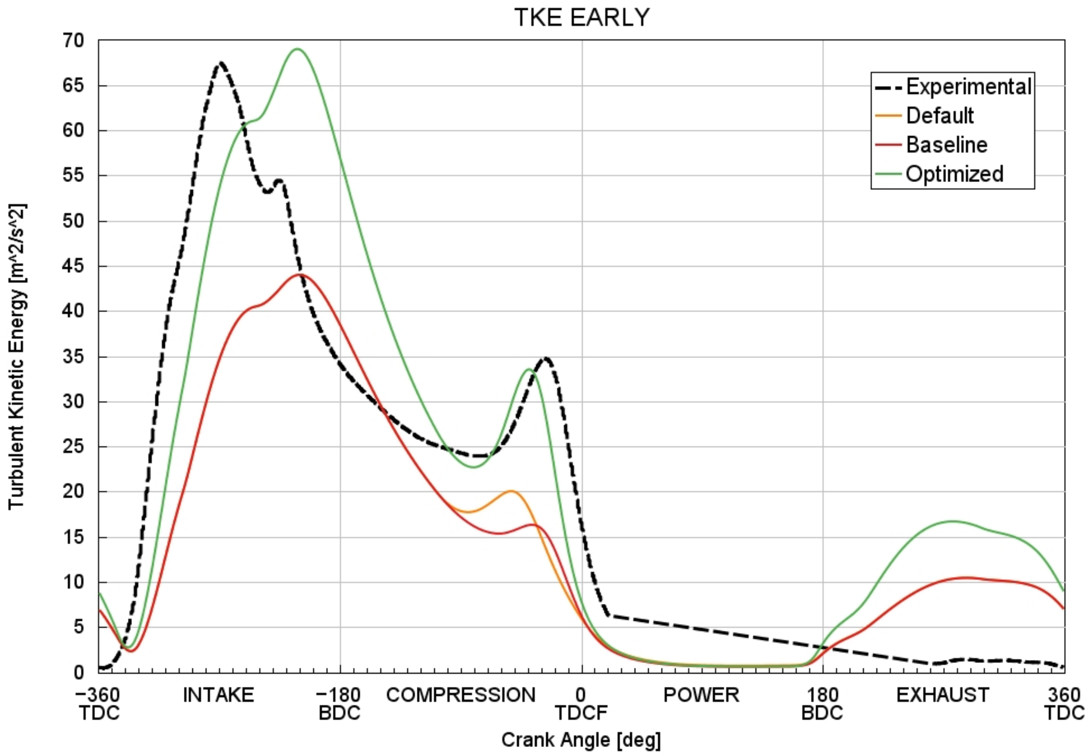

Figure 5.4: Turbulent Kinetic Energy optimization for EIVC

The Length Scale is perfectly simulated, with this precision the flux conditions at the spark will be the same of the actual engine, allowing to perform a solid SITurb Calibration, if needed.

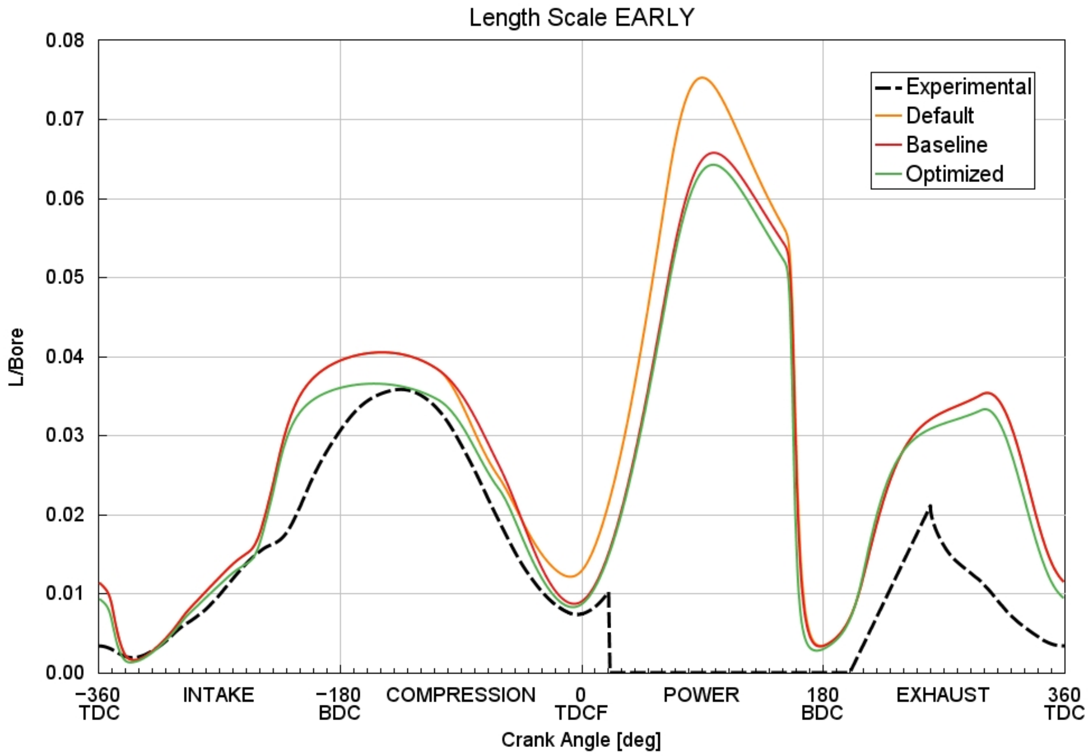

Figure 5.5: Length Scale optimization for EIVC

The Tumble Number behaviour near the TDCf is really good, the mixture conditions are correctly simulated in the most important phase for the combustion.

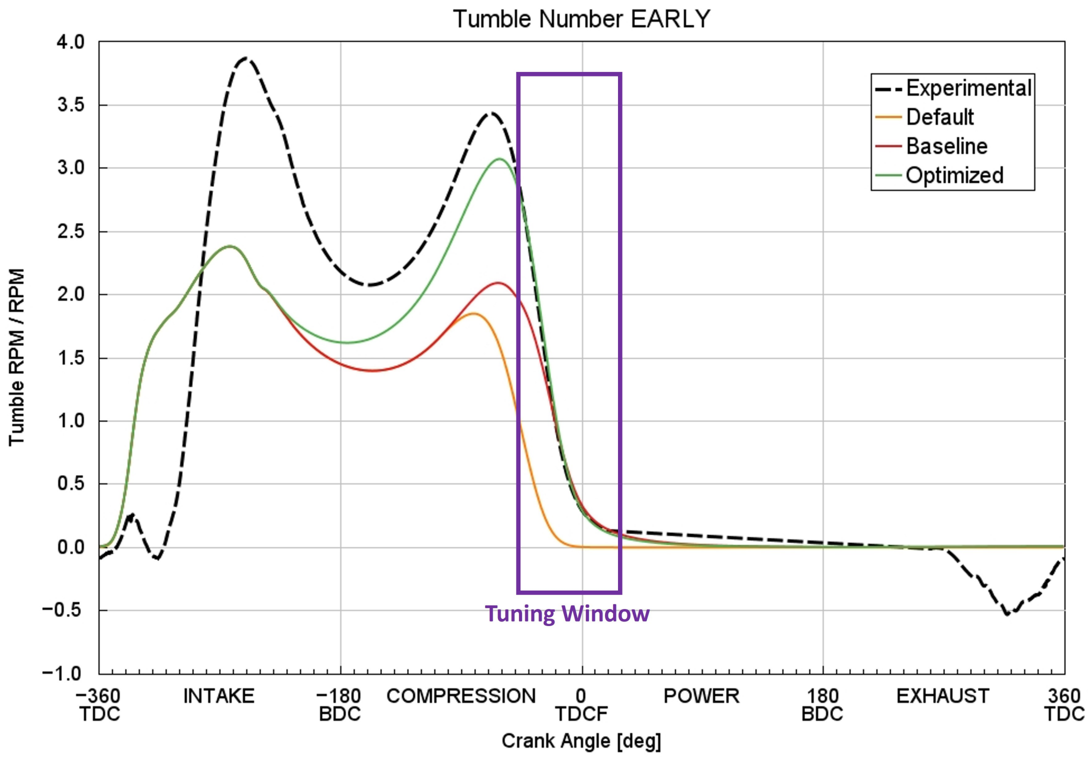

Figure 5.6: Tumble Number optimization for EIVC

Late Intake Valve Closing Results

In Late conditions, the results are very similar to the EIVC one, showing the consistency of the Turbulence Calibration in both the valve strategy implemented in the PHOENICE engine.
The TKE optimized is really close to the experimental one, the trend is correctly simulated and the relative error between the two values is acceptable.

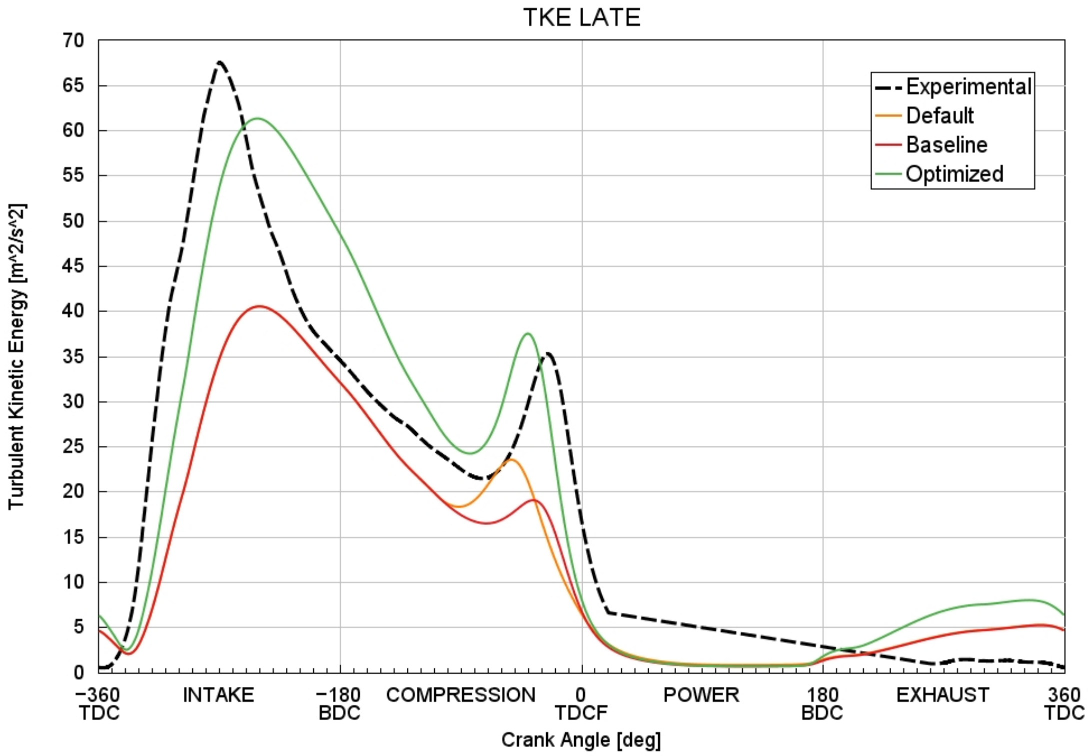

Figure 5.7: Turbulent Kinetic Energy optimization for LIVC

The Length Scale has the same behaviour of the EIVC one, fitting very precisely the experimental LS of the 3D CFD, used to calibrate the model.

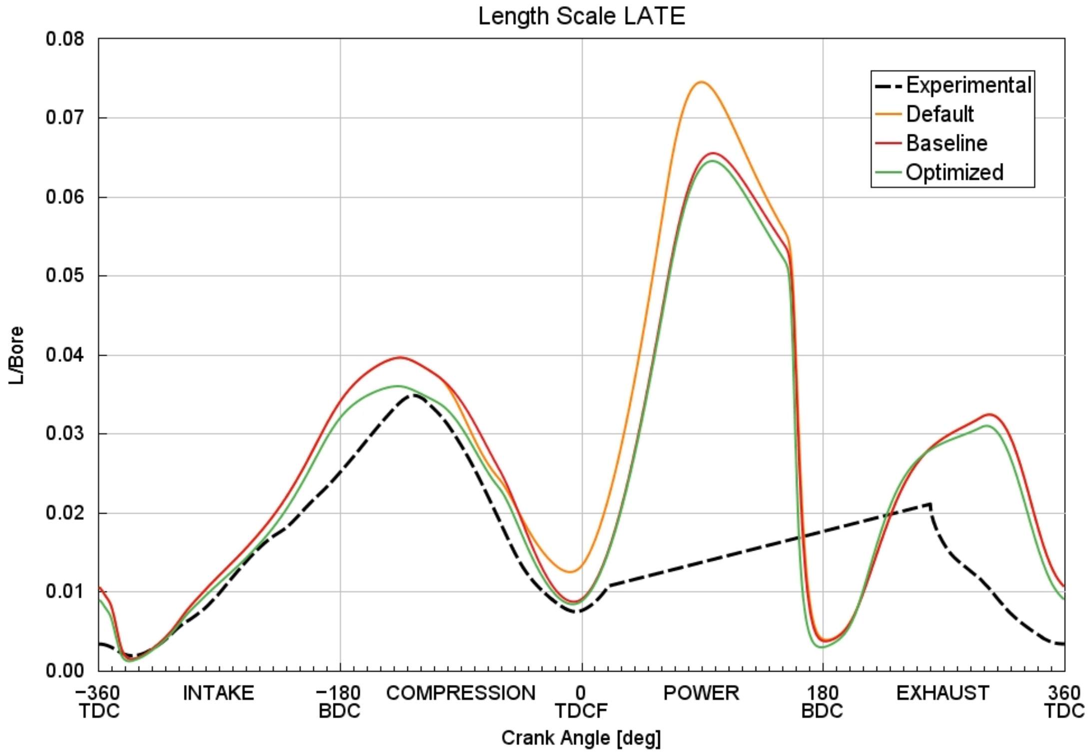

Figure 5.8: Length Scale optimization for LIVC

The Tumble Number is correctly simulated in the angular window of interest, such as the one where the spark timing advance has to be selected.

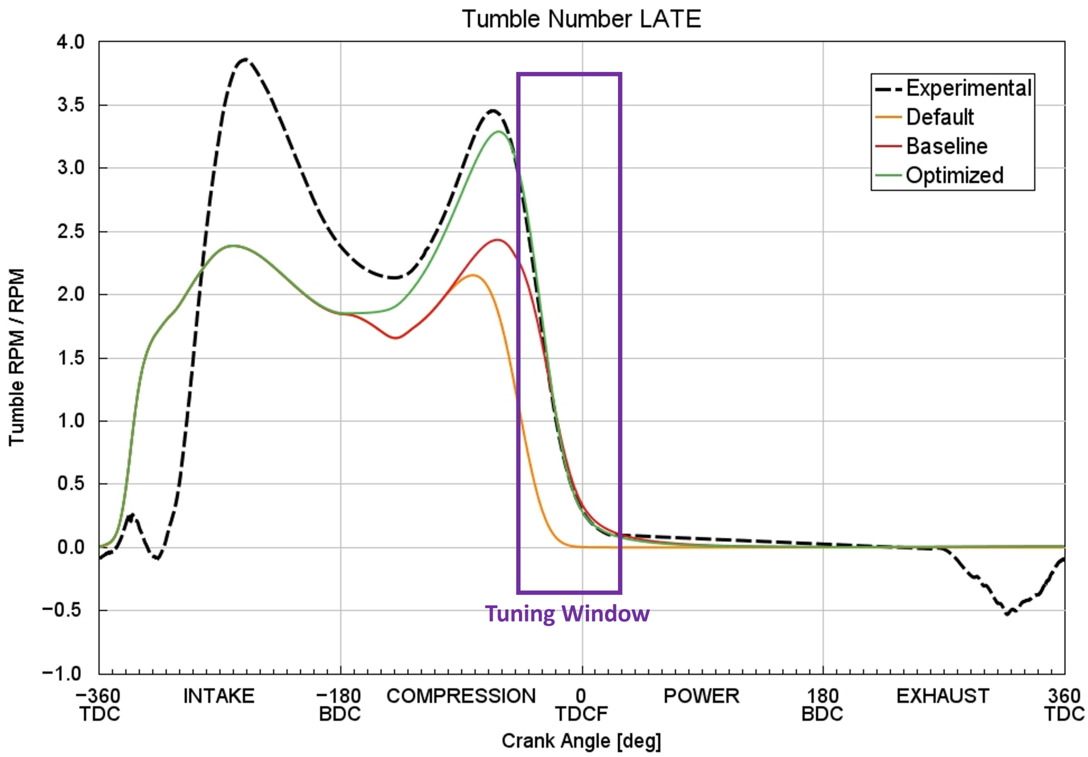

Figure 5.9: Tumble Number optimization for LIVC

5.4 Assessment on the impact on Indicated Thermal Efficiency (ITE) of PHOENICE engine features

Once the calibration of the exhaust and of the turbulence model is correctly performed, an interesting test on the PHOENICE engine performances is to start from the GSE T4, taken into account as a common state-of-the-art turbocharged gasoline engine, and to add the PHOENICE implemented features one at time to it, in order to evaluate the importance of each solution.
In order to analyze different working conditions, a set of 7 key-points is selected. This KP selection is based on their role in the validation procedure, granting results very close to the actual ones. They are the shown in the Table 5.12.

Key-Points selected for the analysis	
Rotational speed [rpm]	**BMEP [bar]**
1000	6
1500	8
2000	10
2500	10
2500	12
3000	10
3500	10

Table 5.12: Analyzed key-points

The evaluations on the performance of the PHOENICE are based on the ITE gains or losses; starting from the baseline model, the final PHOENICE setup is obtained by implementing one solution at a time.

The features added are:

- Compression Ratio (CR) from 10.5 to 13.6 and optimized valve strategy (VVA) for the PHOENICE Miller Cycles;
- E-Turbocharger VNT layout from a simple WG one and EURO 7 exhaust line from a standard EURO 6 system;
- EGR dilution of the charge;
- Lean conditions in order to achieve the Dual Dilution;

5.4.1 Model Validation

In order to perform the analysis on the ITE enhancements, obtained by using the PHOENICE solutions, it is necessary to validate the *Complete PHOENICE Model*. In fact, a representative analysis can be done only if the GT Model is able to correctly compute the main engine quantities, with a small relative error, in order to converge to the experimental data obtained in different working conditions, tested in the test-bed.

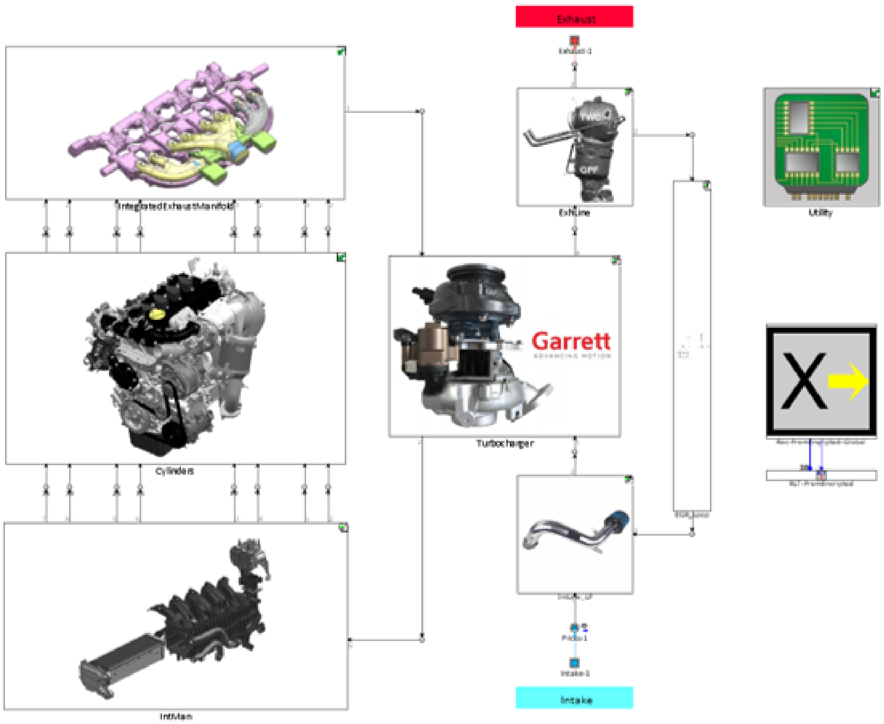

Figure 5.10: GT PHOENICE Model

By setting the experimental boundary data in the model and by setting the target IMEP to the throttle control and the target P2 to the turbine rack control, the Complete Model shows coherent results with the experimental expectations.

Imposed conditions on the validation model	
Reference Object	**Parameter**
Valve actuation	eIVO
	eIVC
Combustion Phasing	Spark Timing
Friction at Crankshaft	FMEP
Ambient	Pressure
	Temperature
	ϕ (Humidity)
Injection Line	Injection Pressure
	Injection Phasing
Oil	Temperature
Air at the WCAC Outlet	Temperature (T5)

Table 5.13: Imposed boundary conditions for the model validation

The parameters used to verify the validation model are the one shown in Table 5.14; some of them are affected by the combustion and the turbulence behaviour, thus it is possible to assert the correct model calibration. In order to correctly link the BMEP to the IMEP, the FMEP has to be an imposed experimental parameter because the Chen-Flynn FMEP calculation may produce different results, that can introduce an error on the BMEP calculation during the comparison between Simulated and Experimental. The validation is based on the performance assessment on the selected key-points shown in the PHOENICE engine map in Figure 5.11.

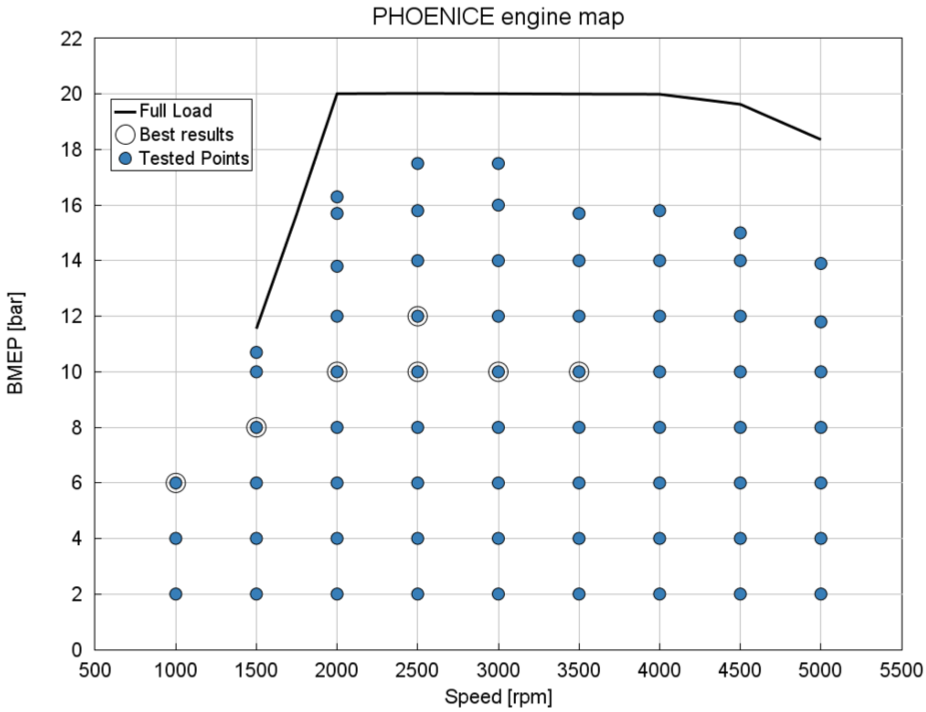

Figure 5.11: Validation Key-Points in the PHOENICE map

The model coherence can be analyzed by looking at the five main sensors quantities identified as, for the pressure, P1 for the pressure before compressor, P2 for the pressure after compressor, P3 for the pressure before turbine (engine-out), P4 for the pressure after turbine, P5 for the pressure after the throttle, at the intake ducts (WCAC outlet). The same check can be done for the temperatures. The validation model is targeting the P2 with the rack position control, so the simulated P2 is expected to be the experimental one; the turbo rotational speed and the air mass flow rate at a target P2 can be important check parameters for the model coherence. The throttle control is targeting the IMEP, so if the model is correctly set, the simulated P5 pressure has to converge on the experimental one. The temperature T5 is expected to fit exactly the experimental one because it is imposed at the WCAC outlet. The calibrated exhaust line can be checked again by looking at simulated T4 and P4 values, that fit correctly the experimental sensors acquisitions. The turbulence and combustion calibration can be verified by looking at P3 and T3 quantities, because if the simulated results converge on the experimental one, the model is performing solid simulations. In order to better evaluate the combustion phase, the Peak Pressure within the cylinder and the respective Crank Angle can be plotted and analyzed respects the experimental measurements, taking into account that only the spark advance is imposed in the simulation. A good match between simulated and experimental can confirm the correct behaviour of the GT model.

Check parameters for the validation model	
Reference Object	**Parameter**
Efficiency	ITE
In-Cylinder conditions	Maximum Pressure
	CA @ Maximum Pressure
Air conditions after compressor	Pressure (P2)
	Temperature (T2)
Exhaust gas conditions before turbine	Pressure (P3)
	Temperature (T3)
Air conditions at WCAC outlet (after throttle)	Pressure (P5)
	Temperature (T5)

Table 5.14: Validation model check parameters

Correlation Plots

The validation is correctly performed if the Correlation Plots, obtained using the Validation Parameters, show values inside the 5% error band respects the experimental value line.

All the parameters of the Table 5.14 are plotted as Correlation Plots, with an optimal result represented by the central line, where the equation *Experimental = Simulated* is verified, such as the Model gives exactly the same experimental values. The two other lines are distant from the optimal one by a 5% error, that is taken as the maximum error achievable in order to verify the model. The seven points chosen for the validation have a set of four high load BMEP KP, that can verify with accuracy the model, and a set of three medium-low BMEP KP at low rotational speed, that show a lower accuracy, with errors slightly over the 5% band.

This is a choice made to include some medium-low BMEP points at low speeds, very important in the WLTC test cycle, in order to study their ITE changes in the next section.

The first Correlation Plot is based on the ITE, that is the core of the entire assessment. It can be seen that only one point is out of the error band. It is related to the 1000 x 6 key-point, that is a difficult point to be evaluated. The reason why it is a delicate point is the low rotational speed of the turbocharger and the retarded spark advance, implemented in order to avoid combustion anomalies, due to model predicted knock issues that come from the low turbulent flame speed at 1000 rpm.

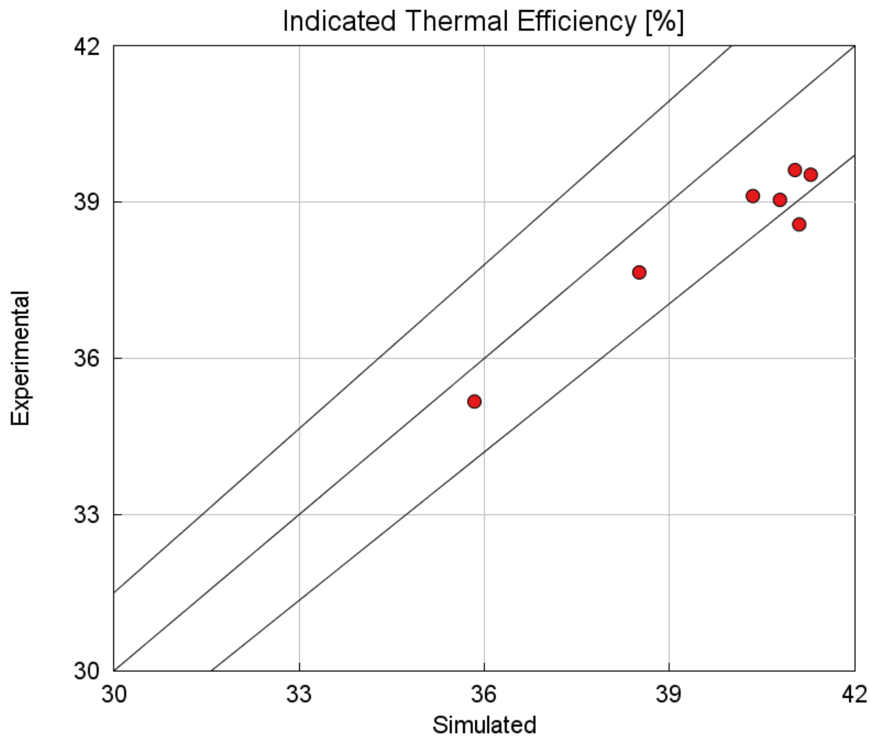

Figure 5.12: Correlation Plot: ITE

The maximum pressure value within the cylinder and its respective crank angle (CA) are important parameters to be evaluated. In fact, these quantities can easily represent the combustion process development. The maximum pressure represents the capability of the ignited charge to develop a pressure increase that can be converted in power. The CA at maximum pressure is related to the combustion phasing and can give information on the combustion speed.
It can be seen that in this case too, three KP are slightly out of the error boundaries. They are the medium-low BMEP points with low rotational speed, such as the 1000x6, 1500x8 and 2000x10.
These three points have a pressure cycle difficult to be simulated with the SITurb calibration available for this work; the spark advance near the TDCf makes the pressure trend complicate to be followed by the simulation logic, generating fairly small errors, but higher than 5%, during the validation process.

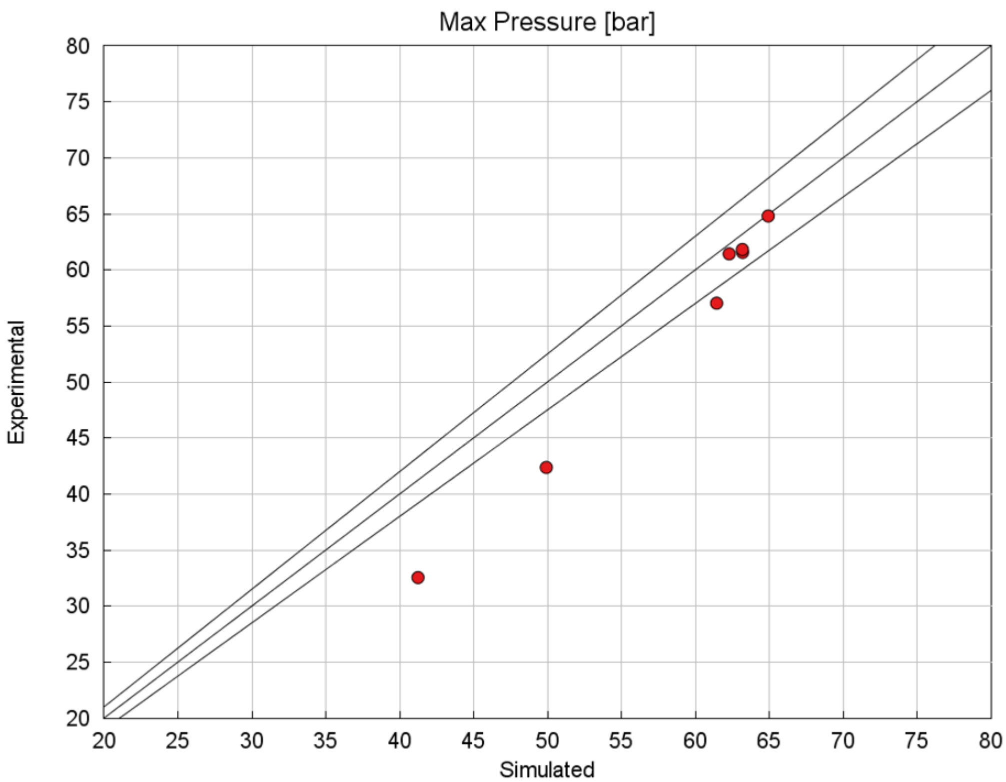

Figure 5.13: Correlation Plot: Maximum Pressure

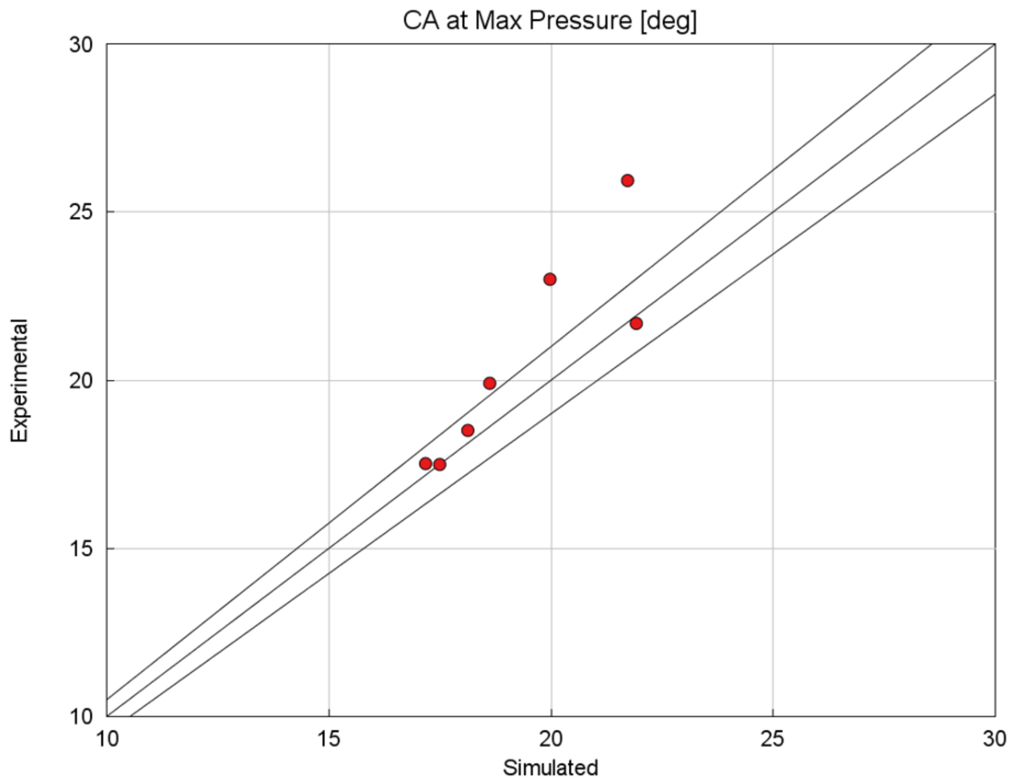

Figure 5.14: Correlation Plot: Crank Angle at Maximum Pressure

The next Correlation Plots are related to three engine positions where the pressure and temperature values are extremely important. They have to be the closest possible to the experimental ones, in order to guarantee the correct mass flow rate and the right boundary conditions of the cylinder object to simulate coherently the combustion.

The first main position is the *Compressor Outlet*, where the pressure P2 and the temperature T2 can be seen. The validation model is based on the VNT rack control calibrated to match the experimental P2, in order to reproduce the same boost conditions, with possibly the same experimental temperature T2, that is calculated by the model. In fact, the boost pressure is always matched and the error in only one of the temperatures is only of some degrees, a tolerable value, even if it is over the 5% validation threshold.

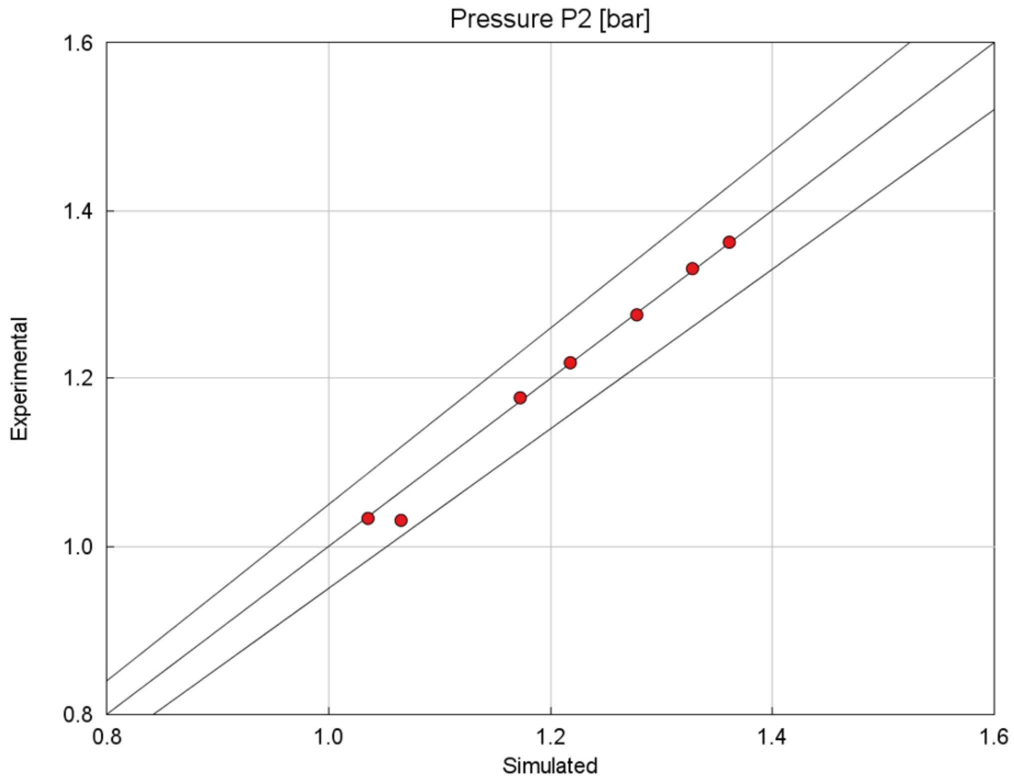

Figure 5.15: Correlation Plot: Pressure P2 at the compressor outlet

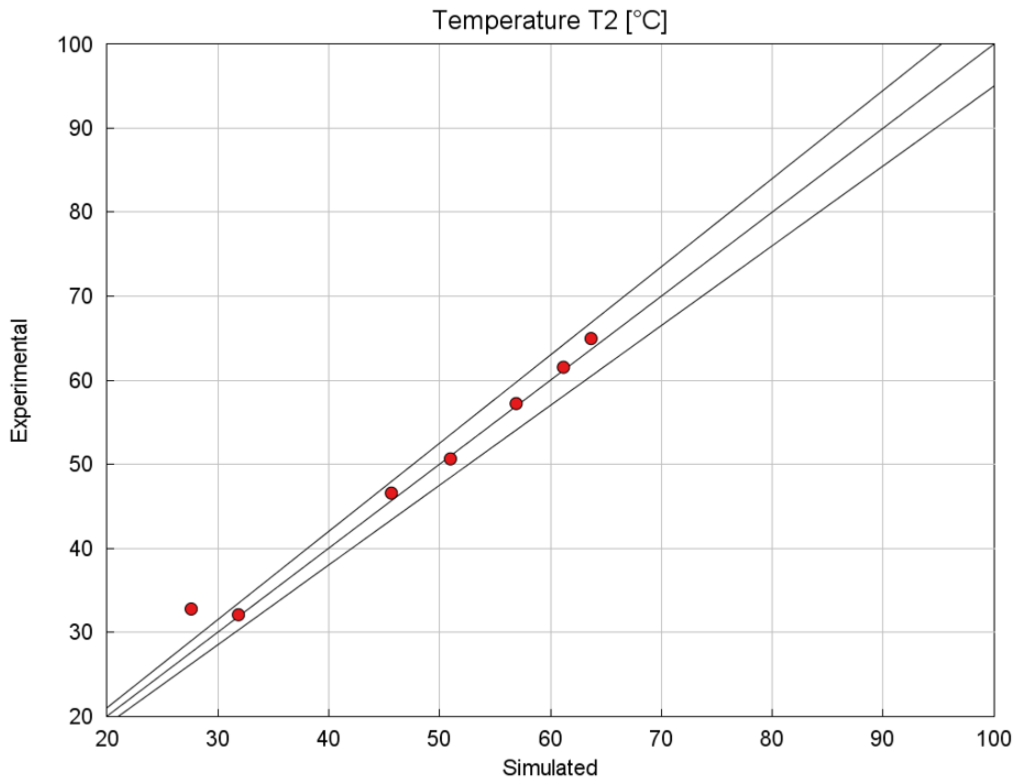

Figure 5.16: Correlation Plot: Temperature T2 at the compressor outlet

The second main position is the *WCAC Outlet*, such as the outlet of the device that cools down the compressed air. It is situated after the throttle, so the pressure P5 can be seen as *the pressure after throttle*. It is very important because this value is the consequence of the throttle adjustments made in order to match the imposed experimental IMEP by the throttle controller. So, if the simulated P5 matches the experimental one, the combination between IMEP and boost pressure (P2) of the model represents consistently the whole intake actual conditions of the engine.
The temperature T5 is imposed by the validation model constraints, in order to simulate the cooling effect of the WCAC on the charge, so it matches perfectly the experimental one.

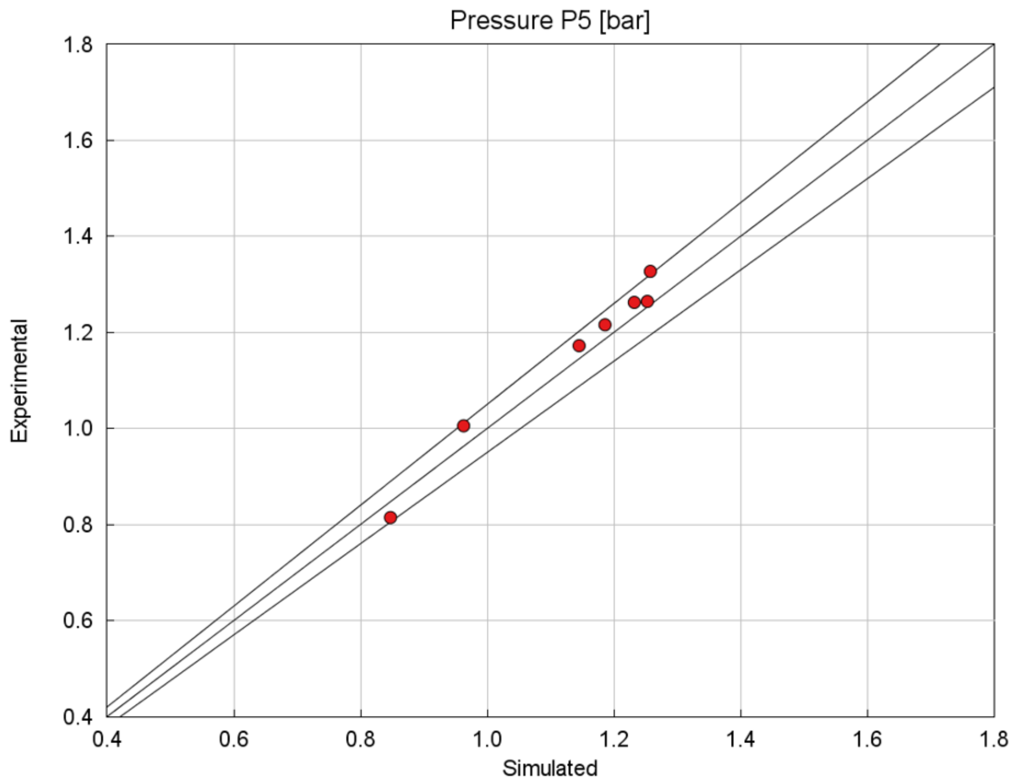

Figure 5.17: Correlation Plot: Pressure P5 at the WCAC outlet (after throttle)

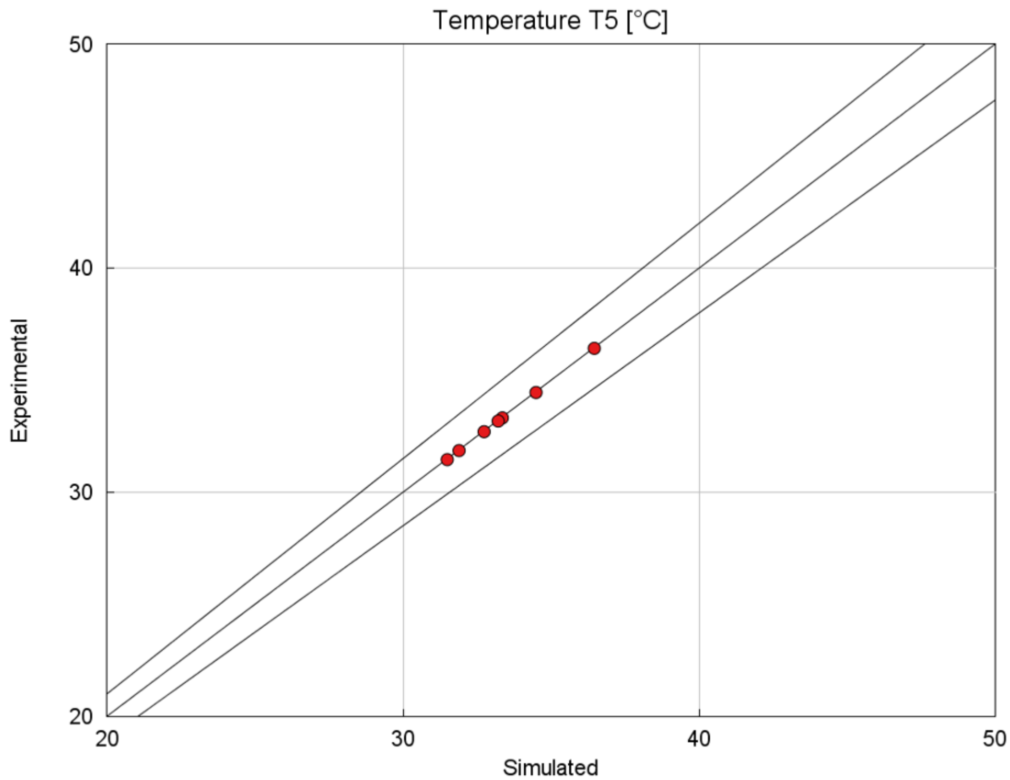

Figure 5.18: Correlation Plot: Temperature T5 at the WCAC outlet (after throttle)

The last main position is the *Engine-out*, which can also be seen as the *Turbine Inlet*, where the pressure P3 and the temperature T3 are referred. These two quantities are strongly influenced by the combustion process and by the back-pressure given by the exhaust line, so a correct P3 can also states that the pressure after the turbine (P4) is correctly simulated too. It can be seen that, even if the 1000x6, 1500x8 and 2000x10 KP have slightly different pressure cycles with respect to the experimental ones, the P3 is always correctly simulated. The same can be said for the simulated temperature T3, so the combustion process is modeled in a satisfactory way.

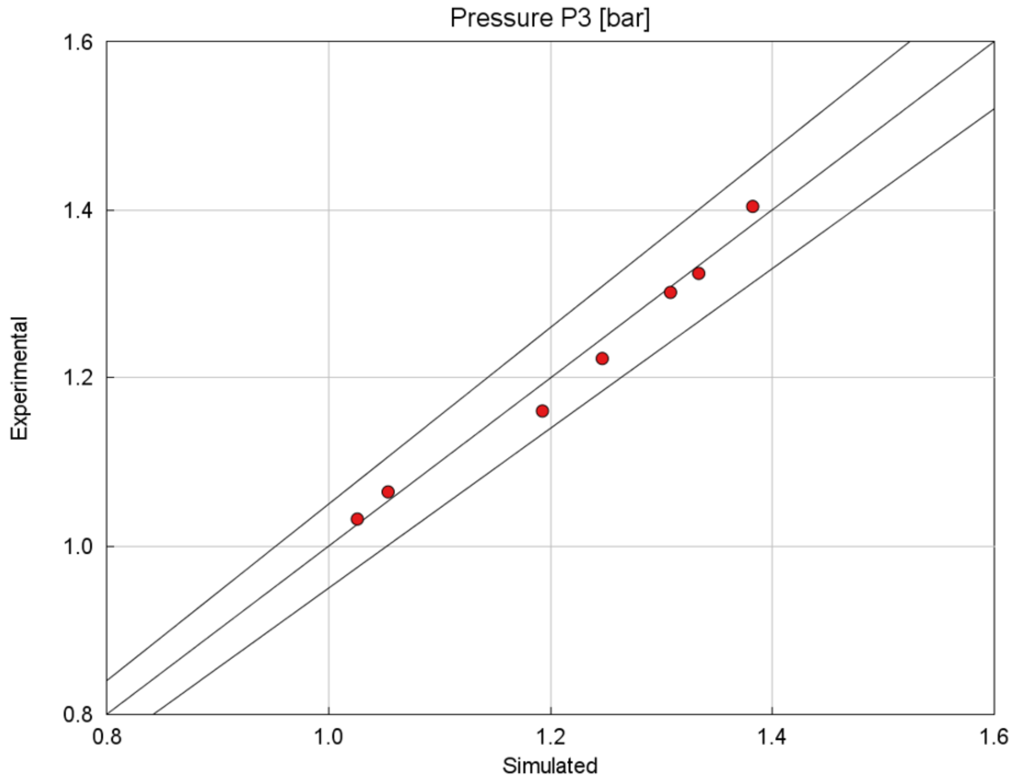

Figure 5.19: Correlation Plot: Pressure P3 at the turbine inlet (engine out)

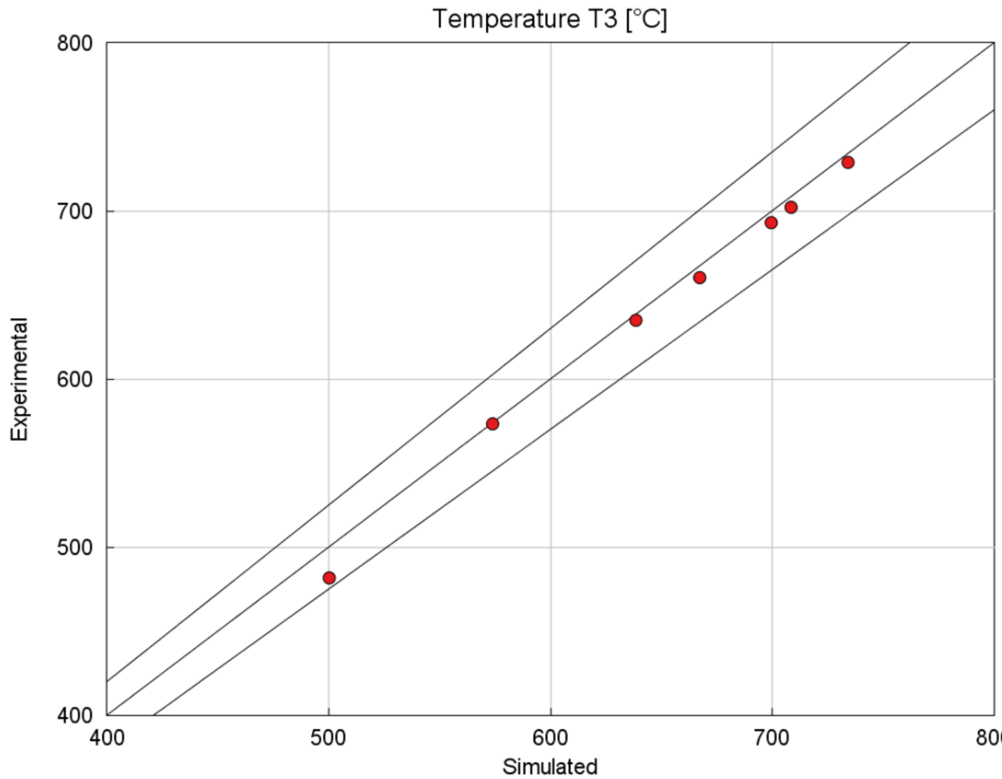

Figure 5.20: Correlation Plot: Temperature T3 at the turbine inlet (engine out)

Best Pressure Cycles

As mentioned before, the best four points have a simulated Pressure Cycle very close to the actual one, allowing to use them as reference points in the validation procedure. The best results of the simulation are related to this optimal key-points. They are the 2500x10, the 2500x12, the 3000x10 and the 3500x10.

The 2500 x 10 KP shows a simulated Pressure Cycle compared to the experimental one as it follows:

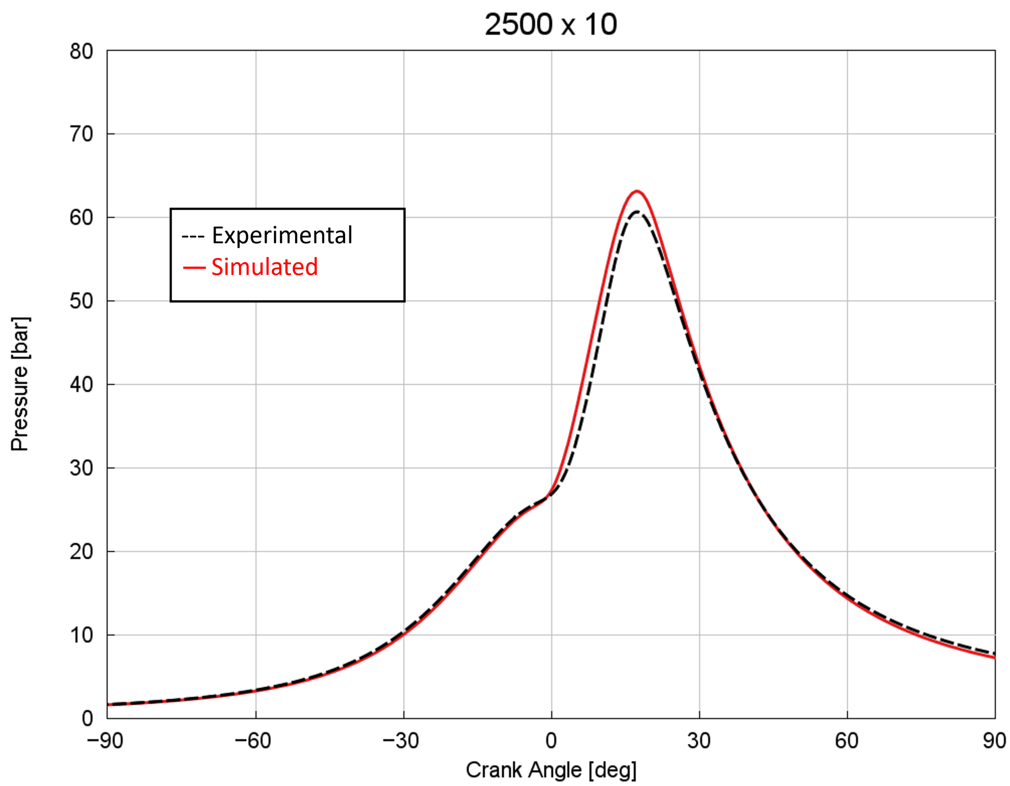

Figure 5.21: Simulated Pressure Cycle for the 2500 x 10 KP

The 2500 x 12 also is a very good simulated KP, with a lower error after the TDCf than the previous one:

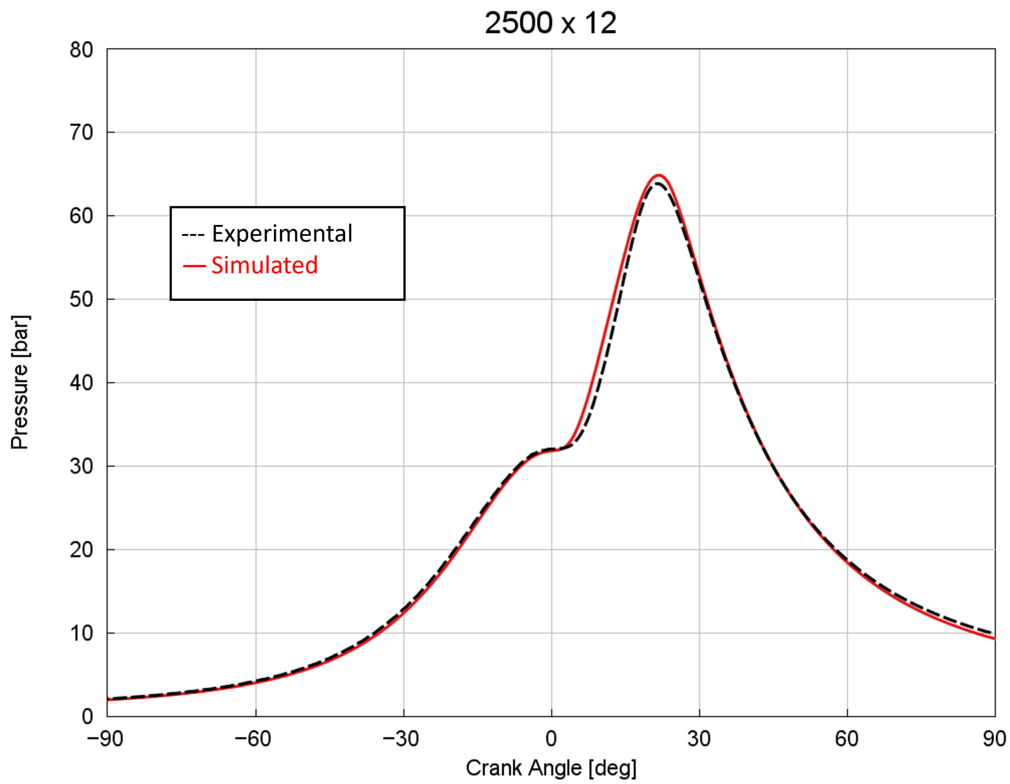

Figure 5.22: Simulated Pressure Cycle for the 2500 x 12 KP

The 3000 x 10, even with an higher rotational speed than before, is still very good in terms of simulated pressure cycle:

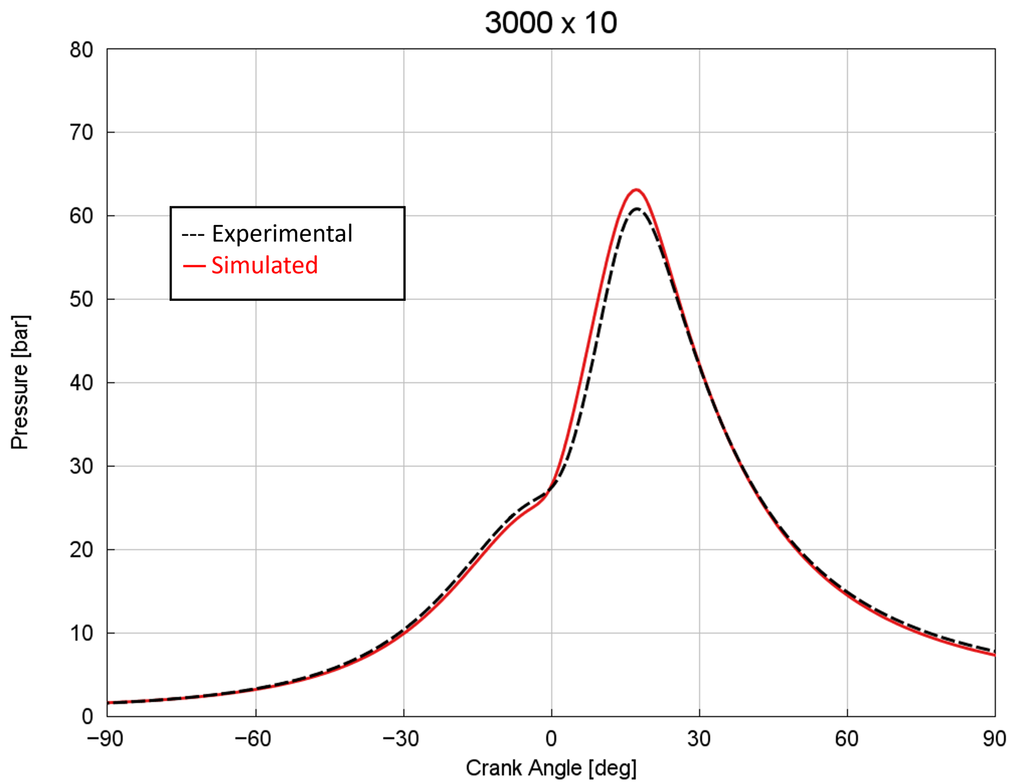

Figure 5.23: Simulated Pressure Cycle for the 3000 x 10 KP

Also the 3500 x 10 shows great coherence between simulated and experimental:

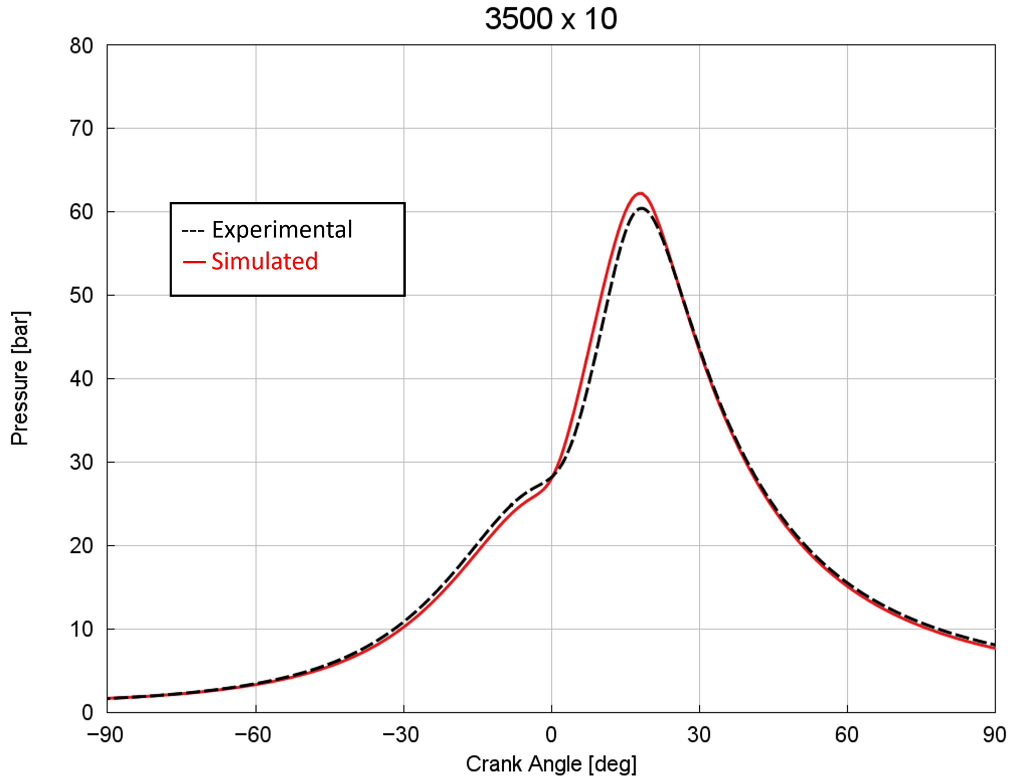

Figure 5.24: Simulated Pressure Cycle for the 3500 x 10 KP

5.4.2 Evaluation of the benefits of the PHOENICE technologies on the engine efficiency

The assessment, as said before, is based on the Indicated Thermal Efficiency (ITE). This choice was made in order to analyze the performance linked only to the engine capabilities, without considering the efficiency of each accessory implemented. So, the ITE comparison is *friction-less*, while the BTE includes in its evaluation all the efficiencies of auxiliaries (that varies from a vehicle to another or, as well, for the same part, by implementing different technical solutions).

Baseline

This configuration is the production one of the GSE T4 engine. The solutions used in this layout are the WG turbo control and the 10.5 CR using stoichiometric combustion, with the VVA standard control strategy for IVO and IVC actuations. The exhaust line has a EURO 6 configuration, with only a TWC and a GPF, granting a lower back-pressure with respect to the EURO 7 layout of the PHOENICE.
It is important to state that the Baseline ITE is simulated as well, with the same GT PHOENICE model used to perform the next simulations too. It is modified to match the GSE T4 technical configuration, by changing sub-assemblies and Case Setup data. So, the values shown below are not took from an experimental campaign.

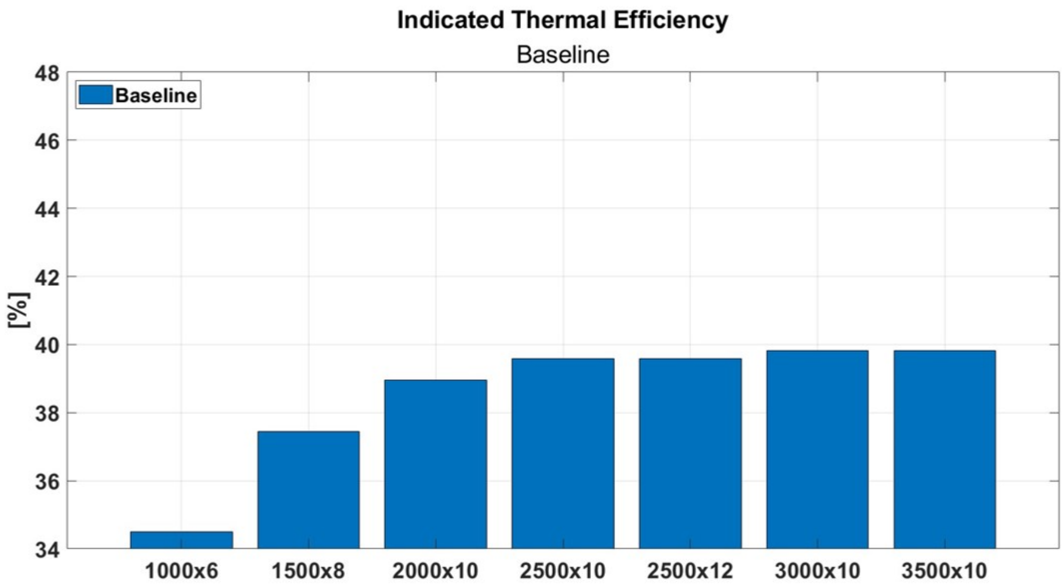

Figure 5.25: Baseline ITE

Baseline with CR 13.6, VVA optimized

This model is the starting one, called Baseline, with the CR increased from 10.5 to 13.6. Also, the optimized VVA allows to perform Miller cycles, both Early and Late, in a much efficient way, providing a more precise and large control window for valve actuations. The effect on the ITE is expected to be very significant, but the higher CR may induce knock complications for high loads conditions. The efficiency gain at high speeds is higher; it may be connected to a stronger dethrottling strategy with the VVA, allowed by a more stable combustion (due to the higher rotational speed). The CR effect on the ITE can be clearly seen for each operating point. In particular it is recognizable at low speeds, where the VVA effect is less aggressive on the performance gains.

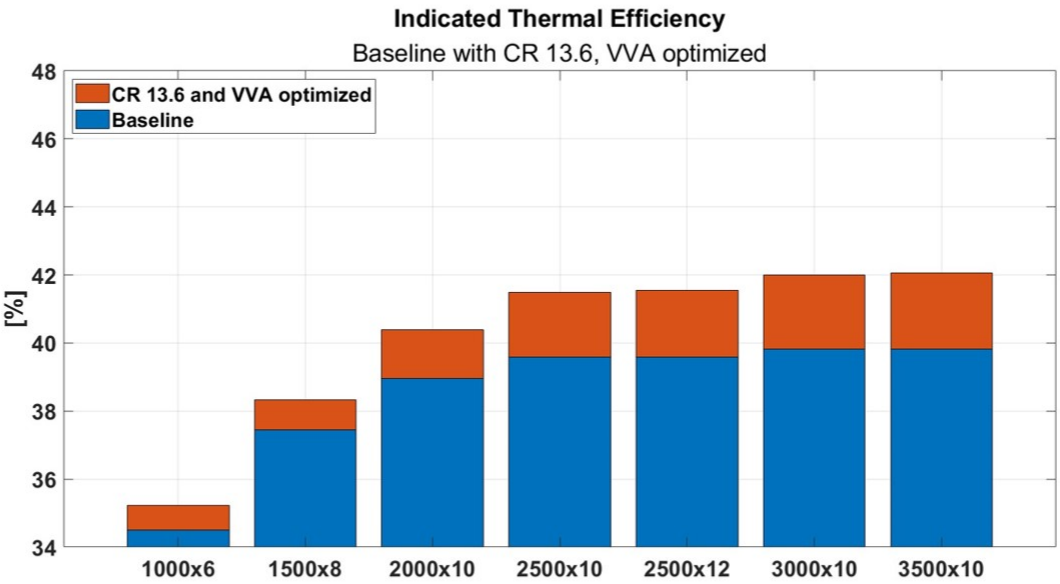

Figure 5.26: CR 13.6 and optimized VVA effects on ITE

PHOENICE ($\lambda = 1$)

In order to further increase the performance of the engine, the WG turbocharger has to be replaced with the e-Turbo, that is VNT controlled. This means that the boost at the intake manifold is controlled by the rack position and the variable geometry allows the turbine to work on many different maps with the best efficiency achievable in every condition. The effect of the electric motor cannot be witnessed in this evaluation, because the points analyzed are steady state KP, but the increased efficiency granted by the rack control of the VNT can be clearly seen on the ITE gain. In particular, the effect is much higher for the medium-high load points

where the turbocharger is providing effective boost. The exhaust line is the one implemented in the final PHOENICE layout. The EURO 7 exhaust, necessary to reduce the tail-pipe emissions, provides a much higher back-pressure. The reduced emissions are a great results, but the presence of the NO-Ox, the SCRs and the EHC on top of the TWC increase the PMEP, leading to an ITE loss. This effect is clear for high load points, where the increased pumping losses become significant due to the higher exhaust gases flow rate (the losses behaviour respond to the equation $\Delta p_{loss} = kQ^2$, where Q is the flow rate). It can be seen that the ΔITE, gained from the adoption of the VNT turbo, is lost due to the losses introduced by the new exhaust.
So, the ITE observed is nearly the same of the Baseline configuration with CR 13.6 and optimized VVA.

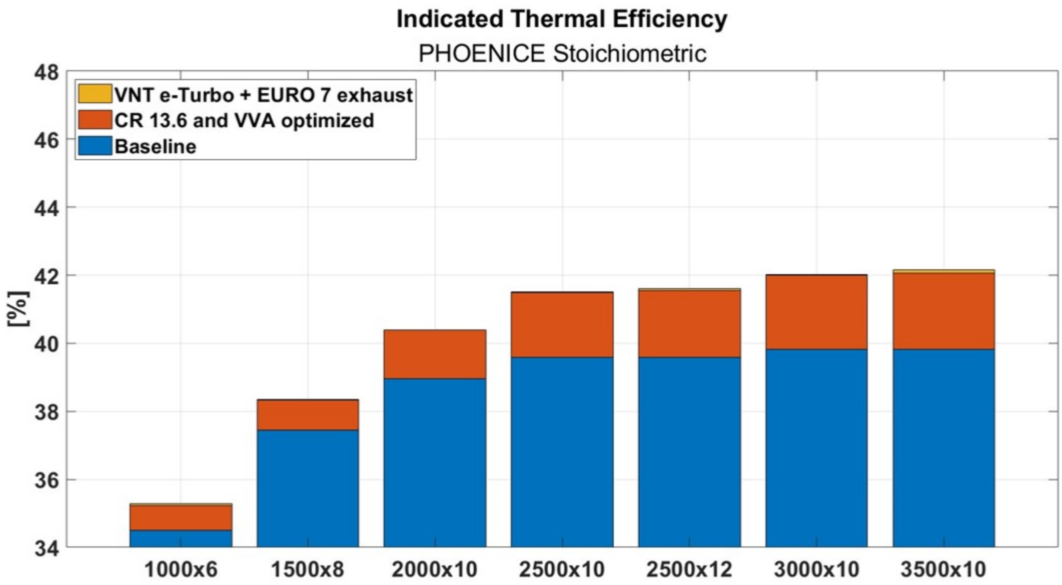

Figure 5.27: E-Turbo and EURO 7 exhaust effects on ITE

PHOENICE ($\lambda = 1$ + EGR)

The PHOENICE engine can run in stoichiometric, EGR and Dual Dilution configurations. The desired EGR rate, obtained from the ECU maps, is imposed in the model in order to evaluate its effect on the efficiency. The results are strongly influenced by the EGR dilution, with a great gain in ITE.
The combination between the PHOENICE features allows to work at 10 bar BMEP with a solid 15% EGR rate, without major stability issues. The ΔITE gained for each KP is nearly the same, with the exception for the 1500 x 8, where the lower

BMEP required allows to perform a strong dethrottling via EGR dilution. The EGR rate, however, cannot go up to 15%, but it has to stay at 9.4% in order to avoid an excessive turbulent flame speed reduction. The same thing happens to the 1000x6, but the lower rotational speed and so the lower combustion stability limits the maximum EGR rate, with a less significant ITE gain coming from the EGR dethrottling.

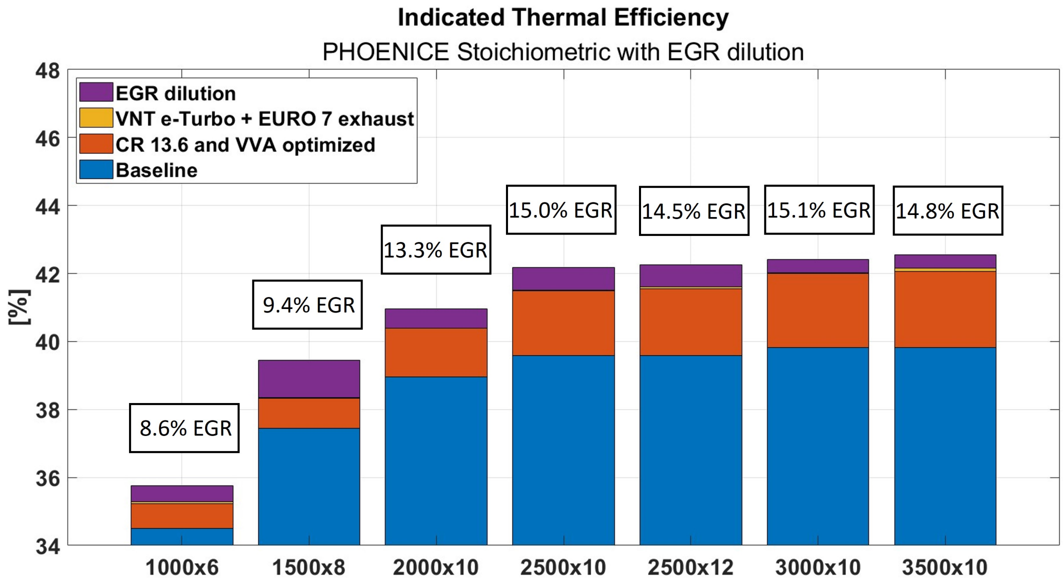

Figure 5.28: EGR effects on ITE

Complete PHOENICE

At last, the final PHOENICE configuration, with the dual dilution solution implemented, is analyzed. The combination between lean and EGR solutions is impressive, the ITE is greatly increased and the combustion stability is granted by the VNT turbocharger, combined to the Swumble motion.
This combination, however, has to be correctly calibrated because the higher dilution provided by the air in lean conditions has to be balanced by a reduced EGR rate. This can be clearly seen in Figure 5.30, where the highest lean KP are coupled to at maximum 2.0% EGR rate for the 3500 x 10, that also has peak efficiency of the whole seven points analyzed. There are different lean grades, that start from $\lambda = 1.1$ up to $\lambda = 1.43$. The highest lean condition ($\lambda = 1.43$) may lead to high combustion instability and it can introduce knock tendencies, due to the lower flame speed and to the necessity to increase significantly the spark advance. This can be seen in the 2500 x 12 key-point, where knock issues reduced the ITE

performance and made clear that the maximum tolerable lean grade is the lowest, such as $\lambda = 1.1$. The same can be stated for low load and speed points, where the combustion instabilities are the strongest limit to the dilution strategy in order to further increase the efficiency.

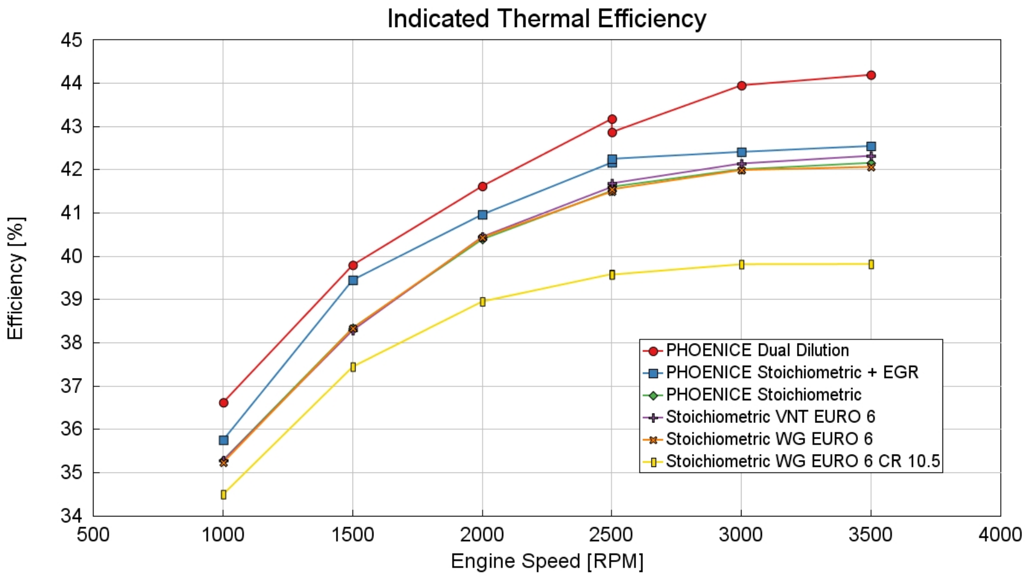

Figure 5.29: ITE comparison for each tested layout

The 2500x12 shows a strong tendency to knock, by the model prediction. This is due to the lean conditions and to the high load, generating a high temperature and pressure environment inside the chamber, with reduced turbulent flame speed. In order to prevent knock, the spark advance has to be reduced, leading to inefficiencies that affect the ITE in this KP, as it can be seen in the Figure below. It would require further investigations.

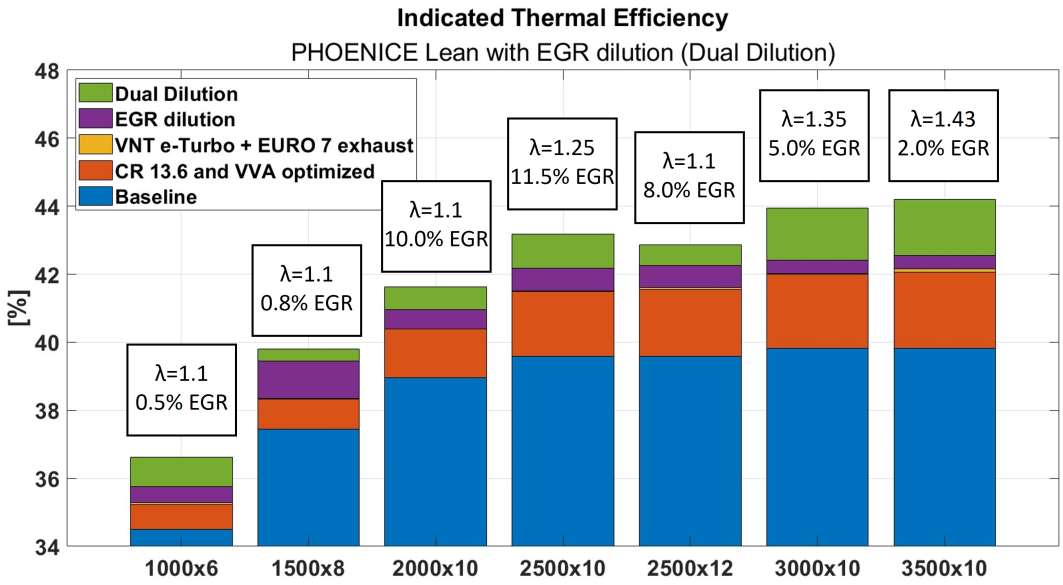

Figure 5.30: Features contributions to the ITE enhancements

The major ΔITE are linked to the combustion behaviour, combining both Stoichiometric + EGR and lean + EGR, and to the Compression Ratio increase with an optimized VVA. During the standard working conditions, the PHOENICE is always very efficient due to the high CR and to the aggressive Miller cycles, made possible by the VNT E-Turbo performances. Also, it is possible to work in EGR diluted or Dual Dilution, in order to reach the peak efficiency of the engine, for example during an homologation cycle or in RDE conditions, when the combustion is stable enough.

Chapter 6

Conclusion

The goal of the PHOENICE Project to reach 47% Gross Indicated Thermal Efficiency (Gross ITE), while reducing the emissions to be Euro7 compliant, can be achieved by implementing innovative solutions.
These are the increased Compression Ratio (CR) up to 13.6, optimized Variable Valve Actuation (VVA) to perform aggressive Miller cycles, with the help of the E-Turbo, that is important to increase efficiency, boost pressure performance and reduce turbo-lag in transient conditions. The Euro 7 exhaust is extremely important to achieve the emissions reduction goals, but it costs an overall efficiency loss, linked to the higher back-pressures that it provides. The new injection strategy too is really significant, because the increased injection pressure, up to 350 bar, and the injection split (in first and second pulse injection) provides strong performance enhancements. At last, the most important feature of the whole project, such as the Dilution strategy, based on highly diluted stoichiometric combustion, up to 20% Exhaust Gas Recirculation (EGR) rate, and Dual dilution, with EGR combined to a lean combustion, going from $\lambda = 1.1$ up to $\lambda = 1.43$. The EGR system is a Low Pressure EGR line that takes the exhaust gases after the close coupled part of the exhaust line and recirculates them at the Compressor inlet. The Dual dilution makes the Euro 7 exhaust line crucial, because in lean conditions the Nitrogen Oxides (NOx) emissions may be higher (especially for $\lambda = 1.1$) and they need to be abated by means of a Selective Catalytic Reduction system (SCR). These combustion strategies need a significant turbulence level to grant a satisfactory combustion stability. So, all the features of the PHOENICE engine contribute to generate the Swumble Motion, studied by IFP, that combines the Tumble with the Swirl turbulent motion. It is able to make aggressive Miller cycles possible, without turbulence decay issues. To enhance this motion, the piston, the intake ducts and the connecting rod are redesigned.
After the experimental data analysis and the GT model calibration, the ITE simulations show consistent results in terms of efficiency gains, in accordance with the

project expectations. The best simulation results obtained in Indicated Thermal Efficiency gain can be seen at partial load for the 3500 rpm x 10 bar Brake Mean Effective Pressure (BMEP) key-point, where a strong lean combustion at $\lambda = 1.43$ with EGR rate of 2.0% dilution stated a **net increase of + 4.38%**, with respect to the baseline performances. Similar conclusions can be made for the rest of the engine map evaluated in this thesis.
The optimized VVA and the 13.6 Compression Ratio effects lead to huge improvements in terms of ITE. The diluted combustion, when possible in terms of stability, shows great efficiency enhancements. In fact, as it can be seen in the experimental analysis, the $\lambda > 1$ + EGR is the desired condition for a large range of working conditions for the PHOENICE engine. The experimental emissions reduction, in terms of engine-out data, is significant in presence of high EGR rates. These high EGR rates, however, make the Coefficient of Variation (CoV) to raise; it has to remain under the 3% threshold, in order to avoid combustion instabilities. The best combination between emissions abatement and CoV has to be studied and tested per each working point, according to the EGR map smoothness required for the final calibration.
At the end of the project, after the transient conditions calibration, the engine will be coupled to the plug-in hybrid system of the designed test vehicle (SUV), in order to achieve a Plug-in Hybrid Electric Vehicle (PHEV) layout. In that phase, the homologation cycles, both Worldwide Harmonized Light-vehicle Test Cycle (WLTC) and Real Driving Emissions (RDE), are really interesting because the actual potential of the powertrain configuration will be displayed. It is expected to be compliant to the outcoming Euro 7 regulations by a good margin, with low fuel consumption that can be translated into low CO_2 emissions.

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