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Vehicle Fuel Economy and Performance Improvement via Cooled EGR, Charge Air Cooling and Thermal Parasitic Load

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Abstract

In recent years, stricter emissions regulations have been applied to reduce effects on the environment. Car manufacturers have to comply with these rules in order to sell cars on the market.

Different solutions are researched to reduce fuel consumed; in this project a downsized engine has been considered. Effects of different EGR rate and ACT and their impact on the cooling system are analyzed.

Having a higher amount of exhaust gas recirculated at a low temperature is affecting the power request on the cooling system, directly coupled to the engine. The latter should be operated at a higher load burning consequently more fuel.

In this project, through engine dyno tests and 1D simulations, the increase in fuel consumption will be calculated and a procedure to reduce it will be developed.

Final results showed an improvement in fuel economy up to 5% for certain conditions, without changing anything in the hardware.

To my parents, Simonetta and Francesco and my love Francesca,
for believing in me in every occasion

Acknowledgements

After an intensive period of eleven months, today is the day: writing this note of thanks is the finishing touch on my thesis. Eleven months passed since I arrived the first time in Windsor. I remember I was really scared about the place and the situation. Before leaving Italy I was questioning myself if it was the correct choice or not. But, day after day, I grew up and now I finally arrived at the end of this fantastic experience being a different person.

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"*Be the change you wish to see in the world*"
Mahatma Gandhi

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List of Abbreviations

A/C	Air Conditioning
ACT	Air Charge Temperature
$ATDC$	After Top Dead Center
BDC	Bottom Dead Center
$BMEP$	Brake Mean Effective Pressure
$BSFC$	Brake Specific Fuel Consumption
$BTDC$	Before Top Dead Center
CA	Crank Angle
CFD	Computational Fluid Dynamics
CO	Carbon Monoxide
CO_2	Carbon Dioxide
COV	Coefficient of Variation
DI	Direct Injection
DOE	Design Of Experiments
EGR	Exhaust Gas Recirculation
FC	Fuel Consumption
FTP	Federal Test Procedure
H_2O	Water
HC	Hydrocarbons
HP	High Pressure
ICE	Internal Combustion Engine
$IMEP$	Indicated Mean Effective Pressure

LP	Low Pressure
MBT	Minimum Spark Advance for Best Torque
$NMHC$	Non-Methane Hydrocarbons
NO_x	Nitrogen oxides
$OFAT$	One Factor At Time
PM	Particulate Matter
$PMEP$	Pumping Mean Effective Pressure
rpm	Revolutions per minutes
SI	Spark Ignition
$stoich$	Stoichiometric
TDC	Top Dead Center
VGT	Variable Geometry Turbine

List of Symbols

ε	Heat Exchanger Effectiveness
η_V	Volumetric Efficiency
λ	Excess Air Factor
μ	Mean
σ_{SB}	Stefan-Boltzmann constant
σ	Standard Deviation
τ	Torque
A	Surface Area
c_p	Specific heat at constant pressure
h	Fluid Convective coefficient
i	Number of Cylinders
k	Thermal Conductivity of the material
K_r	Global Heat Transfer Coefficient
L	Surface Length
l_f	Number of levels for factor f
m_{air}	Mass of air
$\dot{m}_{air\ actual}$	Actual air mass flow rate
$\dot{m}_{air\ ideal}$	Ideal air mass flow rate
m_{EGR}	Mass of Exhaust Recirculated Gas
m_{fuel}	Mass of fuel
n	Engine rotational speed
n_f	Number of factors with level l_f

NTU	Number of Transfer Units
P_e	Engine Power
p_{me}	Engine Effective Pressure
$\dot{Q}$	Heat flux
$\dot{q}_{CD}$	Heat flux per unit area due to conduction
$\dot{q}_{CV}$	Heat flux per unit area due to convection
$\dot{q}_R$	Heat flux per unit area due to radiation
T	Temperature
V_c	Piston Displacement

Chapter 1

Introduction

In the last years, the automotive companies have tried to develop strategies to reduce the amount of fuel consumed by cars as much as possible. This trend is not related only to an increase in fuel cost, which, in contrast, has shown a decreasing behavior since the mid of 2013, as reported in Figure 1.1.

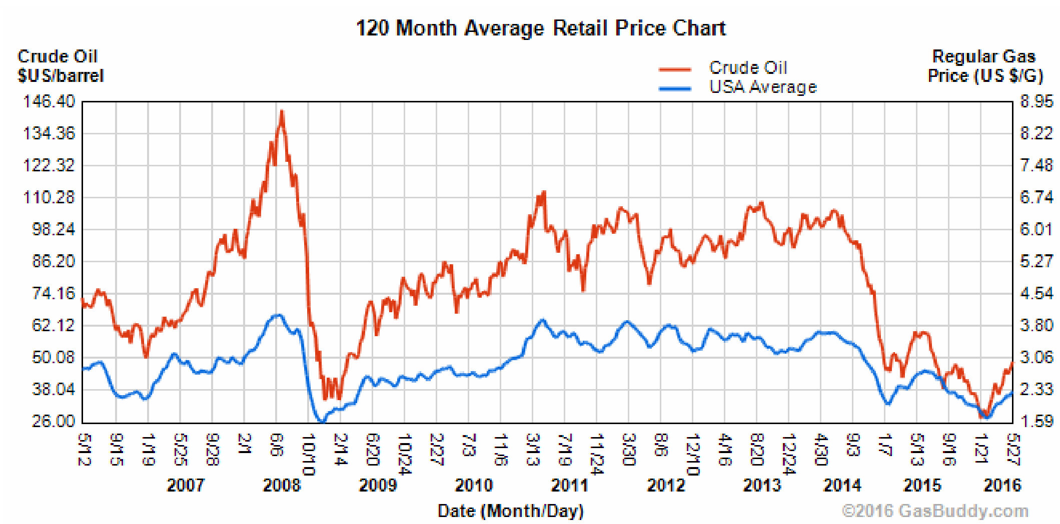


Figure 1.1: *USA average and crude oil price trend in the past 10 years [1].*

More than a decade ago, the attention to fuel efficiency was mainly for an economic reason; however, now the constraints are set by more stringent government regulations.

One of the exhaust components on which companies are dedicated is CO_2 since it represents one of the data considered also by some customers. However, together with CO_2 emissions there are other pollutants to look at when considering the emissions, especially with the EURO 5 and EURO 6 standards which have introduced new limits to non-previously regulated pollutants, such as NMHC (Non-Methane Hydrocarbons), NO_X (nitrogen oxides) and PM (Particulate Matter), the latter has been introduced for gasoline engines, while it was already present for Diesel engines. Car manufacturers need to strictly respect the limits set by the legislations in order to sell and gain market share.

In order to comply with these set of rules, many solutions on various fields are considered. They include lightweight materials and more aerodynamic designs for the chassis and more efficient powertrains. The first step to move further in this field, however, could be individuated in downsizing the engine.

Downsizing, as will be explained in the first part of this thesis, has been highlighted as one of the most promising technique in the near future to reduce fuel consumption and emissions. However, with its usage there are some drawbacks that should be taken into account.

In this project a reduced displacement engine has been considered together with its interaction with the cooling system.

1.1 Objective Statement

The main objective of this project is to evaluate the fuel consumption increment due to the presence of the cooling system and find the best position for the different actuators to minimize this parameter.

In literature and in previous studies, when the engine is tested on the dynamometer, the effects of the engine cooling system are not accounted for since it is not physically present. Data on the fuel consumption are collected without taking into consideration the additional frictional loss caused by additional components (i.e. cooling fan, etc.). If the engine power output required at the wheels is set to a certain value, the engine must provide additional power to operate the added cooling system.

Following this concept, the key idea behind this project is to account for the additional load and convert it to additional fuel consumption. The main inputs that will be considered to evaluate the minimum fuel consumption are the Exhaust Gas Recirculation (EGR) rate and the Air Charge Temperature (ACT). These two are strictly related to the positions of the different actuators in the cooling system and so, a direct correspondence between them could be found and used to operate the actuators to obtain the lowest fuel consumption.

1.2 Procedure

This project was performed in three separate phases:

1. Tests on the engine dynamometer are performed varying EGR rate, manifold temperature, and load working point. The data collected in this phase have been used as inputs for the next ones.

2. Simulations on the cooling system are performed using a 1D thermo-fluid software. In this phase, data from various components of the cooling system are collected and used for the final calculations on the fuel consumption.

3. The data collected in the previous phases are analyzed, and a strategy has been developed to calculate and find the minimum fuel consumption point. The final output is the best actuators configuration to achieve the lowest fuel consumption at every speed and load condition.

1.3 Thesis organization

- In Chapter 2 the useful background to better understand the scope of the project as well as some information from previous studies found in literature are presented. In this part, the main concepts, advantages and disadvantages of downsizing, and the additional components needed for the application of this technique are reported.

- In Chapter 3 the methodology used during the test on the engine, the simulations, and the data analysis are explained in detail.

- In Chapter 4 the results obtained during the development of this project are reported. This chapter is fundamentally divided into two parts. The first one discusses data analysis of the engine experiment. The second part explains the final results, which are based on the model simulations using the engine test data.
- In Chapter 5 final conclusions are drawn and presented.
- In Chapter 6 suggestion for future work and possible improvements are reported.

Chapter 2

Background and Literature Review

2.1 Engine Downsizing

Engine downsizing is a very promising technique in reducing fuel consumption and CO_2 emissions. It is mainly based on the reduction of the engine displacement, while maintaining the desired engine output. With this technique smaller and lighter engines can be produced. However, it is necessary to increase the torque at a given engine speed in order to produce the same power as that of non-downsized engines. In Figure 2.1 it is shown the basic concept of downsizing: the reduction of the displacement, in this case followed also by a reduction in number of cylinders, but obtaining the same power output. In the smaller engine, on the right, it is possible to notice the addition of the turbocharger to recover the loss in power.

In the following a deeper explanation of the working principle, the main additional components needed, the benefits and drawbacks of this technology will be addressed.

2.1.1 Working Principle

The concept behind engine downsizing, as stated before, is reducing the swept volume of the cylinders maintaining the same power output. In doing so the smaller engine is forced to work at higher mean effective pressure with a consequent higher efficiency, as it is shown in Figure 2.2, where the BSFC, which can be directly related to efficiency, are plotted.

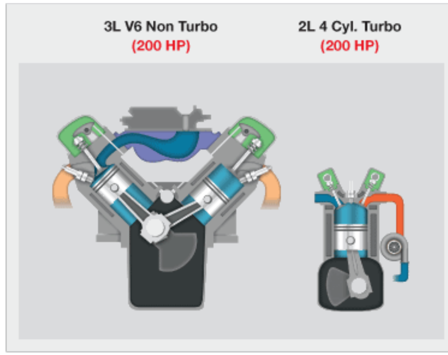


Figure 2.1: *Example of a downsized engine [2].*

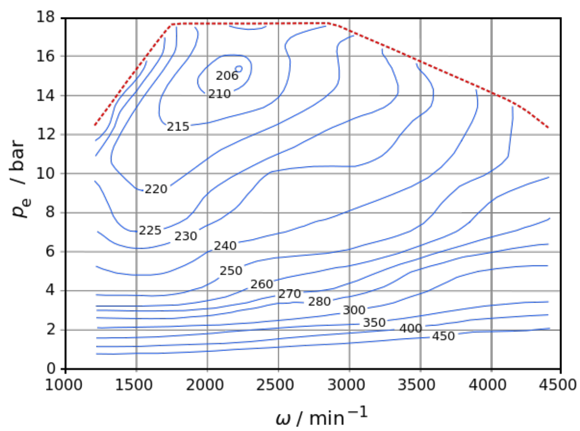


Figure 2.2: *Engine speed load map with iso-BSFC curves [3].*

In everyday driving conditions an engine is most of the times operated partially throttled at low speed and low loads, lowering the efficiency and consequently increasing the fuel consumption. Thus the key factor of downsized engines is to have a very high power density: defined as P_e/V_c, and there are two possible ways to increase its value:

- increasing engine speed n
- increasing engine effective pressure p_{me}

These parameters are related each other through the following formula:

$$\frac{P_e}{V_c} = i \cdot n \cdot p_{me} = 2\pi \cdot n \cdot \frac{\tau}{V_c} \tag{2.1.1}$$

Where i is the number of cylinders, τ is the torque and V_c is the piston displacement. It is clear how the engine speed and the effective pressure could positively influence the power density [4].

High speed could help in increasing the power density, but it has the drawback, as described by Chen-Flynn empirical correlation, of increasing the frictional loss in a great extent. The correlation found by the two shows that speed is affecting more the friction mean effective pressure rather than load [5]. Consequently a significant increase in fuel economy, then, can only be achieved employing the so called high-load concept (i.e. higher mean effective pressure). In this case possible drawbacks could be related to engine packaging, due to complex charging systems and additional components that need to be developed and added to the engine system; however, the overall improvement in fuel efficiency is still very promising.

2.1.2 Downsizing benefits

The main benefits achievable with the downsizing technique can be easily summarized in the following:

- **Fuel consumption reduction:** this is the greatest benefit resulting from this technology. It is the results of the operating range at higher pressure and consequently higher efficiency. It is beneficial for future car manufacturers to comply with the stringent emissions standards. Police et al.[6] have showed, for example, a reduction in fuel consumption ranging $11\% - 20\%$.

- **Warm-up time reduction:** having a smaller engine means also a reduced time to reach its operating temperature. The coolant, the oil and all engine fluids reach their nominal working temperature faster, reducing transient operations which impair the fuel economy. In fact the viscosity of coolant and oil, which is related to friction, is a function of temperature; the lower the temperature, the higher the losses due to increased friction, and consequently the fuel consumption will be higher.

- **Mass reduction:** the choice of a smaller engine assures also that the weight of this component is reduced. It is true that the mechanical stress on the engine components will increase to whistand higher load. However, with the usage of lighter and stronger materials, a downsized engine can help in recovering some weight which is beneficial for a further fuel consumption reduction. It is also true that the additional components employed such as the charge air cooler, the turbocharger and the EGR cooler could hinder the positive effects of having a lighter engine.

- **Losses reduction:** additional benefits of a smaller engine can be found looking at the reduced sliding surfaces and the smaller friction generated by their relative motion. In a standard engine, friction losses become preponderant, especially at low loads; so the benefits of increasing the mean effective pressure can also be correlated to that point [6]. It is widely demonstrated that, in normal driving conditions, only the 30% of the fuel energy is converted to effective power, the remaining part (70%) is lost and redistributed as follow: 30% to the coolant, 30% to the exhaust gases, 10% in friction and parasitic losses, as shown in Figure 2.3. Most of the engine friction is caused by three components: piston-assembly, bearings and the valve train [7].

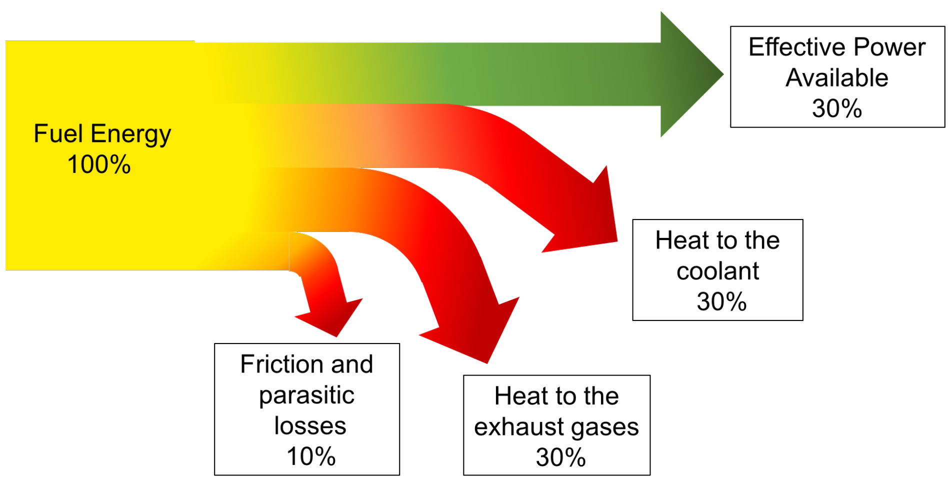


Figure 2.3: *Heat losses distribution in ICE.*

Moreover, another characteristic of downsized engines is that they are operated at less throttled conditions. This unthrottled operation, together with the increased usage of exhaust gas recirculation (EGR) technology, helps in reducing pumping losses.

- **Emissions reduction:** a last benefit of downsizing is related to the reduction in emissions. It represents the direct result of the fuel economy achievable with the usage of smaller engines and it is one of the most important parameter searched by car manufacturers.

2.1.3 Downsizing disadvantages

Although the benefits coming from the engine downsizing concept look very attractive, there are some drawbacks and problems to be considered.

First of all, in order to maintain the same power output, the engine should be able to produce higher torque. This increased torque can be achieved using a forced induction device, such as turbocharger. The application of a turbocharger is particularly helpful since it increases air mass flow rate, and fuel flow rate, consequently. The greater the amount of air, the greater the fuel injected and the higher the available power. A schematic of the concept is show in Figure 2.4. The dashed curve represents a naturally aspirated, with a lower available torque, the continuous curve instead represents the same engine, but boosted.

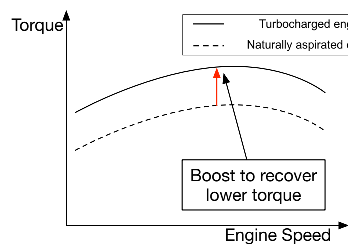


Figure 2.4: *Torque curve comparison between naturally aspirated and turbocharged engine.*

The general trend, as stated above, is to increase the brake mean effective pressure as much as possible without increasing the engine friction significantly. Some main drawbacks can be summarized in the following:

- **Pressure and temperature increase:** these are two direct effects of boosting the engine. Since the air, at the outlet of the turbocharger, is compressed, it increases in density but also in temperature. The higher charge temperature will result in increased tendency of abnormal combustion, such as engine knock. This can cause engine components failure. Many strategies have been developed to overcome this problem. They include additional fuel injection for cooling purposes, charged air cooler (CAC, and exhaust gas recirculation, which will be better discussed along the project.

- **Mechanical stresses increase:** the increased stresses on the engine components is another side effect of turbocharging. As mentioned above, the combustion is carried out at higher pressure and temperature, imposing higher mechanical and thermal loads on the structure. The application of stronger and more resistant materials is mandatory, always looking at their weight and costs.

- **Cost and number of components increase:** this point is mainly related to the innovative technologies and materials that are present behind the development of powerful smaller engine. To assure a good downsizing result, a new generation engine can easily have many additional components depending on the solutions applied. For example, direct injection, CAC, and both high pressure and low pressure EGR circuits can easily increase the cost of the engine as a whole and great attention must be paid in order to minimize their economical impact.

As an overall consideration, engine downsizing, despite the disadvantages highlighted above, can be considered as a powerful technique for the reduction of fuel consumption and consequently in the emissions. As a matter of fact, more and more car companies are developing smaller engines, meaning that the research on this field is going on at a fast pace. With the increase of knowledge, the main problems are being faced and overcome. In the following sections the main components that are added to a downsized engine are analyzed, considering also some advantages and problems related to their implementation.

2.2 Turbocharger

The maximum power a given engine can deliver is limited by the amount of fuel that can be burned efficiently inside the engine cylinder. This is limited by the amount of air that is introduced in every cycle. If the inducted air is compressed to a higher density prior to enter in the combustion chamber, the maximum power an engine of fixed dimensions can

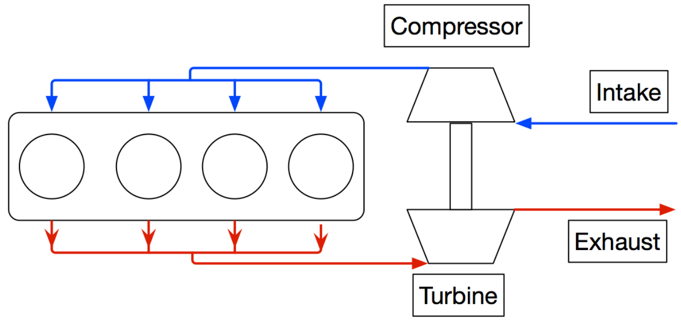


Figure 2.5: *Schematic of turbocharger circuit.*

deliver will be increased. This is the primary purpose of turbocharging [7].

Supercharging Vs Turbocharging It is worth to mention that, before the extensive usage of turbochargers, the first improvement in the increase of air density was made using superchargers.

The base principle is the same. The main difference is that superchargers are directly coupled to the engine's crank shaft, while turbochargers utilize enthalpy in the exhaust gas. They show a faster response in transient conditions (turbo lag, characteristic of turbochargers, is not present), but worse behavior at low engine speeds. The superchargers impose additional friction loss on the engine. On the other hand turbochargers are capable of recovering part of the waste exhaust energy, although they place exhaust back pressure on engines, increasing pumping losses.

2.2.1 Operating principle

The turbocharger comprises a centrifugal compressor powered by a turbine that is driven by the engine's exhaust gas. Hot exhaust gas flows through the turbine's wheel blades, accelerating it and driving the compressor. These turbines are generally made from high-temperature resistant nickel alloys, and can withstand very high temperatures (in the range of 1000°C) and speeds (up to 280,000 rpm). The compressor itself comprises an impeller and a diffuser. The impeller draws in air, accelerating it to a high velocity before forcing it towards the diffuser. The diffuser slows the fast-moving air, raising the pressure and temperature in the compressor housing, compressing the air before it is directed to the engine. In this way, more air can be injected into the combustion chamber, ready to burn the additional fuel needed to maintain stoichiometric conditions and obtain so that more engine power. To prevent the turbocharger from overcharging at high engine speeds, and also to maintain torque at lower engine speeds, the flow of exhaust gases through the turbine and compressor is carefully controlled. At high engine speeds or low load conditions, a wastegate is opened to divert part of the exhaust gas flow away from the turbine, decreasing pressure in the compressor housing. Meanwhile at low engine speeds, the wastegate will close so that the entire exhaust flow can drive the turbine and the compressor [8].

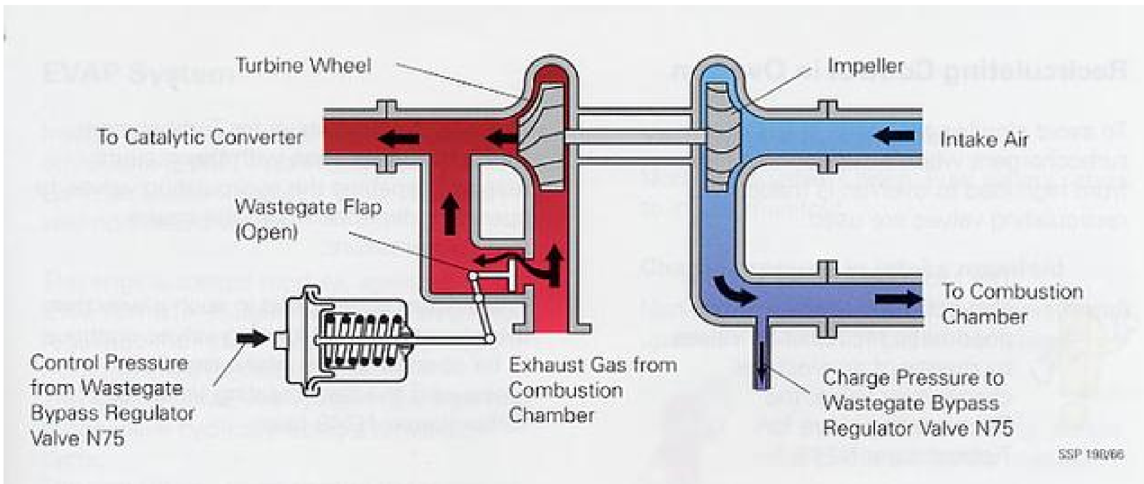


Figure 2.6: *Turbocharger working schematic, with waste gate valve [9].*

2.2.2 Turbocharger schemes

Dealing with turbocharger there are many different types that can be employed. They mainly differ one from the other in construction method or layout. A wide description of all of them is out of the scope of the project, but few words are worth to be spent on this argument [10]. The main advanced schemes which are used are:

- twin-turbochargers
- twin-scroll turbochargers
- variable geometry turbochargers (VGT)

- electric turbochargers

Twin-turbochargers: It is based on the application of two different turbochargers, and there is further distinction to be done in this family depending on the layout: a) parallel twin-turbo b) sequential turbocharging c) staged turbocharging.

Paralleled twin-turbo is the configuration in which two identical turbochargers are used at the same time, each sharing half of the engine's exhaust gases. Commonly used in a V-shaped engine, one turbo feeds off the left bank of cylinders, and the other feeds off the right bank.

Sequential turbocharging is the set-up in which the engine uses one turbocharger for lower engine speeds, and a second or both turbochargers at higher engine speeds. Clearly the two turbochargers have different dimensions: the smaller of the two operates at low speeds and the larger turbo starts working at high speeds. Sequential twin turbos are often called "two-stage turbo" because the smaller turbo will actually continue to run and feed the larger turbo when it activates.

Staged turbocharging can be used when the output pressure must be greater than the one which can be provided by a single turbo. In this case, multiple similarly sized turbochargers are used in sequence, but both operate constantly. The first turbo boosts pressure as much as possible and feeds the second one to increase it even more.

Twin-scroll turbochargers: it is a system design developed to overcome some problems of single-scroll turbo systems by separating those cylinders whose exhaust gas pulses interfere with each other. In this way the kinetic energy from the exhaust gases is recovered more efficiently by the turbine. For example, if a four-cylinder engine's firing sequence is 1-3-4-2, cylinder 1 is ending its expansion stroke and opening its exhaust valves while cylinder 2 still has its exhaust valves open. In a single-scroll or undivided manifold, the exhaust gas pressure pulse from cylinder 1 is therefore going to interfere with cylinder 2's ability to expel its exhaust gases, rather than delivering it undisturbed to the turbo's turbine. In such a way a better pressure distribution in the exhaust ports is obtained and as well as a more efficient delivery of exhaust gas energy to the turbine. The main result in the intake is that an improved quality and quantity of the air charge is entering each cylinder [11].

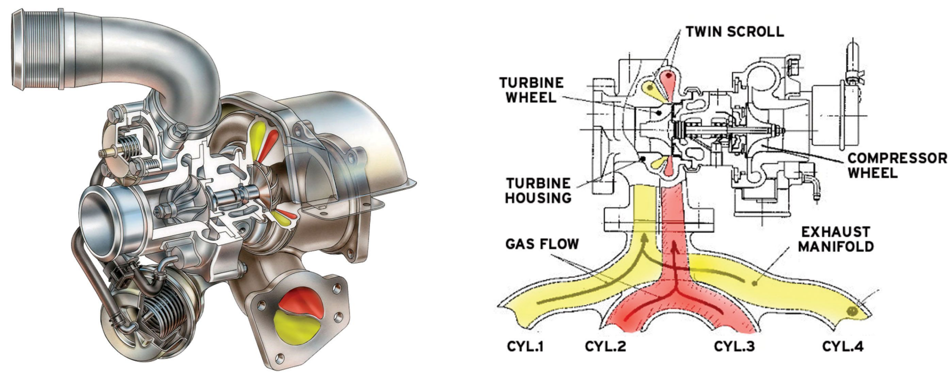


Figure 2.7: *Twin-scroll turbocharger configuration [12].*

Variable geometry turbochargers (VGT): this solution was developed to overcome the need of different size turbochargers at different engine speeds. An oversized turbocharger would be ineffective at low speeds since it would not be able to create a sufficient pressure rise for proper operations. At the same time an undersized one will choke the engine at high speeds, creating high exhaust manifold pressure and higher pumping losses.

The concept behind this component is changing aspect ratio depending on engine speed and the functioning elements capable of achieving the result are moveable vanes which adjust the air-flow to the turbine. The vanes are placed just in front of the turbine like a set of slightly overlapping walls. As rpm rise and exhaust pressure increases, the vanes open, so as to allow all exhaust gasses to the turbine. An example of VGT turbocharger is reported in Figure 2.8.

Electric turbochargers: it is a new developing technology that is spreading at a slow pace starting from luxury and sport cars. The concept behind this technology is the usage of an electric motor coupled to the compressor shaft, as shown in Figure 2.9. The turbo lag can be completely eliminated, due to the fast response of the electric motor.

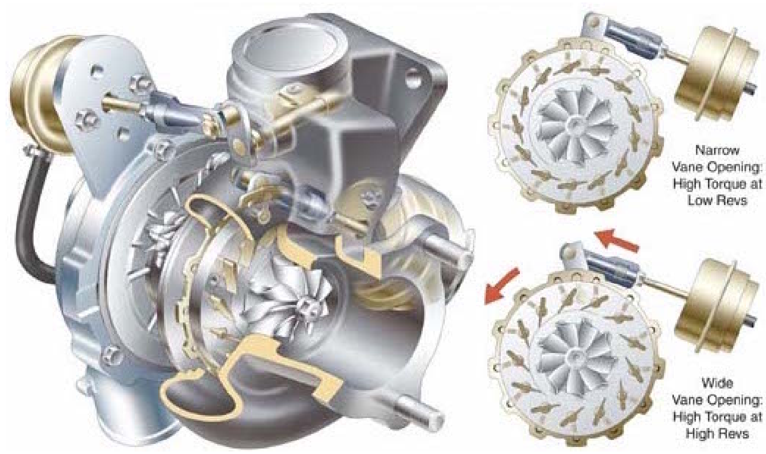


Figure 2.8: *Variable geometry turbine schematic [13].*

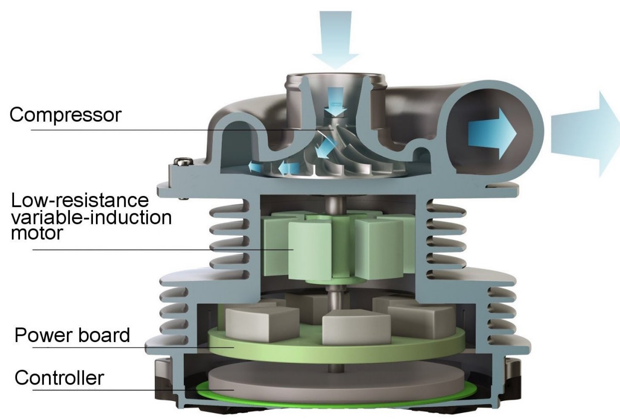


Figure 2.9: *Electric turbocharger schematic [14].*

2.2.3 Turbocharger advantages

It is important to recall that, in the past, the concept of increasing the pressure or density of the intake was present but with the objective of increasing the power output keeping the same engine size. Nowadays, the basic idea is still present, but it is applied on smaller engines and, as extensively stated above, turbocharging has been found as a key technological improvement to meet the stricter requirements in terms of fuel consumption and emissions production.

It is worth to stress that in this project, when talking about turbochargers, it is impossible to separate them from the concept of downsizing. Bearing in mind this synergy, a brief summary of the advantages coming from its usage are presented below [15]:

- **Power output:** generally speaking the power obtained from a turbocharger engine is higher than that from a naturally aspirated one, if the displacement is the same. In downsized engine this effect is exploited in order to have the same power output with a smaller engine. This fact implies lower frictional and thermal losses.

- **Efficiency increase:** to run the compressor exhaust gases are expanded in the turbine, this will lead to an increase in the engine efficiency because part of the wasted energy is recovered.

- **Fuel consumption reduction:** it has been proved that turbocharging is effective in reducing fuel consumption, up to 20% [16], mainly due to the smaller and more efficient engine.

- **Emissions reduction:** this is a direct consequence of burning less fuel and having a greater efficiency. It represent one of the reasons why most car manufacturers are moving toward turbocharging/downsizing techniques.

2.2.4 Turbocharger disadvantages

So far only the advantages of the turbochargers have been highlighted, but this solution is not free of possible drawbacks. In this section the main disadvantages will be adressed and presented in a schematic way, so that they can be easily understood. The same considerations made above will be applied now: it would be impossible to completely separate the turbocharging effect from the downsizing concept.

Here are the main disadvantages needed to overcome:

- **Turbo lag:** it is the time spent between the change in power output after a change in the pedal/throttle position. It is perceived as a moment of hesitation in obtaining the required power, and it is related to the dynamics of the turbine and the exhaust gas expanding in it.

- **Boost threshold:** traditional turbochargers are often sized for a certain *rpm* range where the exhaust gas flow is adequate to provide additional boost for the engine.

The lower limit of this range is related to the kinetic energy of exhaust gases, if it is too low, especially at low rpm, the compressor is not able to produce a sufficient amount of work.

- **Material requirement:** turbochargers in contact with exhaust gas need to withstand high temperature for the duration of the engine life. Together with a good cooling system also high temperature resistant materials should be used, leading to an increased capability of exploiting higher temperature gas.

- **Knock:** this phenomenon is very important and is related to the high pressure and temperature, a deeper discussion on it will be given in the following.

Knock

It is considered one of the main drawbacks of having highly boosted engine, and it is the limiting factor to the maximum BMEP that the engine can produce.

The phenomenon behind this problem is related to abnormal combustion inside the cylinder. Knock is the name given to the noise which is transmitted through the engine structure and it is the result of spontaneous ignition of the end-gases, composed of fuel and air. It is characterized by an extremely fast release of chemical energy in the end-gas, resulting in very high local pressures propagating all across the combustion chamber. Knock is governed mainly by pressure and temperature in the combustion chamber, it is clear then why turbocharging is so critical for the onset of this problem.

Generally speaking, the term knock is referred to an abnormal combustion which lead to the audible metallic sound, but there is a further characterization that could be made:

- **Spark knock:** is recurrent and repeatable but it is controllable by the spark advance: advancing the spark increases the knock intensity and retarding reduces the intensity

- **Surface ignition:** is ignition of the fuel-air charge by any hot surface other than the spark discharge prior to the arrival of the normal flame front.

The onset of knock is not completely well understood, there are a huge amount of possible factors affecting its presence; the fuel characteristic for example is a factor to take into consideration.

Knock represents also a problem for downsized engines since it primarily occurs under wide-open-throttle operating conditions, typical conditions that could be found in these types of engines. It is thus a strong constraint since it limits temperature and pressure during the combustion phase and the compression ratio, reducing the possible thermodynamic efficiency gain coming from a higher compression ratio engine [7].

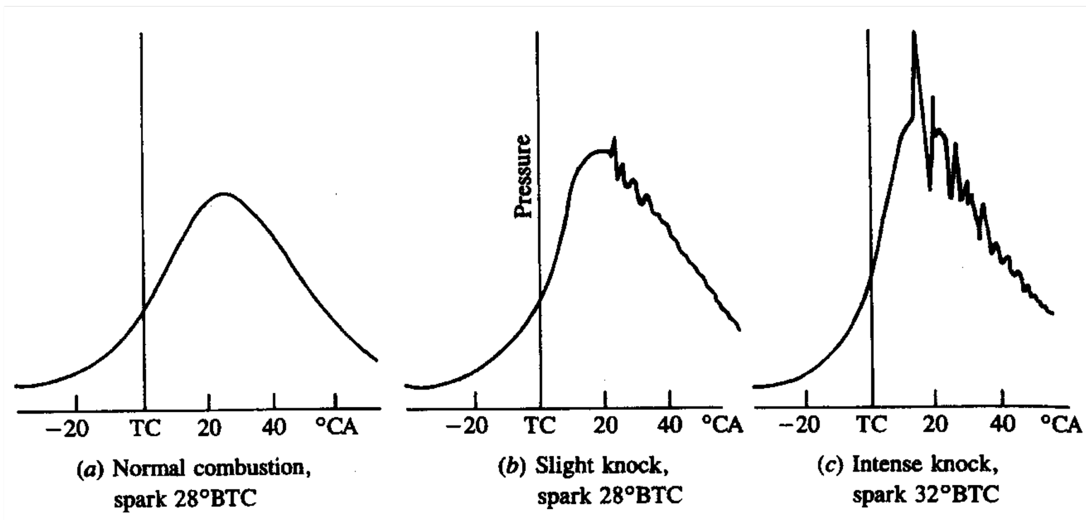


Figure 2.10: *Cylinder pressure versus crank angle traces of cycles with (a) normal combustion, (b) light knock and (c) heavy knock. 4000 rev/min, wide open throttle, 381* cm^3 *displacement single-cylinder engine [7].*

Since knock is potentially harmful for the engine a good detection strategy should be implemented, in such a way it is possible to act properly to overcome this phenomenon. In Figure 2.10 different combustion pressure characteristic are showed, *(a)* represents a normal combustion; observing *(b)* it is possible to note some pressure variations, they represent the onset of knock. *(c)* instead is an intense knock and the pressure variations are very high. Once it is detected, boosting pressure as well as spark timing should be changed to meet proper combustion characteristics.

Knock reduction strategies

In the following possible strategies to reduce knock are presented, some of them will be discussed more in detail along the project.

- **Fuel characteristics:** as previously stated a premium gasoline could influence engine knocking behavior. The octane number as well as longer chain hydrocarbons positively reduce knock presence and intensity. Also fuels with higher latent heat of evaporation like Ethanols, which have also higher octane number, could be seen as a good solution in reducing knock tendency.

- **Air/Fuel ratio:** it is known that knock is at its maximum when the excess air facto (λ), defined as follow, is around

  $$\lambda = \frac{(\frac{m_{air}}{m_{fuel}})}{(\frac{m_{air}}{m_{fuel}})_{stoich}} = 0.9$$

  and it decreases both in enrichment and lean conditions [6]. Mixture enrichment is a technique that has beed used for a long time. It consists in injecting extra fuel so that, when it enters in the combustion chamber, its evaporation reduces temperature and consequently possibility of knocking. This solution worsens fuel economy because the additional fuel injected is not used to produce useful energy. Both conditions, rich and lean mixtures, show the main drawback of impairing catalytic converter efficiency, in fact, to operate at its best efficiency the three-way catalytic converter needs the mixture to be in stoichiometric conditions, as shown in Figure 2.11.

- **Direct injection (DI):** it is widely considered as the key technology in downsized boosted engines. It allows to operate engines at higher compression ratios, it reduces in-cylinder temperature due to the latent heat of evaporation of the fuel that completely evaporates in the chamber and not in the intake, like port injected engines [17]. Advanced and more precise fuel injection strategies can be implemented such as multiple injection, stratified mixture or additional injections during cold starts to reduce the warm up time [6].

- **Variable compression ratio:** this technology is based on the fact that higher compression ratio directly increase engine efficiency, but the limiting factor, most of the

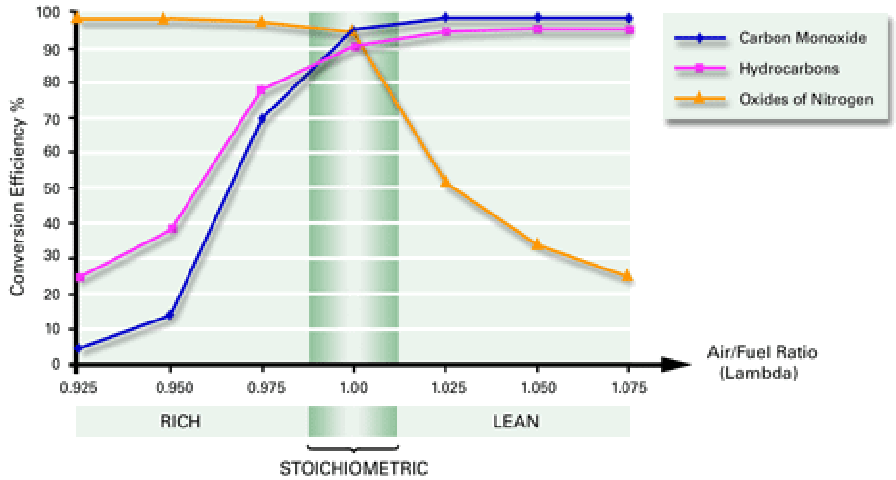


Figure 2.11: *Effects of Air Fuel ratio on catalytic conversion efficiency [20].*

times, and especially in boosted engines, is set by knock. The concept behind variable compression ratio is to use an high compression ratio for light to medium load and, as soon as the load is starting to increase and knock start to appear, reduce the compression ratio in order to limit the side effect of abnormal combustion. This technology is still too expensive to be implemented in commercial vehicle and it is used on laboratory engines.

- **Combustion phasing:** as stated before, a way to reduce knock is to change the spark timing. As soon as the knock is detected spark should be retarded, in such a way its effect is mitigated but the thermodynamic efficiency of the engine decreases [7]. It is a strategy commonly used to a certain extent; combining spark timing with other technologies as direct injection, exhaust gas recirculation and charge air cooling leads to the possibility of eliminating spark retard improving combustion characteristic while reducing fuel consumption [18, 19].

- **Intake air cooling:** this method consists in reducing the temperature of the charge entering the cylinder, it will be discussed more in detailed in the following.

- **Exhaust gas recirculation:** a part of the exhaust can be drawn and injected in the intake manifold, also for this technology an extensive discussion will be done in the following.

The methods highlighted represent some of the most used solutions to reduce knocking, other systems can be implemented, but their description is out of the scope of this project.

2.3 Charge Air Cooler - CAC

The charge air cooler, also known as intercooler, is one of the fundamental components needed together with the turbocharger. The term intercooler is more correct when dealing with two stage compressors since it refers to the cooling action on the air between the first and second compressor, a more correct name should be aftercooler, but the term intercooler nowadays is widely spread and accepted. In downsized engines, it represents one of the first way to reduce many of the disadvantages coming from the usage of high boosting pressure.

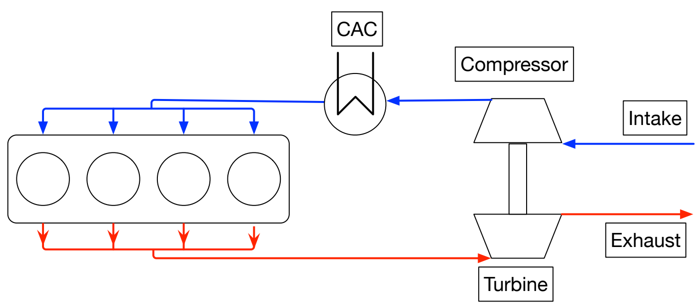


Figure 2.12: *Turbocharger schematic with charge air cooler component*

As it is shown in Figure 2.12, the charge air cooler is placed after the turbocharger, right before the cylinders intake. Its function is to reduce the temperature of air entering the combustion chamber, giving rise to beneficial effects as the increase in air density, the reduction of knock and other effects.

The thermal duty of charge air cooler is very strong, in fact it needs to subtract enough heat to reduce the temperature from around 100 - 200°C down to 60 - 50 and sometimes also 40°C, depending on the working conditions considered. Air charge temperature is therefore a parameter which could affect in a great extent combustion stability and fuel consumption. Generally speaking, having a colder mixture entering the combustion chamber is considered favorable for a volumetric efficiency point of view. In a simplistic way it could be defined as mass of air entering the cylinder over the mass of air which could be drawn in the cylinder $\eta_V = \frac{\dot{m}_{air\ actual}}{\dot{m}_{air\ ideal}}$, where $\dot{m}_{air\ actual}$ is the air mass flow rate actually drawn in the cylinder and $\dot{m}_{air\ ideal}$ is the air mass flow rate theoretically admissible in the cylinder if all the volume would be occupied by air. Considering this relation, it is easy to understand that increasing the density (higher mass for the same volume) of air entering the cylinder is directly affecting the volumetric efficiency. Turbocharging is thus increasing the density, but it also increases the temperature, which is inversely proportional to density, so the effect is not fully exploited; CAC is adding a further improvement lowering the temperature. For this reason the usage of charging technique in general should be always followed by a reduction in temperature, especially at high loads where the possibilities of having knock are much more increased.

2.3.1 CAC: Heat transfer modes

A brief summary on the heat transfer modes that are present in an engine and in heat exchangers in particular will be useful to better understand the functioning of the different types of charge air cooler which will be analyzed in the following.

Conduction

Conduction is the mode in which heat is transferred by molecular motion, through solids and through fluids at rest, due to a temperature gradient. The heat transfer per unit area per unit time in a steady situation is given by *Fourier's law.* Considering one-dimensional temperature variation along the x direction, it can be written as follows:

$$\dot{q}_{CD} = \frac{\dot{Q}}{A} = -k\frac{dT}{dx} \tag{2.3.1}$$

Where $\dot{Q}$ is the heat transfer, A is the surface area through which the heat is transferred, k is the thermal conductivity of the material, and $\frac{dT}{dx}$ is the variation of temperature along the length of the element.

Heat is transferred by conduction through the cylinder head, cylinder walls, and piston; through the piston rings to the cylinder wall; through the engine block and manifolds; in the metal parts of the heat exchangers [22].

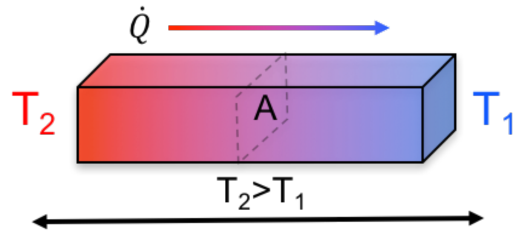


Figure 2.13: *Conduction heat transfer in a solid.*

Convection

Convection is the mode in which heat is transferred through fluids in motion and between a fluid and a solid surface in relative motion. If the fluids are in motion under the effect buoyancy force it is called *natural convection*, if the reason of the motion is an external source the heat transfer mode is known as *forced convection*. In steady-state condition, the heat flux transferred by convection from the moving stream of gases to the cylinder walls can be written using Newton's law of cooling as follows:

$$\dot{q}_{CV} = \frac{\dot{Q}}{A} = h(T_{hot} - T_{cold}) \tag{2.3.2}$$

Where h is the convective coefficient of the fluid considered, T_{hot} and T_{cold} are respectively the temperature of the hot and cold gases facing the heat exchanger.

Heat is transferred by convection through the in-cylinder gases and the solid parts surrounding the combustion chamber; from the cylinder head and walls to the coolant and from the piston to the lubricant oil; from the hot air to the metal parts of heat exchangers and from metal parts to the cooling fluid. Heat by convection is also directly transferred from the engine to the environment [22].

Radiation

Radiation is the mode in which heat is transferred by emissions and absorptions of electromagnetic waves. Radiation occurs from hot temperature in-cylinder gases to the cylinder walls and from the hot external surfaces of the engine to the environment. The heat flux from one "black body" at temperature T_1 to another at temperature T_2 across a space that does not contain absorbing material is written as follows:

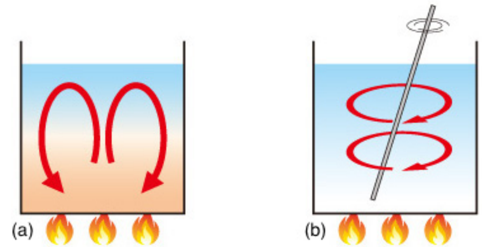


Figure 2.14: *Example of* (a) natural *and* (b) forced *convection. In* (a) *the driving force is represented by buoyancy force, while in* (b) *is the stirring action [21].*

$$\dot{q}_R = \frac{\dot{Q}}{A} = \sigma_{SB}(T_1^4 - T_2^4) \tag{2.3.3}$$

Where σ_{SB} is the Stefan-Boltzmann constant. Actually gases are not a "black body". This difference from black-body behavior is usually taken into account by applying an emissivity factor ε (reduction factor lower than 1).

The radiation term is lower with respect to conduction and convection and is generally negligible for SI engines, even more in heat exchangers [22].

Heat transfer in heat exchangers

The charge air cooler is an heat exchanger in which a cold fluid, which can be either air or a liquid (most of the times water), is forced to pass along the metallic surface which in turn are in contact with the hot gases which must be cooled.

The modes which regulate the heat transfer in steady state conditions are:

- **Convection on hot side:** between hot gas (from the compressor) and heat exchanger metal surface
$$\dot{q}_{CV} = h_{air}(T_{air} - T_{wall,air}) \tag{2.3.4}$$
- **Conduction:** inside the metal elements of the heat exchanger
$$\dot{q}_{CD} = \frac{k}{L}(T_{wall,air} - T_{wall,cool}) \tag{2.3.5}$$
- **Convection on coolant side:** between the heat exchanger metal surface and the

coolant (independently on the type)

$$\dot{q}_{CV} = h_{cool}(T_{wall,cool} - T_{cool}) \tag{2.3.6}$$

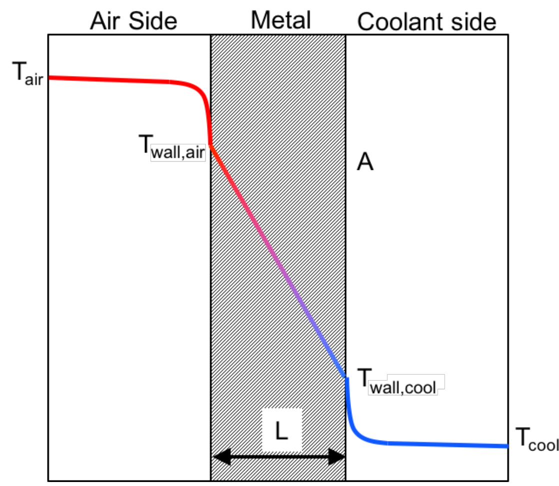


Figure 2.15: *Heat transfer modes in a heat exchanger*

It is possible to observe, as stated before, that no radiation term is present since its contribution is very low and can be neglected. For what concern the convective term it is enhanced if the flow is turbulent and the heat exchange area is wide enough, with these considerations the design of these components is fairly important.

As a general discussion charge air cooler can be analyzed, with electric circuit similarities, considering the whole system as a series of thermal resistances, as shown in Figure 2.16.

The values of the resistances can be retrieved as follow:

- Air side

$$T_{air} - T_{wall,air} = \frac{1}{h_a}\dot{q} \tag{2.3.7}$$

- Metal side

$$T_{wall,air} - T_{wall,cool} = \frac{L}{k}\dot{q} \tag{2.3.8}$$

- Coolant side

$$T_{wall,cool} - T_{cool} = \frac{1}{h_c}\dot{q} \tag{2.3.9}$$

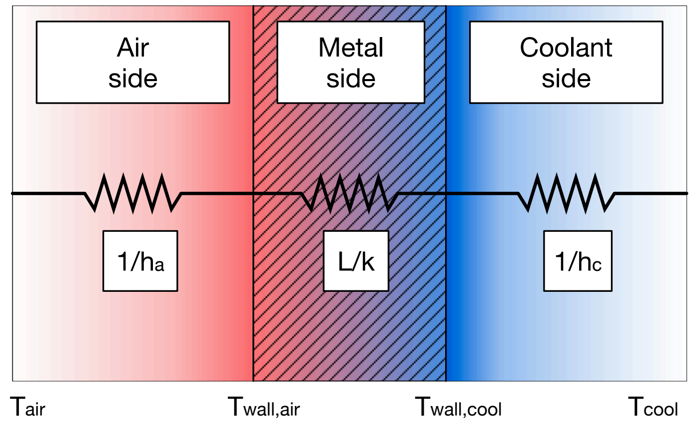


Figure 2.16: *Thermal circuit similarity*

And considering the total temperature difference:

$$T_{air} - T_{cool} = \left(\frac{1}{h_a} + \frac{L}{k} + \frac{1}{h_c} \right) \dot{q} \tag{2.3.10}$$

As a final result, remembering that $\dot{q} = \frac{\dot{Q}_C}{A}$ the total heat transferred is:

$$\dot{Q}_C = \frac{1}{\frac{1}{h_a} + \frac{L}{k} + \frac{1}{h_c}} A(T_{air} - T_{cool}) = K_r A(T_{air} - T_{cool}) \tag{2.3.11}$$

Where K_r is the global heat transfer coefficient of the heat exchanger and, since all the heat exchangers are more complex than a simple metal plate, its value is not theoretically calculated, but evaluated experimentally.

In the process of reducing the charge temperature there is an additional parameter to take into consideration: the charge air cooler effectiveness. Generally speaking CAC effectiveness could be defined as:

$$\varepsilon = \frac{Actual\ heat\ transfer}{Maximum\ possible\ heat\ transfer}$$

Actual heat transfer can be evaluated using the difference in enthalpies of one of the fluids, as the heat rejected is equal to the heat absorbed. The maximum possible heat transfer instead is the energy transferred when one of the fluids undergoes the maximum possible temperature change, for example, the hot charge leaves the CAC at the same entrance temperature of the cooling medium or the cooling medium leaves the heat exchanger at the same entrance temperature of the hot charge. Clearly this is not possible in reality and gives just an idea of the maximum cooling capacity of the CAC. Following the previous considerations and the so called ε-NTU method, the effectiveness can be evaluated as follow [23].

Find which is the maximum possible heat transfer; it will be the minimum of the two equations:

$$\dot{q}_{air} = (\dot{m}c_p)_{air}\,(T_{air,in} - T_{cool,in})$$

$$\dot{q}_{cool} = (\dot{m}c_p)_{cool}\,(T_{air,in} - T_{cool,in})$$

If the limiting factor is the hot gas side, the effectiveness can be written as:

$$\varepsilon = \frac{(\dot{m}c_p)_{air}\,(T_{air,in} - T_{air,out})}{(\dot{m}c_p)_{air}\,(T_{air,in} - T_{cool,in})} = \frac{T_{air,in} - T_{air,out}}{T_{air,in} - T_{cool,in}} \tag{2.3.12}$$

Same reasoning could be done if the limiting factor is the coolant side and the final result will be:

$$\varepsilon = \frac{T_{cool,out} - T_{cool,in}}{T_{air,in} - T_{cool,in}} \tag{2.3.13}$$

What analyzed so far is related to the thermal behavior of the CAC, which represent one of the most important parameters together with the pressure drop. In fact a good charge air cooler not only should reduce the temperature of the air as much as possible, but it must apply a very low resistance to the fluids in motion, otherwise it would spoil the effect obtained by the usage of the turbocharger. Looking at these constraints it is easy to understand that the charge air cooler represents a very important component, and its design must be analyzed carefully. The two main typologies of CAC present different thermal and pressure characteristics and will be analyzed in the following.

2.3.2 CAC typologies

Beside the working principle that is the same for both configurations, a distinction on the types of charge air cooler can be done considering the phase of the cooling fluid:

- *Air-to-Air CAC*, in which the cooling fluid is the external air
- *Air-to-Water CAC*, in which the cooling fluid is water.

Air-to-Air CAC

This is the simplest and most spread type of cooler. It is characterized by a very simple configuration and low cost. It doesn't need additional components other than the heat exchanger and the piping system. There are two possible configurations for the cooling module: full face and brick type. The former is usually placed in front of the radiator and before the A/C condenser, constituting the radiator pack. It is characterized by a width much smaller than the other two dimensions. The latter instead is characterized by a width comparable with one of the other sizes and since it has a more compact design it could be placed in the wheelhouse or in front of the cooling module, allowing more freedom in packaging.

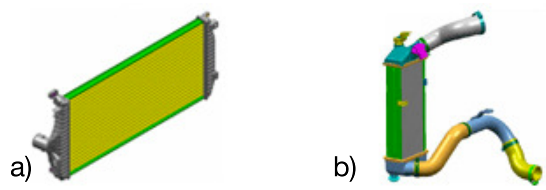


Figure 2.17: *Air-to-Air CAC types.* a) *Full face* b) *Brick type [24]*

The thermal performance of this solution is not as effective as the Air-to-Water one; since the cooling medium is a gas, with a lower specific heat capacity, a larger mass flow rate is needed and consequently bigger core dimensions are used to obtain acceptable performances. In this component the pressure loss can become a preponderant factor and needs to be eliminated. For this reasons the trend is moving towards Air-to-Water charge air coolers.

Air-to-Water CAC

These types of charge air cooler are becoming more and more popular among automotive manufacturers due to their increased performance potential. The cooling fluid, as the name suggests, is mainly water, with some small quantity of ethylene glycol to overcome possible freezing problems. The advantage of having water directly influences the cooling capacity, which is increased due to the higher specific heat capacity but also to the possibility of having higher mass flow rates. The solution is not free from drawbacks, which are mainly related to costs and additional components installation; in fact there is the need of a dedicated water circuit which means also an additional pump and heat exchanger for the coolant and for the air, increasing weight. The advantage of this configuration however is not only related to improved performance but also to the increased flexibility in packaging. Both the heat exchangers used in this configuration are smaller in dimensions and, especially the air one can be put right in between the turbocharger and the intake; in doing so ducting and piping are decreased with a consequent reduction in pressure losses, which are thus mainly linked to the pressure drop in the charge cooler itself [22].

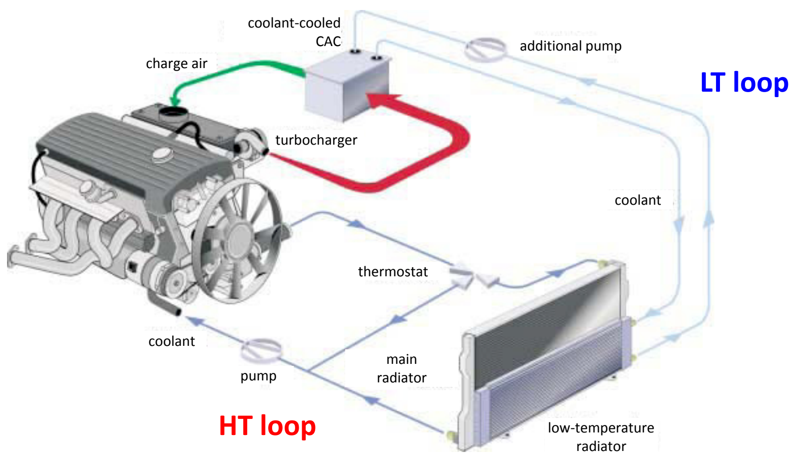


Figure 2.18: *Water charge air cooler with high and low temperature cooling circuits [25].*

The performances of this solution can be further enhanced if an integrated cooling module is implemented. It consists of putting the CAC directly on the engine intake so that the charge is cooled right before entering the combustion chamber, the pipes length is drastically reduced and the overall dimensions are decreased. This solution is the one implemented in the engine tested in this project. An example of possible integrated CAC is shown in Figure 2.19.

The benefits coming from the implementation of this system are related also to advance cooling strategies coming from the implementation of a high temperature radiator, used for cooling the engine, and a low temperature radiator, used for cooling the charge and also in the A/C circuit. At the occurrence a dual level high temperature radiator can be implemented, meaning that it has two sections that can operate independently of each other. Therefore, it is possible in partial load and low vehicle speed to operate one part of the high

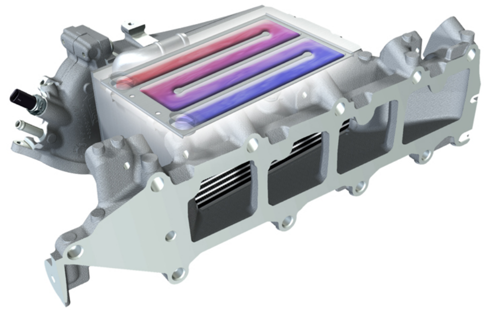


Figure 2.19: *Example of intake module with integrated charge air cooler [28].*

temperature radiator in low temperature mode, and the other part in high temperature mode, improving the A/C system operation. In this way the radiator pack dimensions can be reduced. An exhaustive description of this solution is beyond the scope of this project, but additional information can be found in Malvicino et al. [26] and Stehlig et al. [27].

For all these reasons and even though the complexity and cost of this technology are still high, there is a trend in implementing this solution on an increasing number of cars.

2.4 Exhaust Gas Recirculation - EGR

Charge cooling, and its synergy with turbocharging, has been shown as fundamental technique to reduce the possibility of knocking and increasing combustion stability and fuel economy. Another solution that has been studied in the last years, to further reduce fuel consumption and improve combustion, is exhaust gas recirculation.

2.4.1 EGR working principle and implementation

The concept behind EGR, as the name suggest, is to draw a part of the exhaust gases after the combustion and inject them back in the intake so that they could participate again in the combustion process giving rise to many advantages as knock reduction, fuel consumption reduction, pumping loss reduction, etc.

EGR was firstly implemented in diesel engines to perform an effective NOx emission reduction, but in the last years more and more research has been conducted on its implementation in gasoline engine, and it has been found as a very promising technique in reducing some drawbacks coming from high boosting pressure.

The reduction in NOx is one of the preponderant reasons why EGR is used. NOx gases are some of the most difficult pollutants to reduce and this system is able to achieve the target values. However, attention must be kept in increasing too much the EGR rate because of side effects: unburned hydrocarbon and carbon monoxide could be increased and combustion stability can be worsened impairing the positive effects.

The percentage of exhaust gas recirculation can be defined in different ways [29, 30]:

- Mass based

$$EGR\ rate\ (\%)_{mass} = \frac{m_{EGR}}{m_{fresh\ air} + m_{fuel} + m_{EGR}} \cdot 100 \tag{2.4.1}$$

- Volumetric based

$$EGR\ rate\ (\%)_{vol} = \frac{V_{EGR}}{V_{fresh\ air} + V_{fuel} + V_{EGR}} \cdot 100 \tag{2.4.2}$$

  or, considering the exhaust emissions, that is also the system used in commercial engines:

$$EGR\ rate\ (\%) = \frac{Volume\ fraction\ of\ CO_2\ in\ intake}{Volume\ fraction\ of\ CO_2\ in\ exhaust} \cdot 100 \tag{2.4.3}$$

For what concern the implementation, exhaust gases can be recirculated in two complete different ways:

- **Internal EGR:** if a suitable valve phasing is chosen to trap a part of burnt gases inside the combustion chamber.

- **External EGR:** if a dedicated piping system is used to draw exhaust gases after exiting the combustion chamber.

For what concern external EGR a further distinction can be done depending mainly whether if gases are drawn before or after the expansion in the turbocharger turbine. These two configurations are called:

- **High pressure EGR (HP):** if the exhaust gases are drawn before the expansion in the turbine and are sent back into the intake after the compressor.

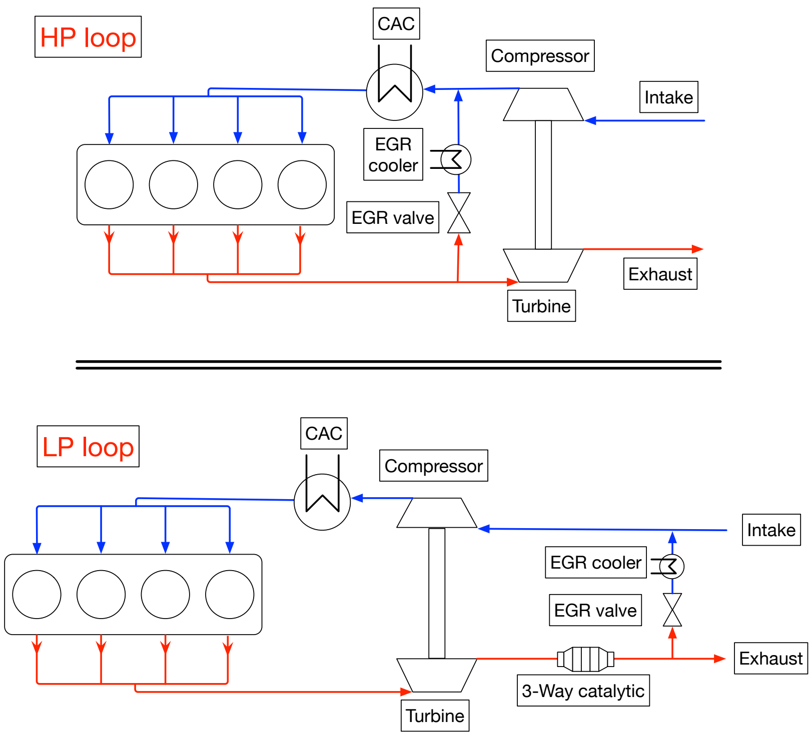


Figure 2.20: *EGR loops comparison. High pressure loop on top, Low pressure loop on bottom*

- **Low pressure EGR (LP):** if the exhaust gases have undergone the expansion in the turbine and the pickup point is generally placed after the catalytic converter.

The two circuits presented in Figure 2.20 are the very basic configurations, but it is quite common to find both of them mounted together on vehicles. The reason is related to the different behavior of the two systems at different working conditions. It is important to observe that, to properly work, EGR circuit needs to have a sufficient amount of EGR flow rate and a sufficient amount of differential pressure between the exhaust and intake must exist. For example the HP circuit is better suited for low loads operating range since a high boosting is not required and thus the exhaust pressure is higher than the intake pressure. On the contrary at high loads, the boost level should be increased leading a differential pressure between exhaust and intake not suitable to drive the HP loop, and so in this case the LP loop is used. To overcome this pressure gradient problem an effective solution is the usage of a variable geometry turbocharger to build sufficient backpressure.

Observing Figure 2.20, it is noticed that in both circuits the EGR flow is cooled. The cooler is similar to the charge air cooler, and sometimes it utilizes the same cooling fluid, while other times, for the same purpose is used the engine coolant. The important fact is that cooling the EGR flow will enhance the EGR positive effects, and all along this project only cooled EGR will be considered.

2.4.2 EGR Benefits

EGR has been proven to be one of the most effective techniques for reducing NOx emissions but it has also shown a very good behavior in improving combustion and reducing fuel consumption in gasoline engines. In the following a brief summary of the main advantages coming from the implementation of cooled EGR will be given.

- **NOx reduction:** as extensively stated, one of the main effect of EGR implementation is its ability of reducing NOx in a great extent, this is due to the reduction in both combustion temperature and oxygen concentration. It is well known that the favorable conditions for NOx formation are the presence of high oxygen concentration at high pressure and temperature, and thus turbocharged engines could produce very high levels of these pollutants. The effect of EGR has been found to operate mainly in two ways:
  - *Thermal effect:* the average specific heat (c_p) of exhaust gases, since they consist mostly in CO_2 and H_2O, is higher than the one of the intake air and thus can absorb more heat in the combustion chamber lowering furthermore the temperature.
  - *Dilution effect:* For a given fuel quantity, a fixed amount of oxygen is required for complete combustion no matter what the EGR level is. Since the oxygen fraction is lower with EGR, it will be more difficult for the flame to encounter fresh air and non-oxygen molecules absorbs heat lowering the temperature.
- **Reduced heat rejection:** the reduction in combustion temperature is not only beneficial for NOx reduction, but it also reduces the loss of thermal energy to combustion chamber surfaces, leaving more available for conversion to mechanical work during the expansion stroke, improving consequently the thermal efficiency [32, 33].

- **Reduced throttling losses:** this is one of the fundamental benefits, in fact to maintain constant the power output of an engine adding inert gas (EGR) the throttle valve must be opened more, resulting in an increased inlet manifold pressure which will consequently reduce the lower area in Figure 2.21

- **Advancing combustion phasing:** Without the usage of EGR the ignition time is set depending on knock, closer to the TDC or even after it, while with the usage of EGR the combustion phasing can be improved, ignition time can be advanced in respect to the TDC allowing thus more time for the combustion to be completed, in fact also the CA50 is improved [31]. The peak pressure is then moved to the TDC increasing the combustion efficiency [32]. The autoignition tendency is reduced too.

- **Reduced need of enrichment:** one of the best strategies to reduce knock was to inject more fuel, which when evaporates it subtracts heat to the combustion chamber, but the usage of EGR is eliminating the need of enrichment because of its effect in lowering the temperature, thus fuel economy is improved as well.

All these benefits taken as a whole will sensibly reduce engine knock tendency, fuel consumption and NOx, easily justifying the great interest behind the research on the EGR system.

2.4.3 EGR Drawbacks

Together with the benefits explained above there are some disadvantages that have to be considered when implementing EGR systems

- **Combustion worsening:** adding EGR beyond a certain limit can worsen the combustion process, increasing in time and sometimes there could be the possibility of misfire [31, 33]. This could lead, in certain conditions, in an increase in some of the emissions as HC and CO [34].

- **Components fouling:** recirculated gases are not clean as the fresh air entering the intake, but they present particles that are the residuals of the combustion process. These particles can be trapped especially in the EGR coolers giving rise to additional pressure losses. Furthermore, for certain EGR circuit configurations, they could impact with the blades of the compressor giving rise to mechanical damages and possible failures.

- **Additional components and control strategies:** to better exploit the beneficial effects of the EGR, additional components as coolers or EGR valve should be introduced, giving rise to an increase in complexity and cost as well as more stringent packaging constraints. The different behavior of the LP and HP circuit at different speed and load conditions need also a dedicated control scheme that must be carefully designed and implemented.

Looking at the drawbacks highlighted above it is clear that the implementation of the system, despite the increased costs, represent one of the best available solution to respect the more stringent emission regulations.

2.5 Results from other authors

In the literature many authors have analyzed the beneficial effects of the components described so far. They all showed that great improvements in terms of increased output power, fuel consumption reduction and emissions reduction can be achieved. Analyzing their results, however, is impossible to describe which is the effect of each solution (turbocharger, EGR, etc.) because all of them were considered together as a unique system; so, i.e., in the investigation of the effects of cooled EGR also the turbocharger will be considered. With this consideration is thus impossible to separate and present the advantages of each single component.

Despite these considerations, most of the papers analyzed show common trends. Most of the times turbocharged gasoline engine with direct injection system and cooled EGR (both high and low pressure) are considered. It is important to state that different authors have analyzed different engine in different speed/load conditions. In doing so the results are not very aligned because strongly dependent on the condition considered, this means also that certain results could show dissimilar and even opposite trends.

In the following the main results are presented, only considering the common findings; for further information, the papers used for this project can be retrieved from the bibliography.

- **Combustion phasing:** the CA50 value can be advanced up to a certain extent, increasing thus the combustion time and consequently leading to a complete fuel burn [31, 32, 35, 36], Francqueville at al. [18] stated that CA50 can be reduced by 1° crank angle every 4-5% EGR.

- **Reduction of exhaust temperature:** as a general effect the addition of EGR in these type of engines lead to a reduction in exhaust gas temperature with the consequent improvement of thermal efficiency. Potteau at Al. [31] show a reduction of up to 100°C; a same result is obtained by Galloni et al. [37].

- **BSFC reduction:** the possible reductions discovered range between 4% to 14% [18, 31, 35, 37]. However these data are very sensible to engine speed and load.

- **Knock suppression:** many authors have described the effect of EGR in reducing knock [32]; the consequences of the implementation of this solution are positively affecting also fuel consumption because there is no more need of fuel enrichment ($\lambda = 1$ even at high loads) [31, 18, 38].

- **Pumping work reduction:** the usage of turbocharger in addition of EGR can help in drastically reduce pumping work. This positive effect has been also recognized by Wei et al. [32] and Su et al. [33]. Also other authors however have claimed the benefits obtained in pumping losses reduction.

- **Emissions reduction:** all authors agree on the fact that NOx are reduced in a great extent, Lattimore and Al. [38] stated a reduction of this pollutant up to 43%. There is also a certain reduction in the CO production, while for the HC there are some contrasting results: Alger et al. [35] showed an increase in HC production due to the reduced combustion stability at high EGR rate, while Potteau et al. [31] instead found an HC reduction with the implementation of EGR. However these results are subjective to the type of engine used and load considered and furthermore, as far as the engine is operated in stoichiometric condition the three-way catalytic is converting

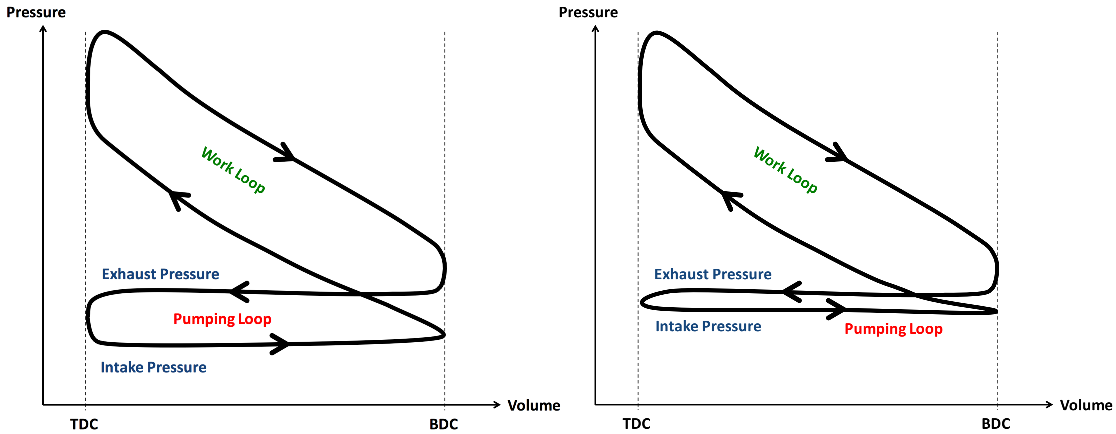


Figure 2.21: *Pumping Loss reduction due to increased intake pressure*

HC and CO with a great efficiency, so the main benefit to look for is the reduction in NOx which is actually found in all the papers.

Chapter 3

Methodology

In this part the methodology followed during the development of the project will be described. Two different procedures have been applied to obtain two different outcomes: the first one is related to tests performed on an engine dynamometer; the second one, instead, is performed using a 1D simulation software. The results obtained, however, are not independent one of the other, but are used in a complimentary manner to reach the final goal proposed.

The final objective, in fact, is to find the optimal configuration for certain actuators in the cooling system, minimizing the additional power required to perform the task and the fuel consumption as well. Components considered in the optimization are for example the fan (controlling its speed), the thermostat and others. The complete list of all components however cannot be written since the procedure is a company trade secret.

Before explaining the procedure followed during the project, a brief schematic of the engine configuration is presented. The EGR circuits, as well as the EGR coolers and CAC are showed in Figure 3.1.

The blue circuit represents the intake part, while the red one is the exhaust side. It is possible to notice the presence of two EGR loops: high pressure and low pressure. Both circuits shows also that EGR coolers are used to cool down the gas before entering in

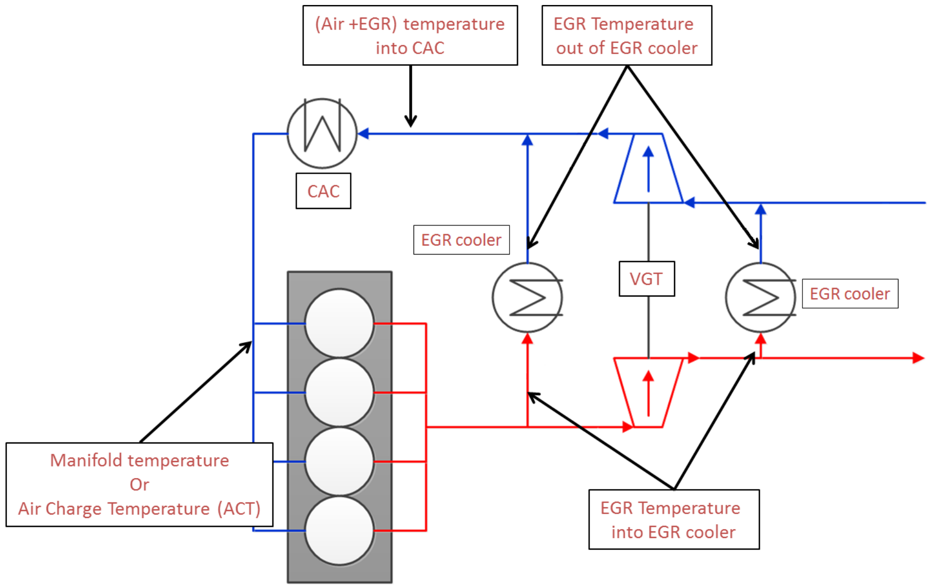


Figure 3.1: *EGR circuits and heat exchangers schematic*

the intake flow. The main temperature monitored during the dyno test are highlighted in the same figure. It is important to remark that Manifold temperature and Air Charge Temperature (ACT) are formerly the same parameters. The only conceptual difference is that ACT can be seen as the desired temperature at the outlet of the CAC, while manifold temperature is the measured one. However, except this difference on the names, which is only related to technical language, the values of the two measured temperatures are the same.

3.1 Engine dynamometer tests

The dyno configuration used for testing the engine is the same one applied also for emission testing cycle. The engine, a 4 cylinder turbocharged spark ignition engine, is mounted on an electric motor/generator dynamometer and the test were run, for every speed/load, at a constant power output. Moreover the working point chosen for the project are all steady state conditions at constant engine speed. The selection of these is based on analysis made over the FTP cycle. The points chosen are all placed at 1500 rpm at different loads:

- 1500 rpm , $BMEP - 2\ bar$

- 1500 rpm , $BMEP - 4\ bar$
- 1500 rpm , $BMEP - 6.5\ bar$
- 1500 rpm , $BMEP - 10\ bar$
- 1500 rpm , $BMEP - 15\ bar$

In addition to these points, a high load condition is added:

- 1997 rpm , $BMEP - 17.34\ bar$

While the tests at 1500 rpm were all run at an ambient temperature of 27°C, the last point instead was performed at an ambient temperature of 36°C, and represents a very severe working condition.

The engine mounted on the dyno does not have the cooling system that is commonly equipped in vehicles. Instead, it uses the test cell's cooling system for the coolant temperature control. In most of research studies the engine is tested without the additional components constituting the cooling module, in doing so, their effect is not accounted when testing the engine.

The cooling system for the CAC is substituted by a separate cooling system of the test cell. Thanks to this solution the desired temperature of engine fluids (engine coolant, engine oil, etc.) is set and it is kept constant for each working condition. The advantage is that variations in the fluids temperature during the engine analysis are not influencing the final results, but from a power and fuel calculation point of view, the results, once the engine is mounted on the complete vehicle, are impaired.

Many sensors are mounted on the engine in order to control a wide variety of parameters, the output of the sensors is then read and stored in a computer from which it is possible also to operate on some of them.

In the tests performed during this project, as already mentioned, the engine speed was kept at a constant value, as well as the power output. Once the engine is fully warmed up and it has reached a steady state operating condition the measurements can be performed. Since the working points (speed/load pair) are considered as steady state, the outputs from the different sensors are averaged over 500 cycles.

The main focus of this project is to highlight the effects of EGR rate and Air Charge Temperature (ACT) on the fuel consumption and for the purpose an ACT and EGR sweep was done; the procedure is briefly described in the following:

1. **Set air charge temperature:** the air charge temperature is the temperature measured at the outlet of the charge air cooler; it directly influences combustion temperature. It is the first parameter to be set since it needs a longer time to move from one value to another due to the fact that the external cooling system needs to adjust the flow rate and temperature to assure the desired ACT. The value chosen for the temperature sweep are: 30 to 90°C with an increment of 10°C. Once the engine speed and desired load are achieved, and the manifold temperature has reached a steady state condition, the test could start and the other parameter to adjust will be:

2. **EGR rate:** the EGR rate is calculated using the Equation 2.4.3, the exhaust flow is analyzed and the EGR valve is opened to achieve the desired EGR amount. The response of this parameter is faster than that of the ACT and the effects of the opening and closing of the EGR valve are almost instantaneous. For these considerations the EGR sweep was made at a constant ACT. The value chosen for the EGR, especially for the maximum one, greatly depend on the manifold temperature and load considered, as will be shown in the discussion of the results. The lower values, however, are varied from 0% EGR (valve completely closed) with an increment of 5% (i.e. 0% - 5% - 10% - 15% and so on until COV of IMEP reaches 3%).

   A consideration has to be made on the minimum value: it never happened to have 0% EGR even if the EGR valve was completely closed due to the presence of internal EGR coming out into the intake manifold during the intake valve opening.

   The next step will describe how the maximum value of EGR rate was obtained.

3. **Set the maximum EGR rate:** during the tests, as explained above, the EGR rate is varied until the maximum allowable limit. This limit can be reached in different conditions:

   - **Coefficient of Variation (COV) of Indicated Mean Effective Pressure**

(IMEP): as commonly done in literature and in other tests, the COV of IMEP is defined as:

$$COV\ of\ IMEP = \frac{\sigma(IMEP)}{\mu(IMEP)} \tag{3.1.1}$$

where σ is the standard deviation and μ the mean value calculated over a consecutive number of tests.

The COV of IMEP signifies the cyclic variation of the IMEP and thus the stability of the combustion process. It helps in determining if it can be considered stable or not. Further more high variations can produce a degradation in vehicle drivability directly perceived by the diver.

During the tests, when the COV of IMEP reaches the maximum set limit (3%), the maximum EGR rate for that point is reached and the test is ended.

- **Insufficient exhaust back pressure:** it could happen that, for certain conditions, the maximum EGR rate allowable is limited by the amount of backpressure (pressure difference between exhaust and intake) instead of the COV of IMEP. In this case the test is ended while the combustion stability is still assured. To overcome this problem of limited backpressure a different EGR loop configuration or other technique to increase the pressure in the exhaust should be applied.
- **Knock limitations:** at high load and temperature it is possible to encounter limitations on the maximum EGR rate due to the presence of knock. These working points should be avoided in order not to damage the engine. Solutions to this particular limitation can be implemented reducing the manifold temperature or enriching the mixture; however the scope of the project is to analyze also the effects of high manifold temperature at high loads keeping the air fuel ratio constant.

4. **Repeat the procedure for higher ACT:** once the maximum EGR limit is reached for a particular manifold temperature the test is ended, all the results are stored and the air charge temperature is increased to the next point: then the procedure described above is repeated until a complete ACT and EGR sweep is performed.

In addition to the procedure described so far, it is important to state that some other parameters have to be check all along the testing phase, in order to obtain the optimal combustion process. The following parameters have to be always checked:

- **CA50:** it represents the crank angle degree at which the 50% of total heat release due to the combustion occur and it directly influences the maximum brake torque obtainable by the engine. A common value set for the CA50 in normal testing conditions is around 7 to 10 degree after the Top Dead Center (TDC), while for very severe conditions this value can be retarded more to avoid the problem of knocking.

  The CA50 is directly controlled by the spark timing; in fact, advancing the ignition event (typically before the TDC) the CA50 is moved closer to the TDC and so, closer to the desired position. From this consideration it is easy to understand why in some severe conditions (at high loads) it is difficult to stay in the prescribed CA50 range. When knocking could appear, the spark event should be retarded to move the peak pressure far from the TDC. Having high EGR percentages it is possible to further advance the spark angle, with a positive effect on combustion phasing.

- **No significant variation in engine parameters:** the tests are all run in steady state conditions and for this reason, before taking the data for each EGR rate there is the need to wait a certain amount time in order to let the engine stabilize. When something is varied, for example the EGR valve opening position to admit more EGR flow, a transient behavior is starting to appear; in order to have accurate results no significant variation should be noticed in parameters, such as EGR rate, COV of IMEP, and CA50. When everything is stabilized then it is possible to start collecting the data for the 500 cycles and then average them to obtain one value for each parameter evaluated.

- **Knock presence:** at higher loads knock, which is detrimental to the engine, can occur and could impair the tests or even worse, damage the engine. Bearing in mind these possible problems, thus, the in-cylinder pressure is always monitored in order to promptly act against the onset of knock by manually adjusting properly the spark timing (retarding it when some spikes start appearing on the trace). This problem was found in particular during the severe test condition.

One additional step to do for running the tests at high load and ambient temperature is to increase the latter prior the beginning of the test and let it stabilize around the desired value.

Once all the tests have been run, among the parameters collected, one of the most important results is the Brake Specific Fuel Consumption (BSFC). It allows the comparison of different working condition since the quantity of fuel burn is divided by engine power output, obtaining so a result in g/kWh. Since the tests were ran at constant torque and rpm the power output was constant, and with a simple multiplication it is possible to calculate also the fuel consumed measured in g/s as follow:

$$FC\ [g/s] = BSFC \cdot \frac{Torque \cdot rpm \cdot 2\pi}{1000 \cdot 60 \cdot 3600} \tag{3.1.2}$$

where *Torque* is measured in Nm.

The calculation of the fuel consumption in grams per second will be useful in the second part of the project when it will be compared with the results from the simulation on the cooling system. From the engine dyno tests performed a 4-dimension map is built in order to retrieve, from the 3 inputs: EGR rate, ACT and load, the corresponding BSFC value. In doing so the effect of the manifold temperature and of the EGR rate is analyzed and the configuration which leads to lowest fuel consumption can be found. However it is important to notice that, for intermediated load values (i.e., 5 bar) the BSFC is obtained using an interpolation technique, thus the results could be less accurate than a complete analysis at the specified load.

As it is explained in the procedure above, the tests represent a critical part of the project development and it is essential to pay close attention to have good results to be used in the second and final part.

3.2 Cooling system simulation

Recalling that the scope of the project is to analyze the influence of the EGR rate and air charge temperature on the fuel consumption, there is the need to consider also the power required to cool both the EGR cooler and the CAC. During the dyno tests this fact was not a big concern since the cooling cart can assure "unlimited" cooling capacity and it is not subtracting power to the engine. Analyzing instead a real engine, mounted on a vehicle, to obtain the same temperatures there is the need to use the cooling system, in doing so additional power is required and it is drawn directly from the it.

Considering a common cooling system used on the engine there are additional components that were not mounted on the dyno cell, these include the cooling fan, the air conditioning system, the frontal radiator, etc. These components are mounted on the engine mainly through belts connections and they are directly drawing the power from the engine. Bearing in mind this last statement it is clear that, to obtain the same power available at the wheels, the engine, with all the additional components mounted, should produce a higher power. A simple formula to have clearer in mind the concept could be the following one:

$$Power\ output_{total} = Power_{wheels} + Power_{cooling\ system} \tag{3.2.1}$$

So, for example, if the power requested to travel is 90 kW and the cooling system is requiring 2 kW to cool down the engine, the total power request to the engine will be 92 kW. The calculation done is very rough and it gives only an explanation of what there is behind the project. For more reasonable results, the connection efficiency and power losses in the whole process need to be considered, however the main idea is that this additional power is translated in additional fuel injected and so higher fuel consumption.

From the tests described in the previous section the best manifold temperature and EGR rate to minimize the consumption were found, but once the cooling system is added as a variable however the lowest fuel consumption point could be different.

Generally speaking, from a point of view of BSFC, higher load conditions result in lower fuel consumption, but in the case considered, the additional load is "wasting" a part of the fuel only to cool the engine. It is important to say that the fuel is not really wasted, since cooling is fundamental for the proper functioning, however the goal is to minimize the power request for this purpose. There is a common belief, from the point of view of combustion, that the colder the air/fuel mixture the better the process and fuel consumption. From a point of view of cooling power, instead, having colder and colder fluids (EGR or air) is translated in higher power request.

After all of these considerations the final scope is to analyze then if the possible improvement in fuel economy coming from a colder mixture entering in the combustion chamber are impaired by the additional power required by the cooling system, or if it is better to burn an hotter charge reducing the cooling effect.

Just to fix the ideas, for example, there could be a working point in which the lowest fuel consumption is obtained with a high EGR rate and low manifold temperature, but the cooling system will require the engine to burn 15% more additional fuel to cool down that high flow rate at the desired low temperature. For the same working point, a higher manifold temperature could be accepted; the effect of the higher temperature on fuel consumption could result in an increase of 2% for example, but at the same time the cooling system is requiring only 5% more fuel to be burn. For this particular working point is then found that is better to have an hotter mixture, but reducing the load on the engine. This reasoning however is not always straightforward and cannot be applied as a general rule; in fact, each point has to be analyzed carefully.

The final outcome will be thus the evaluation of whether if it is better to increase the cooling capacity ti have a lower fuel consumption or increasing the manifold temperature to have, consequently a lower power drawn from the engine. Most of the times a trade off between these two objectives must be found.

3.2.1 Simulation procedure

The procedure followed in this second part is schematically described in the following, while some points will be discussed more in detail.

1. Modifications on 1D cooling system model
2. Creation of a Design of Experiment (DOE) matrix
3. Final setup of the simulation

4. Collection and analysis of the results

This procedure is then followed for each working point and will finally give the results required.

Modifications on 1D cooling system model

To simulate the influence of the cooling system on the engine and on the fuel consumption, Flowmaster, a 1D Computational Fluid Dynamics (CFD) software able to simulate also the thermal behavior has been used. The analysis carried on during the simulations is steady state, for certain particular conditions it could be a drawback, but in the case of this project the main focus is on driving conditions where are not present steep velocity gradients or acceleration, moreover the tests on the dyno are done at constant load and speed conditions.

The model used for the simulations is based on the real components of cooling system used on the vehicle, and was already created for previous studies. It represents only the components present in the cooling system itself. Every component is modeled based on real dimensions and real data for heat transmission. These data are obtained from suppliers or previous studies. The combustion is not modeled in every simulation, but what is important is the combustion heat rejection to the coolant for every rpm and load. It is, though, obtained through look-up tables based on previous engine dyno tests. The same procedure was applied for certain parameters such as the EGR heat addition to the coolant. However, after the collection of data from the dyno tests, the model has been changed. An EGR cooler is implemented using a heat exchanger with the same dimensions and cooling capacities of the real one mounted on the vehicle. The flow is modeled, as well, taking into account the actual amount of mass flow rate $[kg/s]$ and the actual temperature measured on the previous tests. The same procedure is done also for the CAC.

Many parameters of different components, mainly pressures, temperature and flow rates are collected as results from these simulations and are used for further calculations in the next steps.

Creation of a Design of Experiment (DOE) matrix

The creation of a strong and reliable design of experiments plan it is very important to better analyze all the possible interactions between the different components of the cooling system. For each component furthermore there are many different possible positions. For example, considering the cooling fan, its speed could be varied from 0 rpm (completely shut off) to its maximum speed, and it could also assume intermediate values, for example with an increment of 500 rpm. Only considering the fan it is clear how many simulations have to be made to tests all the possible speeds. Extending the reasoning for more components, the process is incremented to reach an extreme number of possible interactions. For this reason a good predefined simulation plan must be implemented.

To have a good understanding of what is written in the next page, it could be useful to define what parameter/factor and levels mean:

- A **factor** or **parameter** is one of the element studied in the process. In this case, for example, the cooling fan, together with other cooling system components, is the parameter analyzed

- The **levels** represent the number of a parameter possible different values. In the case of computer bits which could only be 0 and 1, the levels are just 2, while, for example, the cooling fan rotational speed can vary from 0 to its maximum speed. In this case the values are not discrete, but vary in a continuous way, so the easiest thing to do is to pick only some discrete values. In this way, imaging that the speed is ranging from 0 to 2000 rpm and an increment of 200 rpm is chosen, the levels would be 11.

The idea behind a good procedure plan is trying to highlight, among all the possible parameters, which is the effect of one on the others together with the magnitude of this interaction. The design of experiment is thus a technique used more and more in companies to face this problem when dealing with a great amount of factors.

One possible way of finding interactions is to vary one parameter at time, keeping all the others constant *(One Factor at Time – OFAT)*. This approach could be good when dealing with few elements with two or three levels. In OFAT, the first factor is fixed as a "good" value, the next factor is examined, and on and on to the last factor. Because each experimental run considers only one factor, many runs are needed to get sufficient information about the set of conditions contributing to the problem. Another limitation is that when factors change, they generally change together, so it is impossible to understand the best solution by pointing to a single, isolated factor.

DOE, on the other side, provides information about the interaction of factors and the way the total system works, something not obtainable through testing one factor at a time while holding other factors constant. Another advantage of DOE is that it shows how interconnected factors respond over a wide range of values, without requiring the testing of all possible values directly.

When talking about DOE there are mainly two different ways of building the plan map:

- Full factorial plan
- Partial factorial plan

The **full factorial plan** consists in analyzing all the interactions between the different parameters, so every factor at every level is considered. It is certainly the most complete test that is possible to plan since all the interaction are analyzed, but it will require longer time; it is feasible only when considering few parameters with few levels. The reason of that statement is easily explained when looking at the following formula to calculate the amount of interaction needed to investigate all the levels:

$$N^\circ \ interactions = \prod (l_f)^{n_f} \tag{3.2.2}$$

Where l_f is the number of levels of factor f and n_f is the number of factors which show a number of levels l_f. For example, having $A = [1,2,3]$, $B = [4,5,6]$ and $C = [7,8]$ the number of iterations required is:

$$N^\circ \ interactions = 2 \cdot 3^2 = 18 \tag{3.2.3}$$

It is now clear how much the computation time is increased when dealing with many factors at many levels. So this procedure is quite time consuming and it was found to be not feasible in the development of the project and a partial factorial plan is used.

A **partial factorial plan** experimental design consists of a carefully chosen subset (fraction) of the experimental runs of a full factorial design. In this case some levels are not taken into considerations and so some interactions are not evaluated, however this technique leads to good results when the amount of time available is not sufficient to perform a complete design. There are many different techniques when dealing with this type of design, but the main concept behind all of them is the fact that population should be as widespread as possible. In this project an Optimal Latin Hypercube strategy is used to build the design matrix, and a good spread of the factor is assured.

Two more important reasons, related to the cooling system itself, should be mentioned on the choice of using an optimal latin hypercube rather than a full factorial plan:

- The number of factors considered to build the DOE is rather high due to the high number of components over which is possible to act to obtain different cooling system configurations. Considering that there are 6 components (factors) that are varied within 5 to 10 possible positions (levels) it is easy to understand that, in the best case possible (all 6 factors with 5 levels), the required number of iteration would have been $5^6 = 15625$.

- If the above concern is not enough to promote the usage of a reduced plan, it must be noticed that in real conditions some actuators positions are not allowed because they would cause the engine not to run at the proper temperature or they would cause the cooling system malfunctioning. Considering that, using a partial plan will reduce in a great extent the possibility of picking these points together with the required computational time.

For the purpose of the project, Isight, a software which, after setting up all the parameters and their levels, automatically generates the design matrix has been used, and for each working point 600 runs have been performed, assuring a good distribution of all possible actuators positions.

Final setup of the simulation

After implementing the 1D system model and the design of experiment matrix, to finally perform the simulation some additional information are still needed: in particular, for what concern the vehicle speed. This parameter is important since it will be needed in the 1D model to calculate the amount of air flowing through the radiator pack, and different speeds would result in different cooling capabilities.

For what concern the engine, it is not important if the vehicle is traveling at a certain speed rather than another. From the moment it could assure a determined torque and angular speed the test on the dyno are performed properly. But when considering the cooling system and its integration on the engine, the speed is becoming a parameter rather important. At fixed torque and rotational speed, the vehicle speed can be different depending for example on the gear inserted, the slope of the street and the aerodynamic resistance. The vehicle speed, in the case of this project, has been extrapolated from the FTP cycles, adding additional variables to the already complex matrix. In fact for each working condition 2 different speeds were considered: one for the city cycle and the other for the highway; exceptions were made for the 10 and 15 bar cases where the speeds were found to be the same in both conditions. As it is shown in Figure 3.2 during the tests the speed behavior is very transient, and it is not easy to determine a proper steady state speed. To achieve these results, however, a weighted averaging technique has been applied.

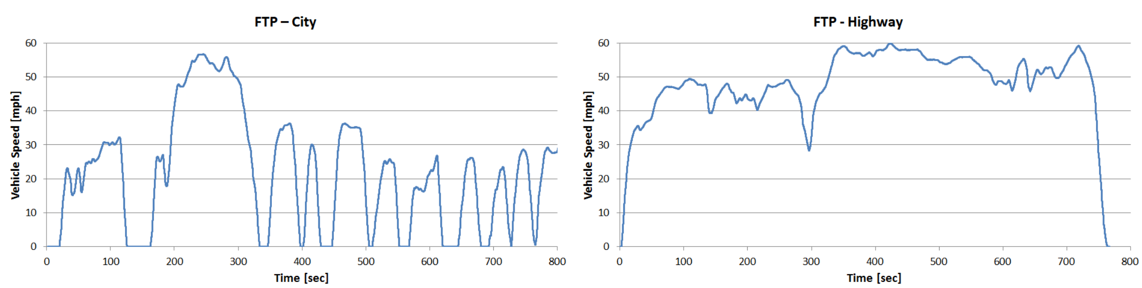


Figure 3.2: *Vehicle speed characteristics during FTP cycles: City on the left, Highway on the right*

The weighting factor considered is the percentage of usage of each gear in each cycle. It is clear that, in city driving, lower gears would be used most, resulting in lower speed, while, in highway, most of the times the last gear is engaged. With this operation, and considering a sufficient large range of engine rotational speed near the loads required, an almost constant speed has been found. These speeds tend to stabilize to a certain value as the ranges are increased.

After calculating the speeds for every working point the 1D software is setup in such a way to communicate with the DOE one. Information are then flowing both as inputs and output from one software to the other and the final results are stored for further analysis.

3.2.2 Collection and analysis of the results

After all the runs in the DOE are performed the final results can be retrieved and processed for the final analysis.

The results from the simulations are firstly processed to sort out the conditions which could lead to an improper use of the cooling system. Some constraints are applied to filter the results, so, for example, the engine coolant temperature, as well as the engine oil temperature and other parameters must be in certain ranges. It is fundamental to highlight that these constraints are set depending on engine durability tests and are chosen to assure a sufficient engine life. This assumption is very important and, together with the fact that during dyno test the combustion stability was always maintained, they assure a possible satisfaction on the customer point of view. The engine in fact will be durable and fuel efficient.

After all the data have been sorted and filtered, interactions between the different components of the cooling system and the fuel consumption can be discovered, finally the development of a strategy to increase the fuel economy for every working condition could be found.

The main outcomes to look at are the EGR rate, the ACT and their influence on fuel consumption at different loads. From these three inputs, using Matlab, a surface $BSFC = f(EGR, ACT, load)$ has been built, allowing the calculation of the brake specific fuel consumption for loads which were not directly investigated.

The major drawback of this technique is that, if some points are placed too far, a linear interpolation is sometimes too rough to give accurate results. For some other points (points at the extremes of the ranges) there could be also the need of extrapolating their value, reducing even more the accuracy. However, the points considered in the tests were almost evenly distributed and the results considering the objective of the project could be considered very satisfactory.

Before moving to the analysis of the results it is worth to stress again the importance of the two separate studies: the dyno tests and the 1D simulation. Considering only one effect at the time you could focus your attention on two opposite sides:

- **Focus only on the engine:** In this case the greatest result possible is the one at the lowest fuel consumption point, no matter which is the manifold temperature or the EGR rate. The cooling system is not taken into consideration and its thermal duty and power request are not entering in the calculation. The best point is thought to be the one at the highest EGR rate at the lowest temperature possible, clearly this is an extreme condition the cooling system has to whistand.

- **Focus only on the cooling system:** This is the opposite condition. The ideal point would be at 0 power request from the cooling system, but this is not clearly acceptable, since it would lead to combustion and materials problems. To reduce the power request to cool, the temperature should be risen, but a too big increase could produce the opposite effect.

The solution to the problem is to look at the engine and cooling system as a whole. In doing so, a tradeoff between the two points of view, and the effects of one component should not hinder the effects of the other.

Choosing whether if it is better to cool more or if it is better to let the engine run with no additional loads on it, would not probably be straightforward and the advantages and drawback of both concepts must be analyzed properly.

An extensive explanation of the results found will be provided in the following.

Chapter 4

Results and Discussion

In this chapter the main results obtained during the project are presented and discussed. As with the dyno-simulation methodology, the results are accordingly presented in two different sections.

First, the data obtained from the dyno tests is presented. Combustion stability, gas temperatures and fuel consumption are discussed in terms of EGR rate and ACT.

Secondly, after considering the combustion process, the results from the simulations on the cooling system will be provided. The best actuators configurations for the different loads are located and their effect on the fuel consumption are evaluated.

4.1 Engine dyno tests results

The output data collected during the engine tests are presented divided in the different load conditions.

It is important to remember that the tests were all conducted at 1500 rpm and an ambient temperature of 27°C, except the severe condition at 1997 rpm and 36°C. For each load, the air charge temperature is set to the required one and EGR sweeps are made.

4.1.1 Results at BMEP = 2 bar

Effects of EGR on COV of IMEP

This condition does not represent a stressful test for the engine, in fact all tests were run without problems regarding knock and the maximum admissible EGR rate is found, for each temperature, when the COV of IMEP was higher than 3, as shown in Figure 4.1

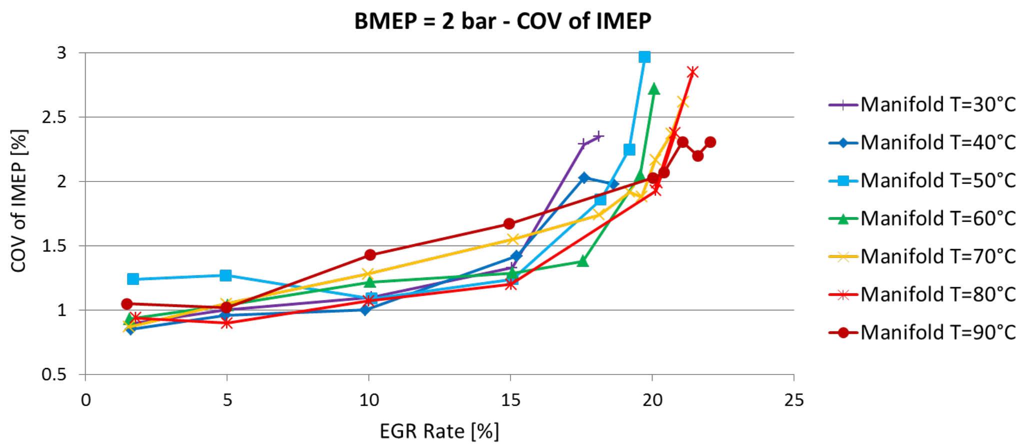


Figure 4.1: *COV of IMEP at different EGR rate and ACT - BMEP = 2 bar*

It is evident the effects of high EGR rates on the instability of the combustion process since the amount of fresh mixture available to be burnt is reduced in a great extent. Another consideration could be done observing the lines representing the different manifold temperatures, they are overlapped, meaning that, changing the temperature has not a major influence on the COV of IMEP.

It is important to note that the sensors are measuring always a certain amount of EGR even if the EGR valve is completely closed (there are no data at 0% EGR). This is due to the fact that a certain amount of exhaust gases is flowing back into the intake due to pressure difference.

Effects on gas temperature

For the low load condition considered the exhaust gases are leaving the combustion chamber at a fairly low temperature, as shown in Figure 4.2. It is observed, that the different manifold temperatures are not influencing the exhaust temperature, which is only related to the different load condition considered.

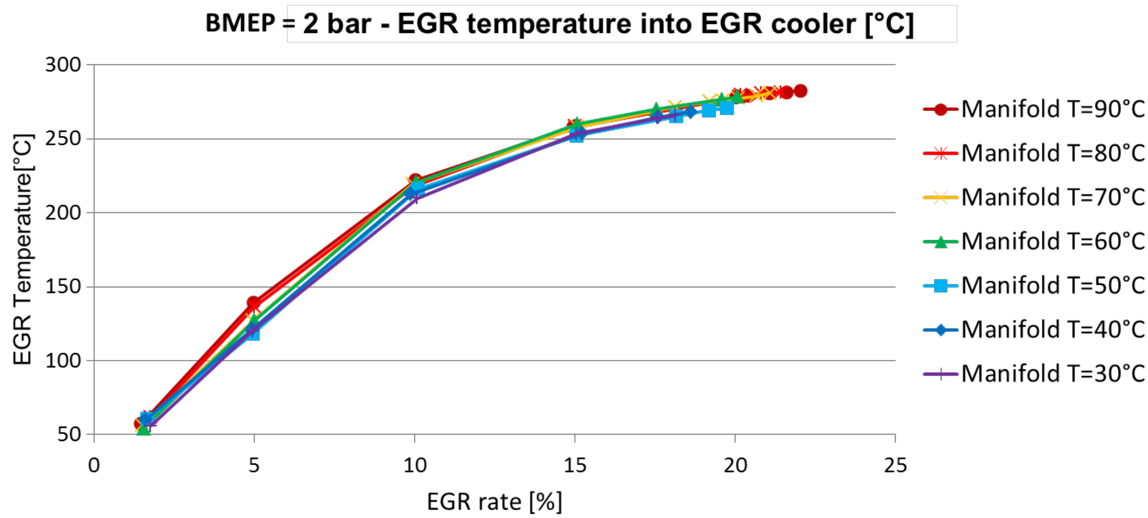


Figure 4.2: *Exhaust gas temperature entering the EGR cooler - BMEP = 2 bar*

The EGR was cooled in the EGR cooler, which is connected, as well as the CAC, to the external cooler of the test cell. In doing so, the temperature of the exhaust gas out of the EGR cooler is greatly reduced; the final outlet temperature if shown in Figure 4.3.

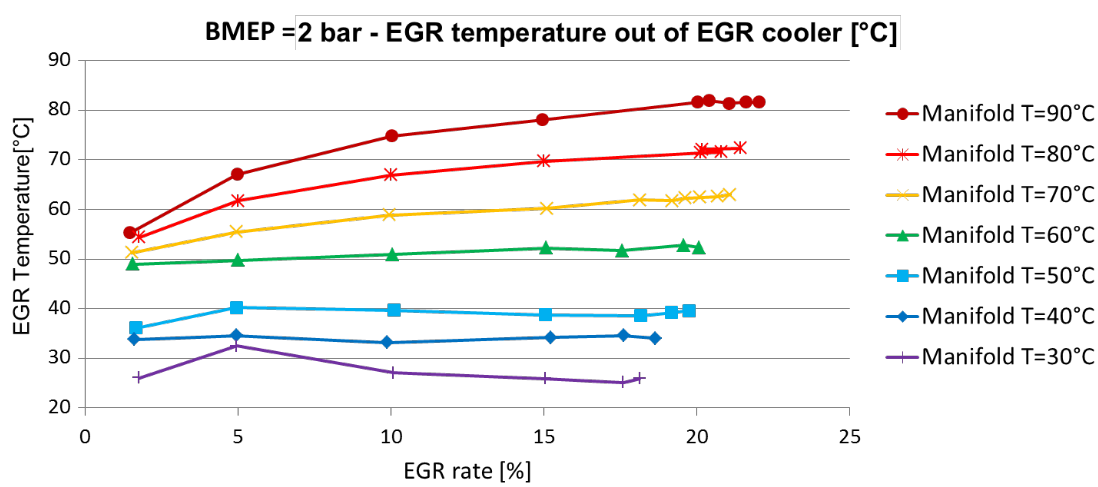


Figure 4.3: *EGR temperature out of the EGR cooler - BMEP = 2 bar*

Looking at the graph, the effect of the connection of the EGR cooler to the cooling cart is demonstrated; in fact, to obtain low manifold temperature the cooling cart is flowing high amount of coolant at a low temperature. This flow is not affecting only the CAC, but also the EGR cooler; the temperature of the recirculated gases is already well below the desired manifold temperature.

An interesting behavior, shown in Figure 4.4, is found when considering the temperature of the flow entering the charge air cooler. The fresh air, after being compressed, and the EGR flow, which, as previously described, is cooled in the EGR cooler, are mixed together and are cooled prior entering the combustion chamber. This cooling is necessary to increase the available power especially at high loads when boosting is required and the air leaving the compressor is at a very high temperature and low density.

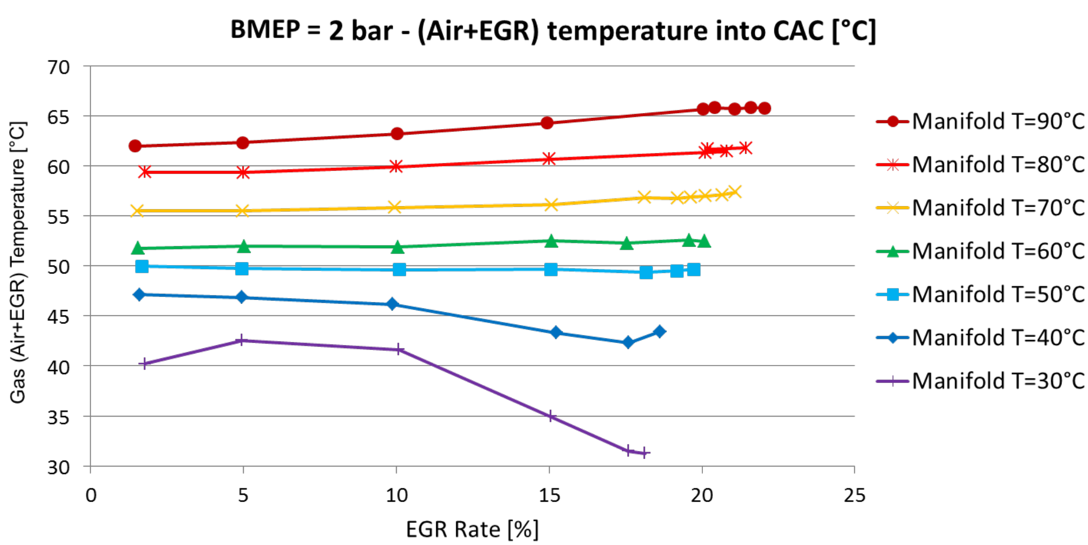


Figure 4.4: *Temperature of the mix of fresh air and EGR before entering the charge air cooler - BMEP = 2 bar*

Looking at the temperature trend it is possible to observe that the charge air cooler is acting as a cooler only when a manifold temperature of 30°C or 40°C is required. In these two tests the charge temperature is decreased to reach the desired target, even if the difference between the inlet is not very high: maximum 12°C for both cases. For the 50°C test the charge is already entering at the desired temperature, so in this case the CAC is not either cooling or heating. For tests at temperatures equal or higher than 60°C, the

CAC was used as a heater. For example, for the 90°C manifold temperature, the mixture is entering at a temperature around 65°C and the increase is then around 25°C. The reason of this effect can be explained considering that:

- the load condition is not requiring the application of high boost, so the air is not compressed at high pressures and the temperature is only slightly increased. The temperature rise along the compressor is only in the range of 13°C, from ambient temperature of 27°C to an average temperature of 40°C
- the EGR gas is cooled in a great extent in the EGR cooler before being mixed with the air out of the compressor, furthermore the exhaust mass flow rate is small compared to the one of the fresh air. Therefore even if the EGR temperature is higher than the 40°C of the air, the temperature increase is small.

All these results on the fluid temperatures before entering the different coolers are used to build the maps implemented in the 1D cooling system model. These data are fundamental to assure that the simulations reflect the actual behavior found during the tests.

Effects of EGR on BSFC

The effects of the different ACT and EGR rate on the brake specific fuel consumption represent one of the major results of this project. The collection of these data is of paramount importance in building the maps for the final analysis on the evaluation of the fuel consumed by the cooling system.

It is important to recall that the brake specific fuel consumption is obtained by dividing the fuel mass flow rate by the engine power and it is measured in [g/kWh], so, if the output power is maintained constant, the lower the value of the BSFC, the higher the fuel economy.

In Figure 4.5 the trend of the BSFC for the different test points is presented. From the graph it is clear that the EGR rate has a great impact in reducing the BSFC, thus the general trend is to have the maximum EGR rate admissible while having at the same time a stable combustion. As highlighted in Table 4.1 an increase of 15% in EGR rate, leads to a reduction in the BSFC around 3.32%. These numbers are not obtained for a specific

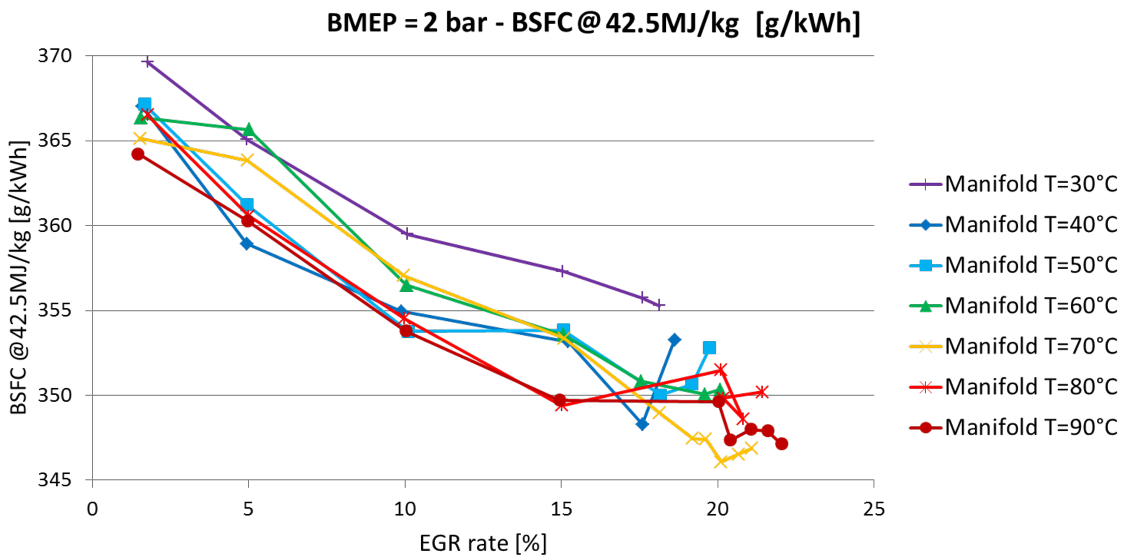


Figure 4.5: *Effects of EGR rate and ACT on the BSFC - BMEP = 2 bar*

temperature, but are considering an average BSFC value for the chosen EGR rate. For example, at 5% EGR, the BSFC value was obtained considering an average value of 362 g/kWh, which is not coming from a particular manifold temperature but it is just chosen as a general reference to quantitatively quantify the reduction in BSFC. The same reasoning is applied for Table 4.2, and Table 4.3.

Table 4.1: *Reduction in BSFC due to EGR rate - BMEP = 2 bar*

EGR	**5%**	**20%**
BSFC [g/kWh]	362	350
ΔBSFC [%]	-3.32	

Looking more in detail Figure 4.5 it is possible to see that the lines are not straight and it is impossible to highlight an effect of intake manifold temperature on BSFC. This scattering in the data is believed to be caused by inaccuracy of fuel flow measurements as such a low flow rate condition.

Some considerations could be made observing the trend on the 30°C manifold temperature. It shows the highest BSFC values in respect to the other temperatures, meaning that the fuel consumption is reduced for higher charge temperatures. This could be explained considering the fact that, for light load operations, the combustion is promoted at higher temperature.

Effects of EGR on spark advance

The positive effects of EGR and ACT on the combustion phasing are showed in Figure 4.6, where the spark advance, in crank angle degrees, is plotted over the EGR rate at the different manifold temperatures.

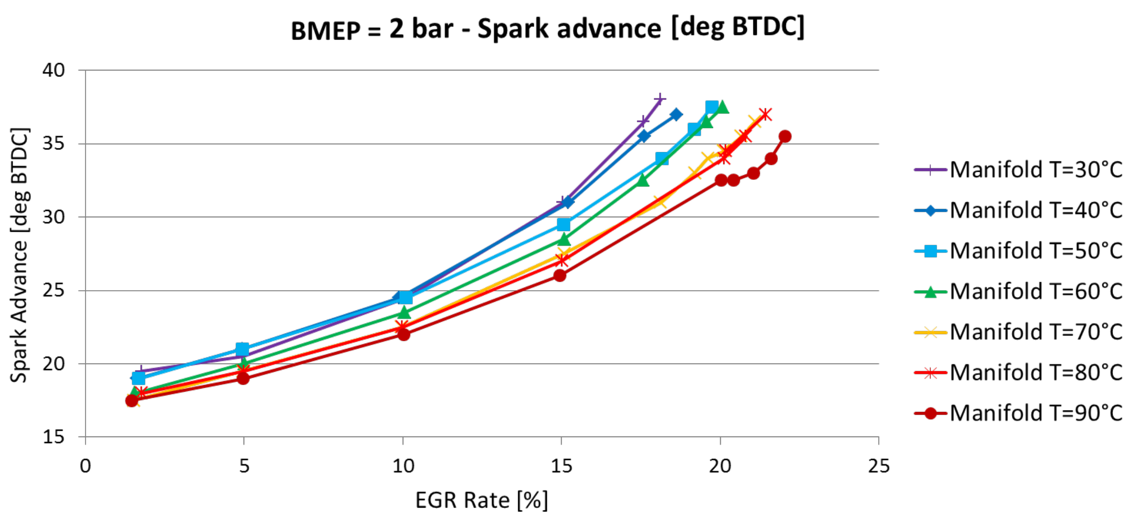


Figure 4.6: *Effects of EGR and ACT on the crank angle spark advance - BMEP = 2 bar*

As expected from previous studies in literature the influence of the EGR on this parameter is very prominent, 10% increase in the EGR rate required 10 degrees of spark advance to maintain the CA50 constant. Considering this parameter, it is possible to note a certain trend for what concern the manifold temperatures, at lower values the spark could be advanced more. When the temperature in the combustion chamber is low, the burning rate is reduced due to a lower flame speed and there is the need to ignite the mixture well before the TDC, thus the highest values of spark advance are found for high EGR rates at lower temperature. This effect will be of fundamental importance for high load conditions.

4.1.2 Results at BMEP = 4 bar

Effects of EGR on COV of IMEP

Also in this working point the effect of knock is not present, even at high manifold temperatures, meaning that, as shown in Figure 4.7, the tests were concluded when the combustion started to become unstable.

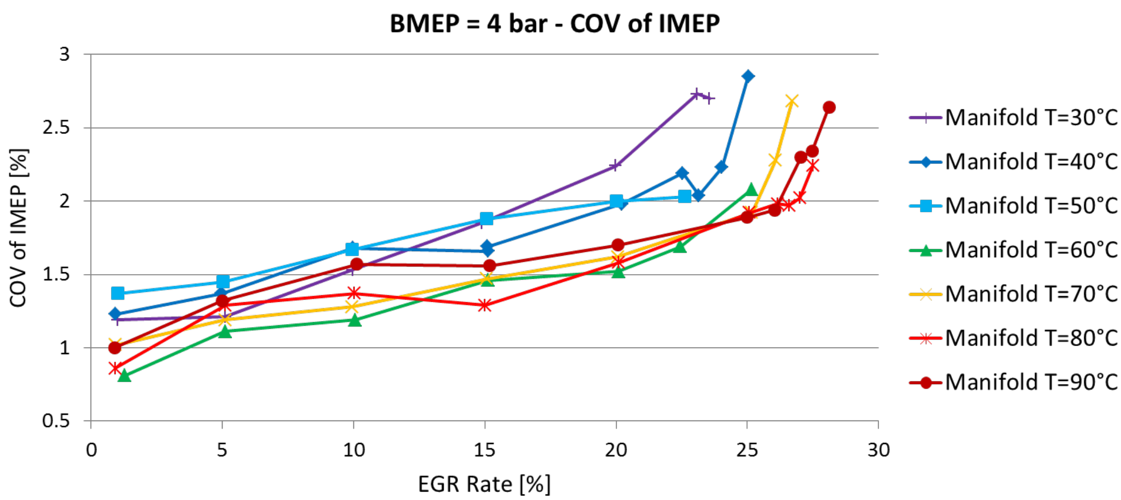


Figure 4.7: *COV of IMEP at different EGR rate and ACT - BMEP = 4 bar*

It is difficult to highlight a trend due to the continuous overlapping of the curves, but the effect that could be noticed is that, a higher temperature could allow the usage of higher EGR rates and at the same time it tends to stabilize more the combustion in this region. Considering for example the curves at 30°C and 90°C, the same value of $COV of IMEP = 2.5$ is obtained respectively at 22% and 27% EGR. This effect of increasing the maximum limit of EGR rate will be highlighted better when considering the BSFC curves.

Effects on gas temperature

As previously stated, the temperatures of the exhaust gas do not depend on the value of the manifold temperature considered in the test, but are just varying depending on the load considered. What is expected is then an increase in the temperature, as shown in Figure 4.8, with a consequent increase in the EGR cooler duty.

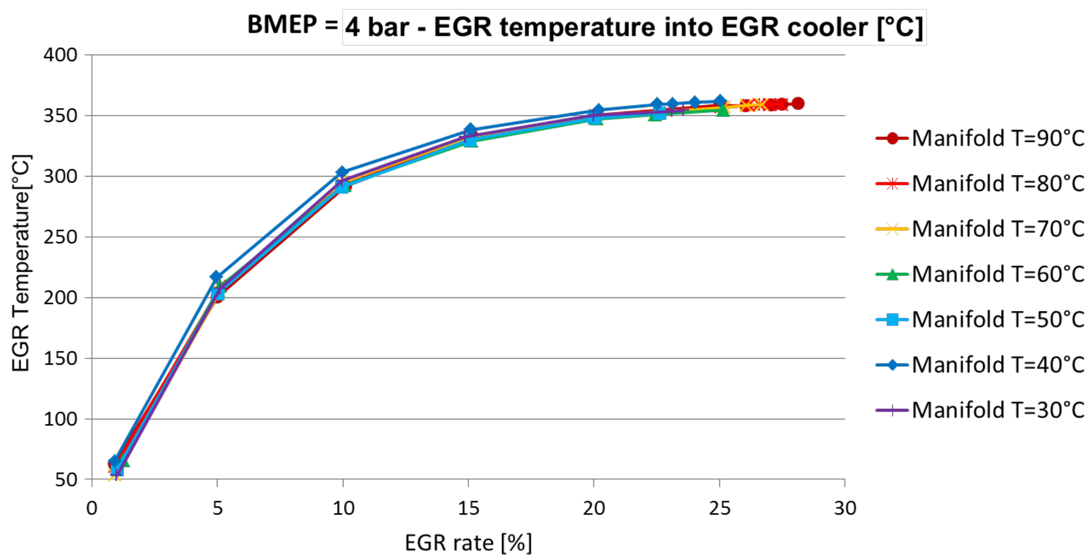


Figure 4.8: *Exhaust gas temperature entering the EGR cooler - BMEP = 4 bar*

The maximum gas temperature entering the cooler is now changed from the value of 290°C in Figure 4.2 to a value of 360°in Figure 4.8. This increase in temperature entering the cooler will be reflected also on the outlet one, in fact, as shown in Figure 4.9, now the EGR temperatures are higher than the ones in Figure 4.3.

Also for this working point no boost is required and consequently the rise in temperature in the compressor is around 13°C, from 27 to 40°C. The main difference however is on the quantity of fresh air flowing, in fact increasing the load, there is the need to increase the amount of air to burn more fuel and obtain a higher output power. Following this reasoning, it is possible to see that the temperature at the CAC inlet is slightly lower than the condition at 2 bar, even if the temperatures of the EGR gas is higher than the previous condition.

Comparing Figure 4.10 and Figure 4.4 no great differences could be noticed, except the fact, above mentioned, that for the 4 bar load the gas temperatures entering the CAC are slightly lower than the 2 bar condition.

Also in this case the charge air cooler is being used as a proper cooler only for the 30 and 40°C manifold temperatures. For all the other ones, it is used as a heater. It is particularly interesting the fact that observing the 50°C curve, in Figure 4.10, the mixture is entering at a lower temperature than the target one, so it is heated in the EGR “cooler”; during

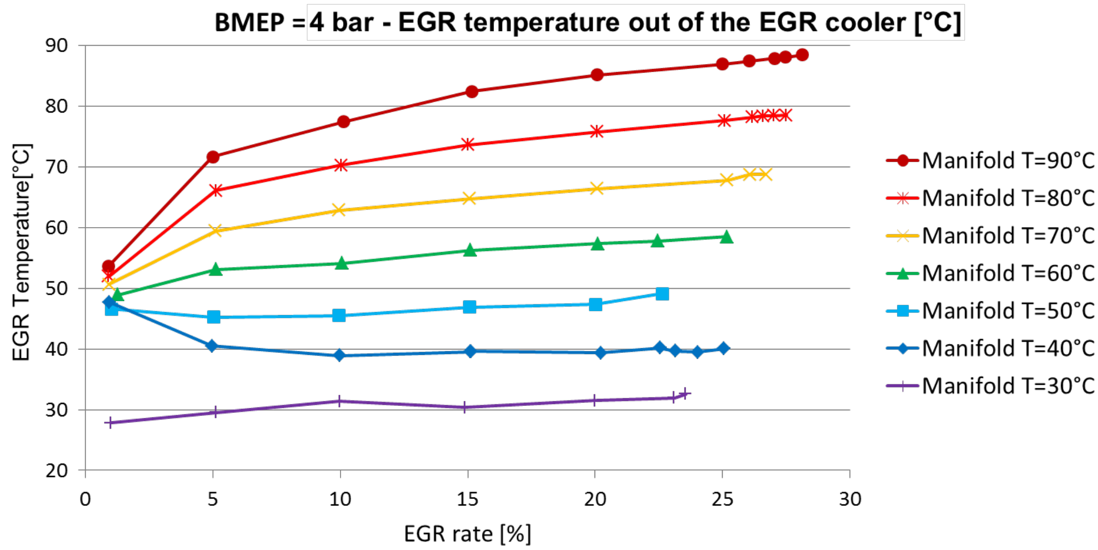


Figure 4.9: EGR temperature out of the EGR cooler - BMEP = 4 bar

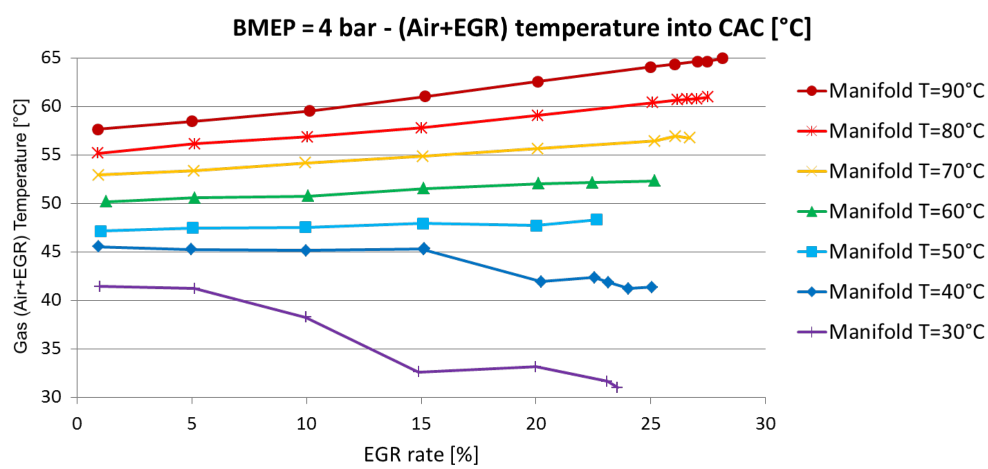


Figure 4.10: Temperature of the mix of fresh air and EGR before entering the charge air cooler - BMEP = 4 bar

the 2 bar tests, instead, the flow was at a temperature comparable to the target and no significant change was observed.

Effects of EGR on BSFC

The effects on the brake specific fuel consumption are respecting the decreasing trend expected at high EGR rates. For this load condition also a certain effect of the ACT could be observed. The curves are still overlapping a bit, but a general behavior could be highlighted.

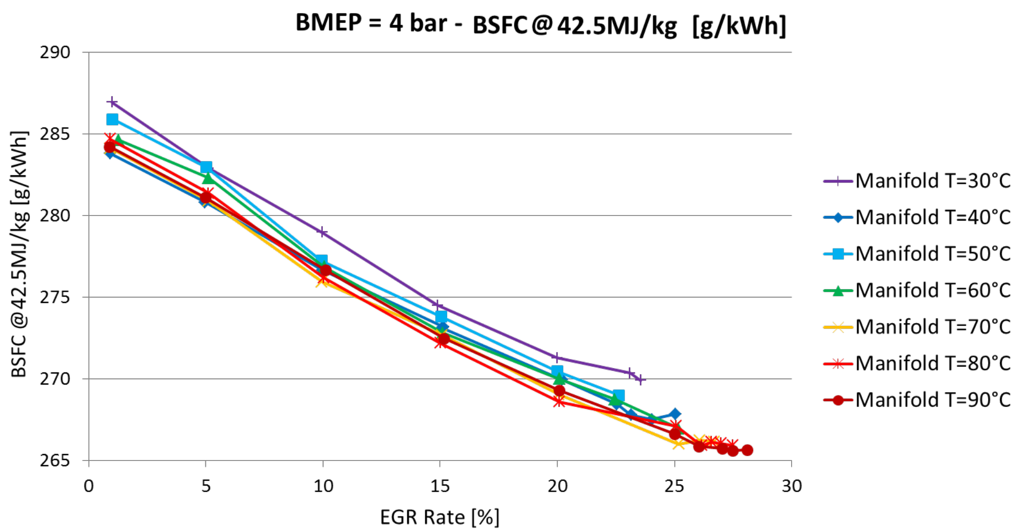


Figure 4.11: *Effects of EGR rate and ACT on the BSFC - BMEP = 4 bar*

Observing Figure 4.5 and comparing it with Figure 4.11, there has been a great reduction in the BSFC range only due to the increased load condition; the ranges are respectively 370-345 [g/kWh] and 285-265 [g/kwh], with an average reduction of 23% only doubling the load. This is in line with what expected considering the brake specific fuel consumption as a function of the load, the curve which is in a parabolic shape: decreasing for increasing load values down to a minimum and then increased again for increasing loads. A schematic behavior is shown in Figure 4.12.

In Figure 4.11 it is possible to start observing a certain trend. The lowest BSFC value is obtained for the highest temperature; in particular, 90°C curve is showing a higher EGR rate admissible (5% more if compared to 30°C) with a decrement in BSFC of 1.85%)in respect to the 30°C case).

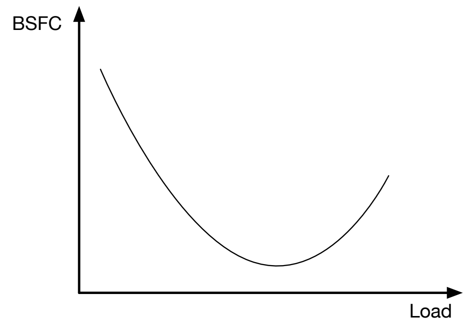


Figure 4.12: *Schematic of BSFC curve at increasing load values*

In Table 4.2 average values for the BSFC at the different EGR rates are considered and increasing it from 5% to 25% could lead to a reduction of about 5.32% in the BSFC.

Again the EGR effect is proven to be very effective in reducing the BSFC, more than the reduction in ACT.

Table 4.2: *Reduction in BSFC due to EGR rate - BMEP = 4 bar*

EGR	**5%**	**25%**
BSFC [g/kWh]	282	267
ΔBSFC [%]	-5.32	

Effects of EGR on spark advance

For what concern the spark advance, the trend is similar to the one showed in Figure 4.6. High EGR rates, as well as lower ACT could allow the possibility to set a higher advance.

It should be noticed that in Figure 4.13 the minimum value, set when the EGR valve is closed, is lower than the one in the 2 bar test case. In both cases the minimum value is set in order to assure that the CA50 is around 8 degrees after TDC in order to get the maximum torque from the given mass of fuel. For higher loads, where the tendency to knock is more problematic, the spark will be retarded more, up to being set later than the TDC.

Another consideration should be made observing the maximum spark advance obtainable and the BSFC curves. They have two opposite trends: the maximum reduction in

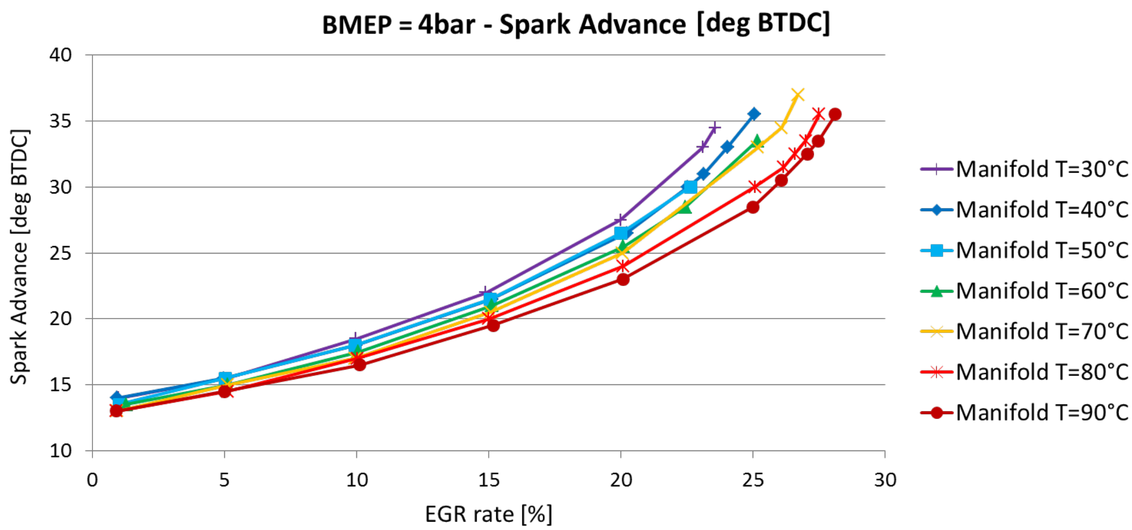


Figure 4.13: *Effects of EGR and ACT on the crank angle spark advance - BMEP = 4 bar*

BSFC is found to appear for higher temperature; while the maximum spark advance is found to appear at lowest temperature. Observing Figure 4.13, the 90°C curve is showing, at maximum EGR rate, the same spark advance of the 30°C one. Following this consideration, a higher EGR rate at higher temperature is favorable since it reduces the BSFC maintaining the same combustion phasing.

4.1.3 Results at BMEP = 6.5 bar

This test condition is starting to increase the stress added on the engine. Boost is not used yet, but attention should be made when controlling the spark timing to avoid knock.

Effects of EGR on COV of IMEP

For this load the maximum EGR rate was found when the COV of IMEP was higher than 3 %, except for the tests performed at 80 and 90°C, where the maximum EGR rate is reached due to insufficient pressure difference between the exhaust and the intake. In the previous test, in fact, the throttle valve was only partially open, reducing the pressure in the intake manifold. The general trend, followed also before, is that, increasing the EGR rate, the throttle valve should be open to admit more air and burn the mixture in a stoichiometric ratio. In addition to that, increasing the load the valve must be opened even more; for this reason, at 6.5 bar load the throttle valve is completely open and the pressure in the intake manifold is increased up to a certain value similar to the exhaust one, decreasing thus the driving force for the exhaust recirculation.

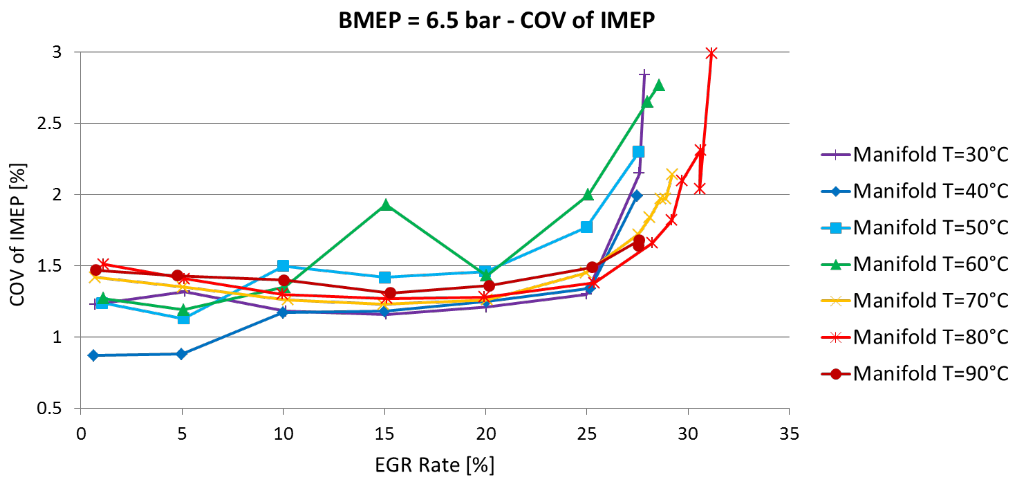


Figure 4.14: *COV of IMEP at different EGR rate and ACT - BMEP = 6.5 bar*

Observing Figure 4.14, especially the 90°C curve, it is very clear the effect explained above. The maximum EGR rate is only 27% with only a $COVofIMEP = 1.7$; for all the other temperature the limitation on the EGR admissible was found looking at the value of the COV of IMEP. The 80°C test is a borderline test because the maximum EGR rate is found when the pressure difference between the intake and the exhaust was not high enough to go further in the recirculation, and it also shows a $COVofIMEP = 2.99$.

Effects on gas temperature

As expected, increasing the load also the temperature of the exhaust gas is increased. The trend, similar to the previous ones, is showed in Figure 4.15. The increment in inlet temperature is followed by an increment of the outlet one as well, as reported in Figure 4.16.

Since no boost is required, the temperature of the fresh air is not increased and the considerations applied for the loads in the previous sections are still valid. The trend of the CAC temperature inlet is shown in Figure 4.17. Also in these tests the charge air cooler

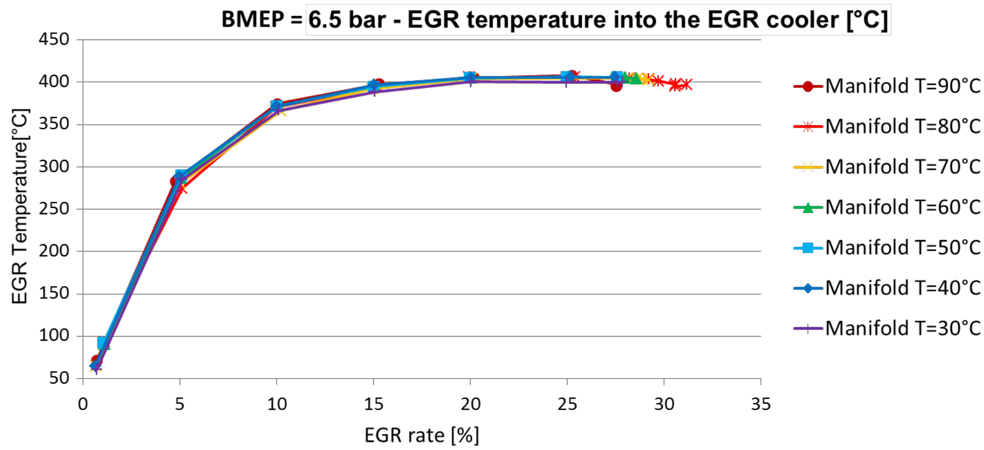


Figure 4.15: *Exhaust gas temperature entering the EGR cooler - BMEP = 6.5 bar*

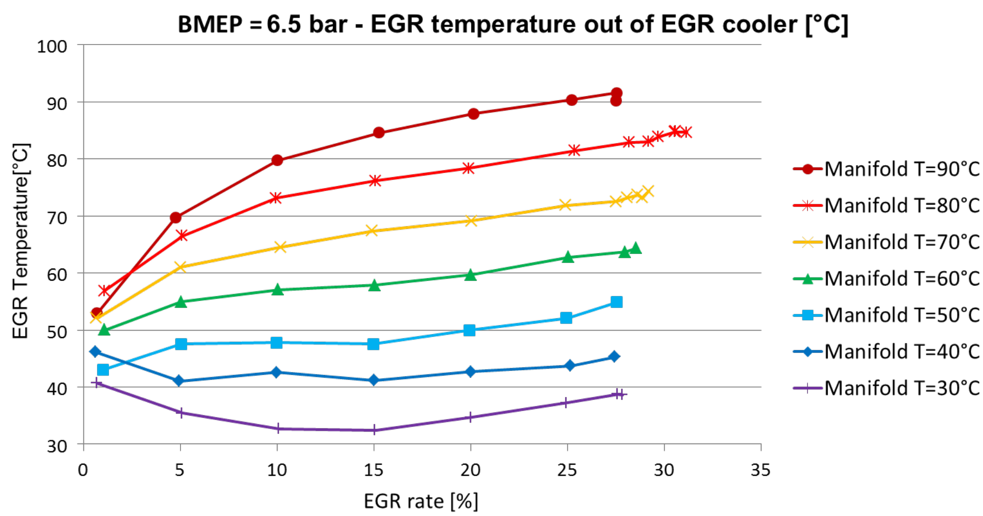


Figure 4.16: *EGR temperature out of the EGR cooler - BMEP = 6.5 bar*

has been used as a real cooler only in the 30 and 40°C conditions, for all the other points it has been used to heat up the mixture.

Effects of EGR on BSFC

For the BSFC the same reasoning done for the previous tests is applied, the range over which the values are placed is reduced further, meaning that this load condition is still in

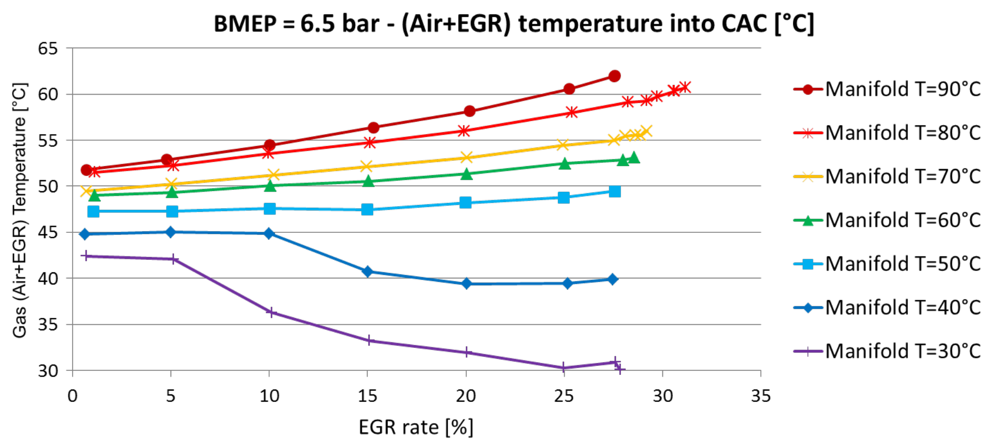


Figure 4.17: *Temperature of the mix of fresh air and EGR before entering the charge air cooler - BMEP = 6.5 bar*

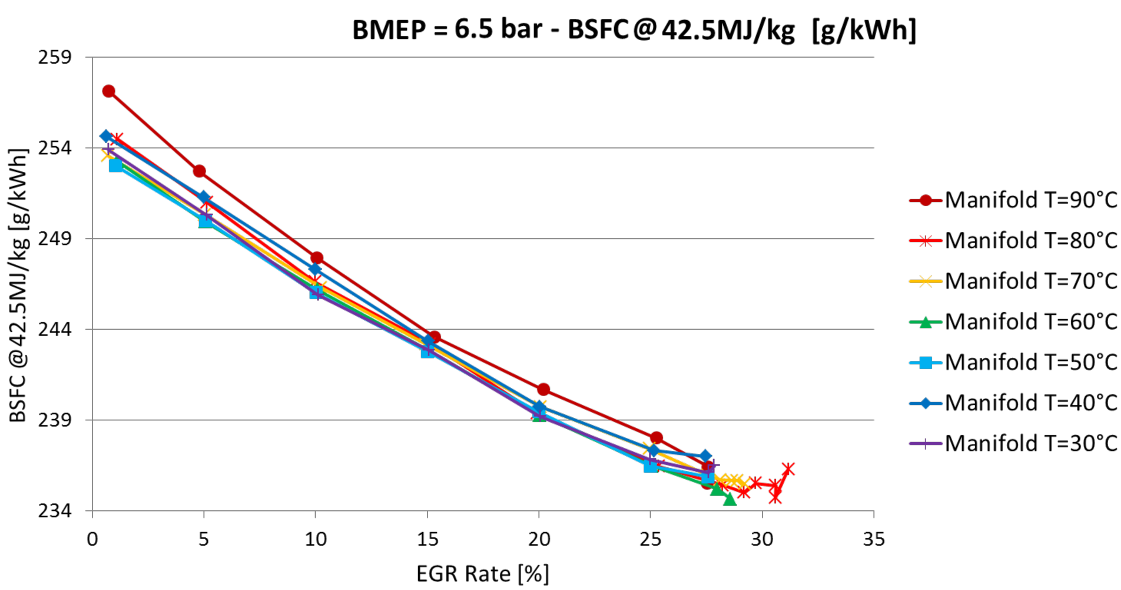


Figure 4.18: *Effects of EGR rate and ACT on the BSFC - BMEP = 6.5 bar*

the descending branch of the curve in Figure 4.12.

For this load the BSFC range is between 254 and 235 [g/kWh], as reported in Figure 4.18, which corresponds, in respect to the previous load, to a decrement of 10.1% in the average BSFC values. The EGR rate is further increase up to 30% for the manifold temperature of 90°C. The percentage decrease of the BSFC due to the variation in the EGR rate is highlighted in Table 4.3

It should be noticed that a certain trend in the temperature is appearing, and it is actually the opposite of the previous twos. In this case in fact, the higher BSFC curve corresponds also to the highest manifold temperature.

For the engine type and speed considered, the 6.5 bar condition represent, as will be observed analyzing the other loads, the transition point over which it is better to have colder mixture rather than hot ones and over which the BSFC will start to follow the ascending branch of Figure 4.12.

Table 4.3: *Reduction in BSFC due to EGR rate*

EGR	**5%**	**27%**
BSFC [g/kWh]	252	236
ΔBSFC [%]	-6.34	

Effects of EGR on spark advance

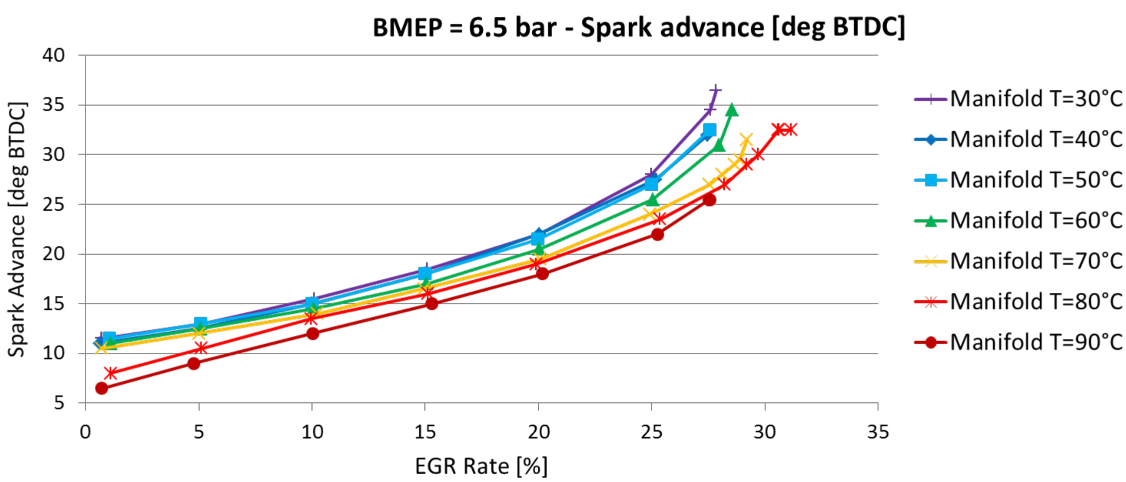


Figure 4.19: *Effects of EGR and ACT on the crank angle spark advance - BMEP = 6.5 bar*

For the spark advance the reasoning followed during the previous analysis could be applied as well. In this condition the effect of temperature is even more predominant, and considering also the BSFC trend, it is better now to have high EGR rate at a low temperature.

It should be observed that the minimum value of the spark advance 0% EGR is closer to the TDC, around 6 to 10 degrees BTDC.

4.1.4 Results at BMEP = 10 bar

For this working point the intake was boosted in order to meet the target load while delivering enough EGR.

Effects of EGR on COV of IMEP

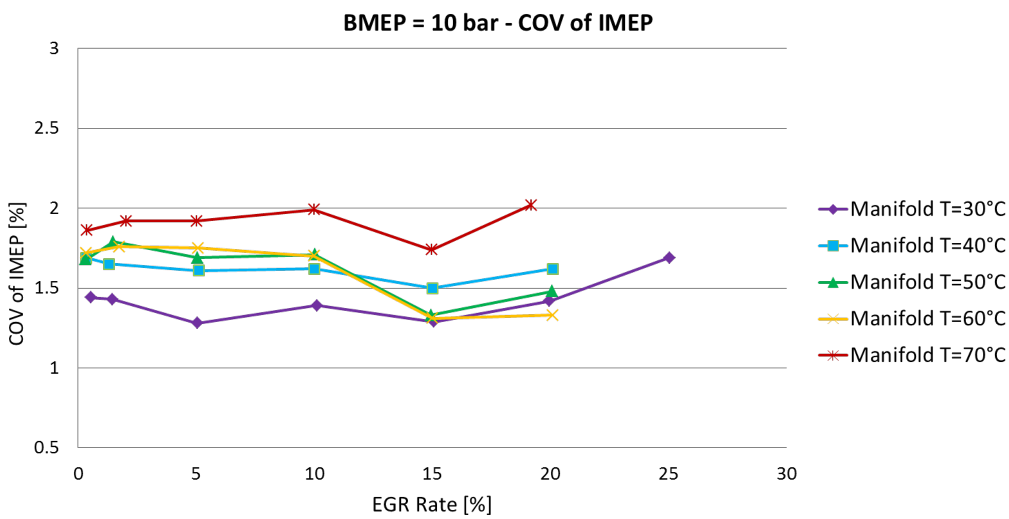


Figure 4.20: *COV of IMEP at different EGR rate and ACT - BMEP = 10 bar*

In this series of tests the maximum EGR rate is obtained no more due to the increase in the COV of IMEP value, as shown in Figure 4.20, but due to insufficient boost or onset of knock. Great attention must be kept in this condition to avoid knock and possible damages to the engine.

It should be noticed that the manifold temperature of 80 and 90°C are no more present. This is because of knocking problems, in fact at this load, such high temperature easily favor the onset of knock. It has been decided not to go further than 70°C because it was already representing a very severe condition difficult to be controlled properly.

Effects on gas temperature

The temperature of the exhaust gases entering the cooler are now slightly lower than the previous case. The maximum value, as showed in Figure 4.21, is around 360°C.

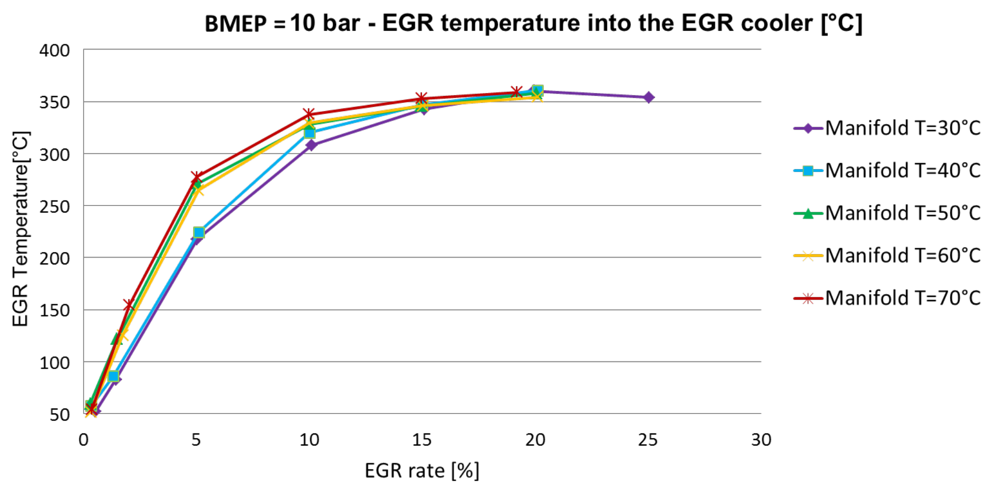


Figure 4.21: *Exhaust gas temperature entering the EGR cooler - BMEP = 10 bar*

The exhaust gas, now, after being cooled, has almost the same trend independently on the manifold temperature desired. It is important to notice, in Figure 4.22, that the downward spike at 1% EGR is related to the way the temperature is measured. In fact, prior to that point the EGR valve is closed and what is measured is an high temperature due to metal surfaces radiation, once the valve is open, a certain amount of EGR is starting flowing and the measurements are accurate.

From this load on, it is possible to note the usage of turbocharging observing the trend of the temperature at the entrance of the CAC. Boosting is increased as the EGR rate is increased, meaning that also the fresh air leaving the compressor has increased its temperature, this explains the trend observed in Figure 4.23.

It should be pointed that in this case the CAC is used as a cooler for high EGR rates and the 30 and 40°C manifold temperature conditions, while for low EGR rates and other temperature it is still used as a heater.

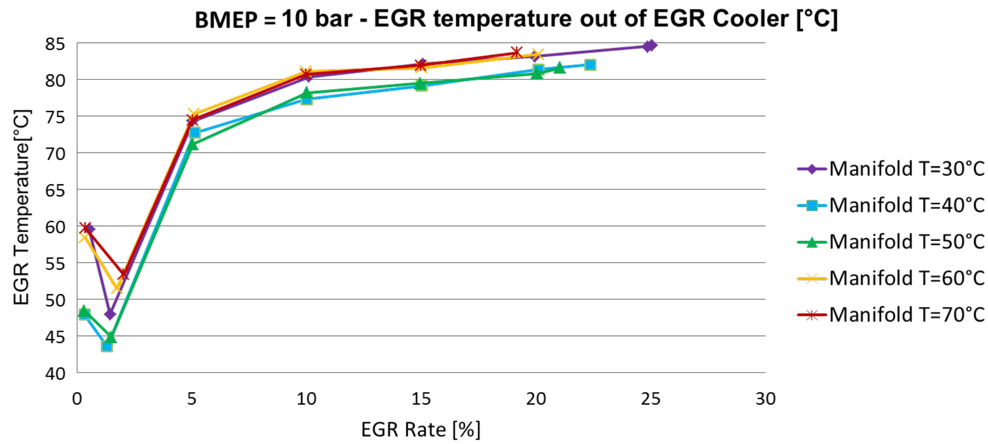


Figure 4.22: EGR temperature out of the EGR cooler - BMEP = 10 bar

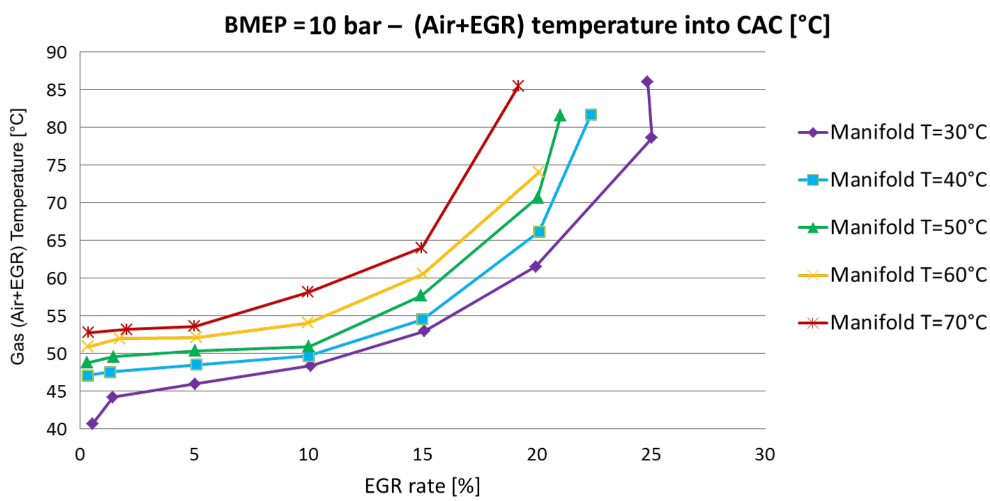


Figure 4.23: Temperature of the mix of fresh air and EGR before entering the charge air cooler - BMEP = 10 bar

Effects of EGR on BSFC

As previously mentioned, the BSFC is starting to increase. The 6.5 bar load was placed near the minimum point of Figure 4.12, meaning that, for loads higher than that, the BSFC has to increase. This is what is happening for these results: the range is now increased to values from 265 to 235 [g/kWh], as reported in Figure 4.24, which is an increase in 1% on the average.

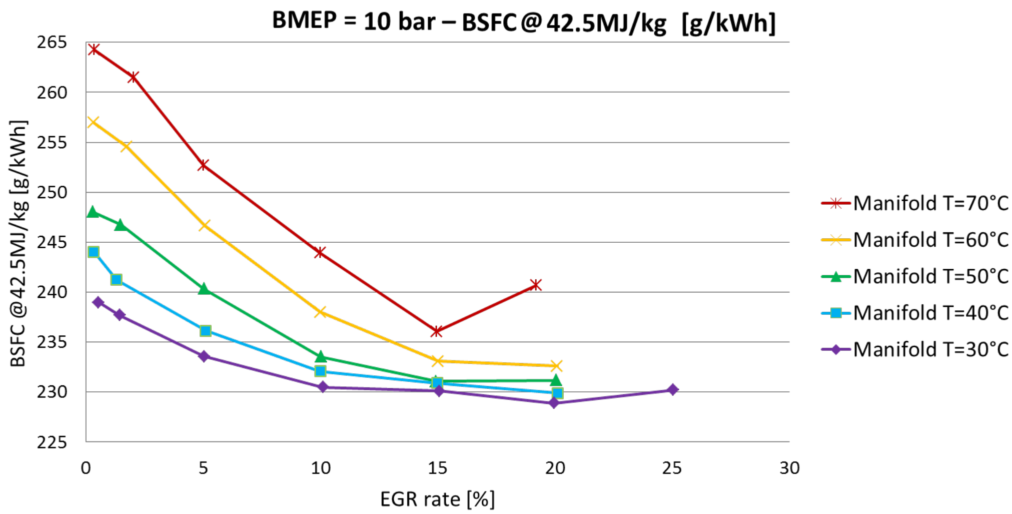


Figure 4.24: *Effects of EGR rate and ACT on the BSFC - BMEP = 10 bar*

It must be observed that now a clear temperature trend is appeared. The EGR rate is still playing an important role in reducing the BSFC value, but, especially at low recirculation values, the ACT effect is predominant. Around 1% EGR, reducing the manifold temperature from 70 to 30°C lead to a reduction on the BSFC value from respectively 262 to 237 [g/kWh], almost a 9.5% reduction.

If considering the effect of the EGR rate at 30°C manifold temperature, it could be observed that at 1% EGR the $BSFC = 237$ $[g/kWh]$ while at 25% EGR the $BSFC = 230$ $[g/kWh]$, a reduction of only 2.95% is observed.

However, when high EGR rate at low temperature is used, the decrease in the BSFC is higher because is not considering the two effects (of low temperature and high EGR rate) as distinct, but together, in synergy.

Effects of EGR on spark advance

The trend of the spark advance is particularly interesting when considering this load, in fact, as it is shown in Figure 4.25, the minimum spark advance is at negative values, meaning that the spark event is actually after the TDC; it should then more correctly called spark retard.

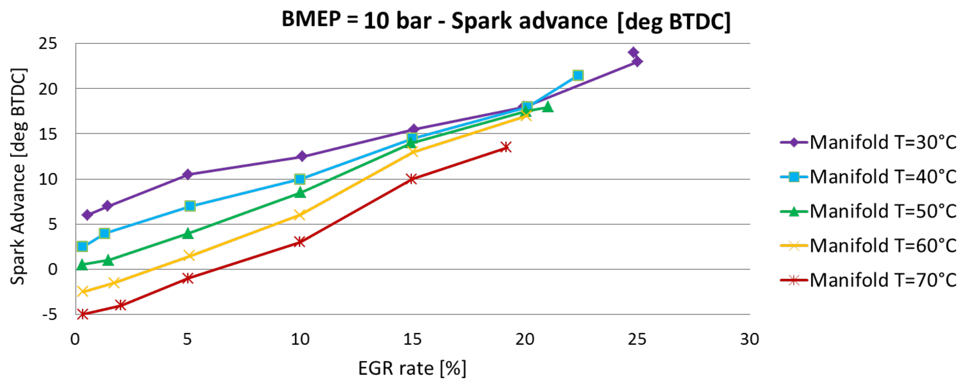


Figure 4.25: *Effects of EGR and ACT on the crank angle spark advance*

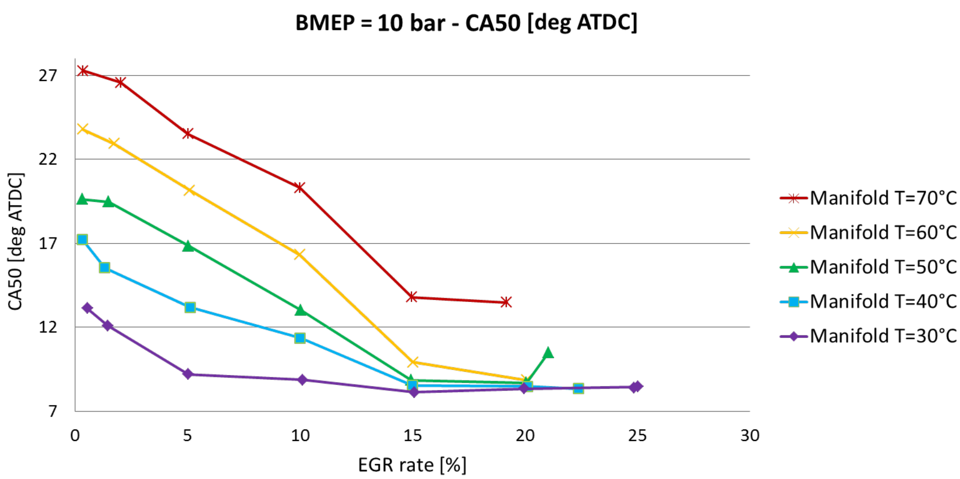


Figure 4.26: *Effects of EGR and ACT on the CA50 - BMEP = 10 bar*

Figure 4.25 is particularly interesting because it helps in understanding the effect of temperature on the combustion phasing. At 60 and 70°C in fact the combustion would be knock limited if the spark is not retarded enough, this fact justifies also the increase of the BSFC for these two temperatures. Exhaust gas recirculation and reduction of manifold temperature are, in literature, two solutions proposed to reduce knocking; looking at Figure 4.25 it is evident the great effect they have to counteract these abnormal combustion events.

The effect on the combustion phasing mentioned above could be also observed looking at Figure 4.26, where the CA50 at different EGR rates and manifold temperature is plotted.

A value around 8, which is the angle after the TDC at which the 50% of the heat is released, is the target, so the closer to this value the better is the combustion process. It should be noticed that the target is not reached only for a manifold temperature of 70°C.

4.1.5 Results at BMEP = 15 bar

This condition, as will be observed by the small amount of data available, was very severe for the engine. All the tests were ended due to the instability of combustion, this is demonstrated by the fact that at 60°C only two points were collected.

Effects of EGR on COV of IMEP

The analysis for the COV of IMEP is similar to the one at 10 bar, the only difference is that the points at 60°C, being particularly knocking limited, show a $COV of IMEP > 3$. The results are reported in Figure 4.27.

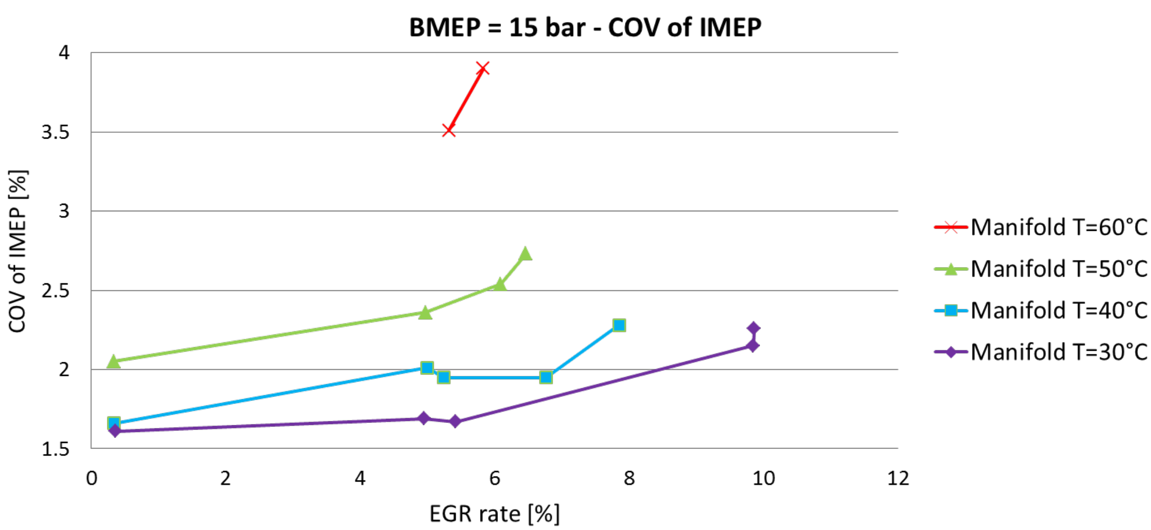


Figure 4.27: *COV of IMEP at different EGR rate and ACT - BMEP = 15 bar*

Effects on gas temperature

To reach the necessary power output to test the engine at the desired load, high levels of boost must be used. This fact is then reflected on the temperature of the gases entering the different coolers. The exhaust reaches a temperature around 400°C before entering the EGR cooler (Figure 4.28).

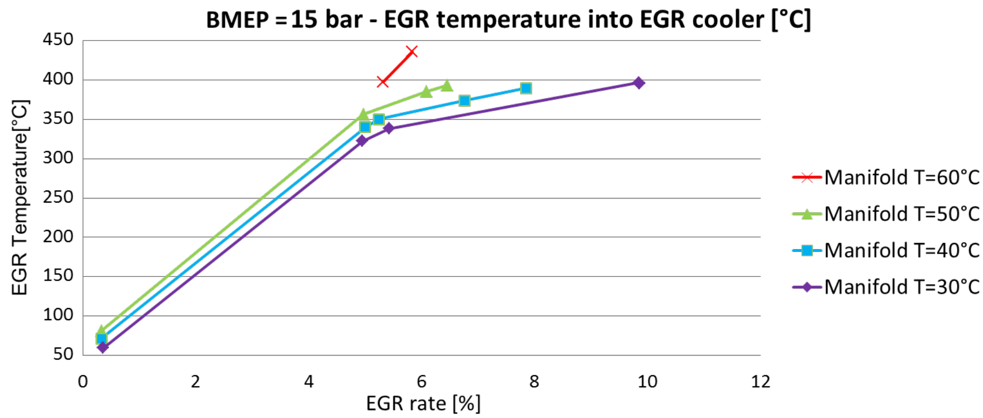


Figure 4.28: *Exhaust gas temperature entering the EGR cooler - BMEP = 15 bar*

The EGR cooler, despite the increase in temperature of the inlet gas in respect to the previous condition (10 bar), is still able to lower down the exhaust temperature at 80 °C prior its mixing with the fresh air (Figure 4.29).

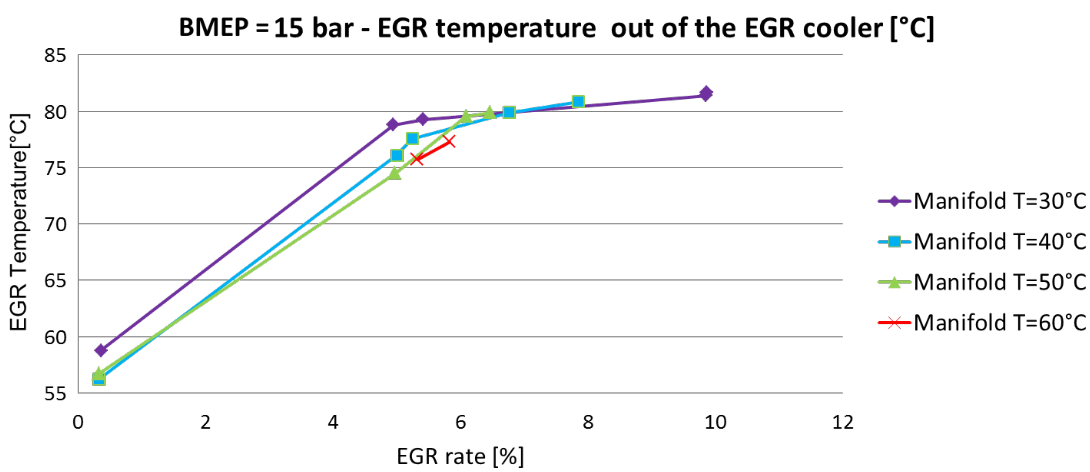


Figure 4.29: *EGR temperature out of the EGR cooler - BMEP = 15 bar*

The mix is then compressed and, due to the need of high boost values, the temperature is increased to a certain extent in all conditions. The range is between 80 and 120 °C, as shown in Figure 4.30; this is the first load for which the CAC is used as a proper cooler for all the EGR rates and ACT, and actually its thermal duty is particularly intense since, for example considering the 30°C curve, it has to lower, in average, the temperature from 90 to 30°C.

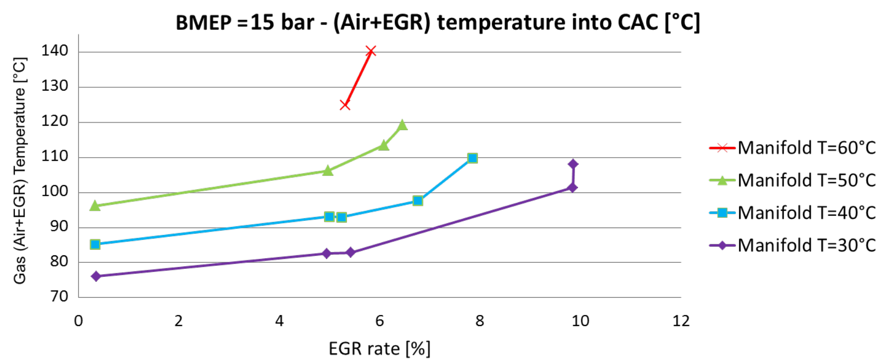


Figure 4.30: *Temperature of the mix of fresh air and EGR before entering the charge air cooler - BMEP = 15 bar*

Effects of EGR on BSFC

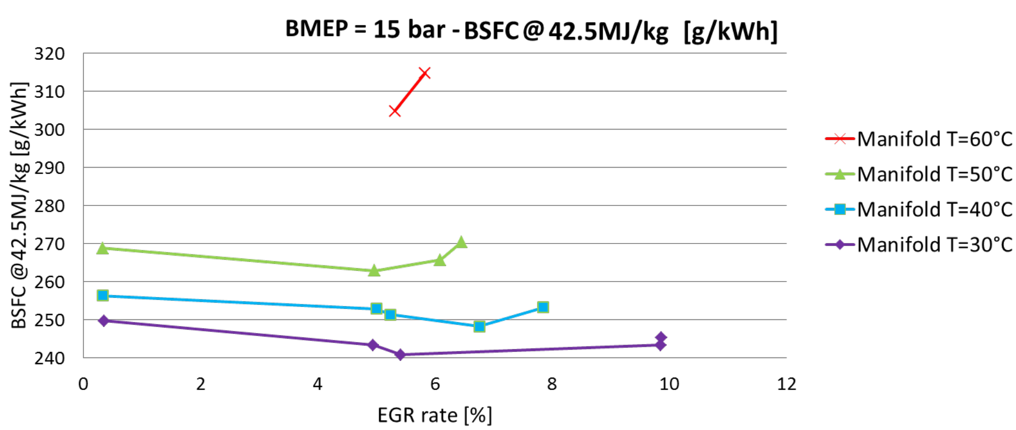


Figure 4.31: *Effects of EGR rate and ACT on the BSFC - BMEP = 15 bar*

The BSFC, as expected from all the previous considerations, is further increased in respect to the previous load point. From Figure 4.31 it is found that the 60°C manifold temperature, due to its extremely high instability, is worsening a lot the BSFC. The average range for all the other temperatures is between 260 to 250 [g/kWh], while for the 60°C the results are around 310 [g/kWh]. The average increase in respect to the previous condition is around 5.88%.

Effects of EGR on spark advance

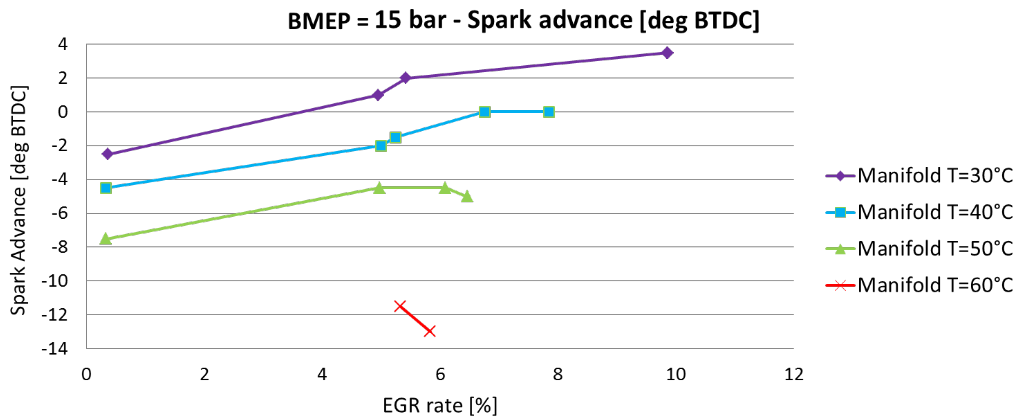


Figure 4.32: *Effects of EGR and ACT on the crank angle spark advance - BMEP = 15 bar*

The spark advance trend, Figure 4.32 is showing that, when the EGR valve was closed, all the points were knock limited, in fact the spark is retarded by 2 to 8 degrees. The general trend is the one already showed for all the working points, the main difference is that for certain conditions the spark event is set at points always after the TDC (50°C curve) or slightly before it.

The best results is found for the 30°C condition, where an advance of 3.8 degrees is reached, but when compared with the 35 of the previous tests it is clear why this load is showing very poor BSFC values.

It is important also to note that the maximum EGR rate is located around 10%. This lowering in the maximum amount admissible was already observed for the test at 10 bar, but now it is more accentuated. This low point is limiting the EGR positive effects, as extensively described in the previous section, and the causes have to be associated to the great instability of the combustion process.

The CA50, similarly to the previous case, is showing values far from the required one,

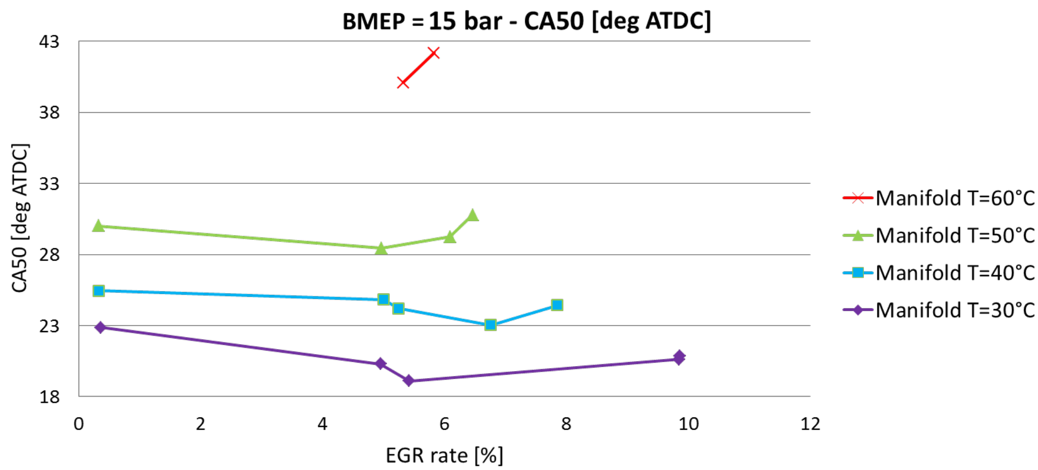


Figure 4.33: *Effects of EGR rate and ACT in the CA50 - BMEP = 15 bar*

Figure 4.33, the lowest result (19 degrees) is obtained with high EGR rate at a manifold temperature of 30°C.

4.1.6 Results at BMEP = 17.34 bar

This is the most severe condition considered in the test. It is run at a temperature of 36°C, an engine speed of 1997 rpm and a $load = 17.34\ bar$ and it is critical for the engine. At this particular working point, a study on the effects of the enrichment ($\lambda = 0.8$) on the fuel consumption is performed. For this last case a manifold temperature of 50°C is set and the tests are run trying to maintain the same CA50 of the 40°C test; in doing so a direct comparison could be made. The enrichment was analyzed to highlight whether if, at this high ambient temperature, it better to cool the engine with the cooling system or injecting additional fuel in the chamber, which evaporates and subtract heat.

Effects of EGR on COV of IMEP

The trend showed in Figure 4.34 for the COV of IMEP is in line with the previous ones. As for the other tests, where boost is applied, also here the maximum EGR rate admissible is obtained at the onset of knock. It should be observed that, the curve at 70°C is at very high values of COV of IMEP, higher than the target of 3; the best result is obtained instead when the mixture is enriched.

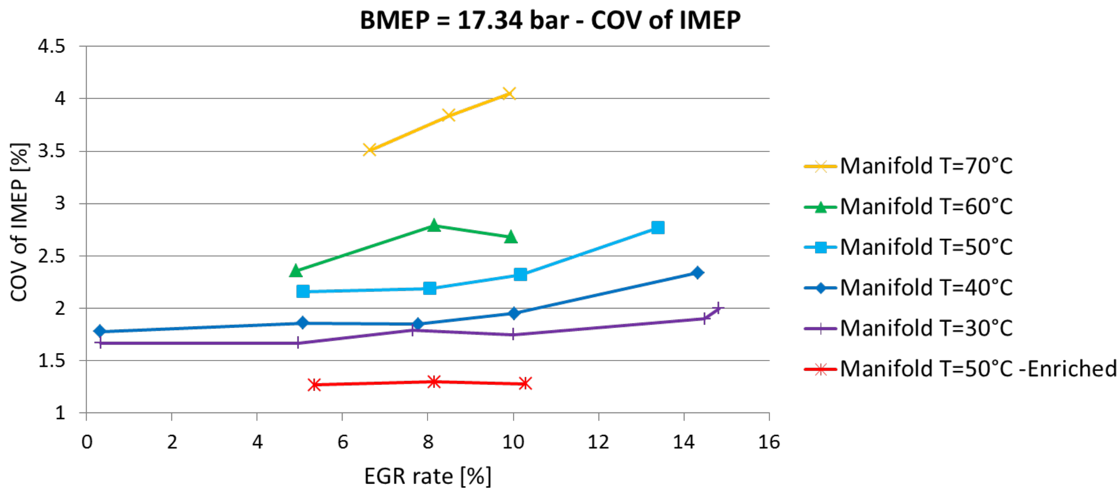


Figure 4.34: COV of IMEP at different EGR rate and ACT - BMEP = 17.34 bar

Effects on gas temperature

The effects of the increased load, but also of the increased ambient temperature are clearly visible when analyzing the temperature of the gases entering the different coolers.

The exhaust temperature prior the entrance in the EGR cooler is much higher than the previous test, around 420-500°C. The effect of the enrichment (red curve) on lowering the temperature inside the chamber is clearly visible in Figure 4.35. This mean that this technique could be favorable to reduce knock problems, the main drawback, as will be highlighted in the following is the great increase in the BSFC value.

The EGR cooler is still very effective in reducing the temperature of the exhaust gas, in fact, as observed in Figure 4.29, also in this condition the average outlet gas temperature is around 83°C, as retrieved from Figure 4.36. This is a very good result if it is considered the very stressful conditions of this test.

As previously mentioned in the 15 bar load, the CAC is now acting, in all temperature conditions, as a proper cooler. The mixture of air and recirculated gas is always entering the heat exchanger at a temperature greatly higher than the target one. The reason of this increase in charge air cooler inlet temperature could be addressed looking at three factors: higher ambient temperature, higher boost levels and higher EGR temperature prior to the

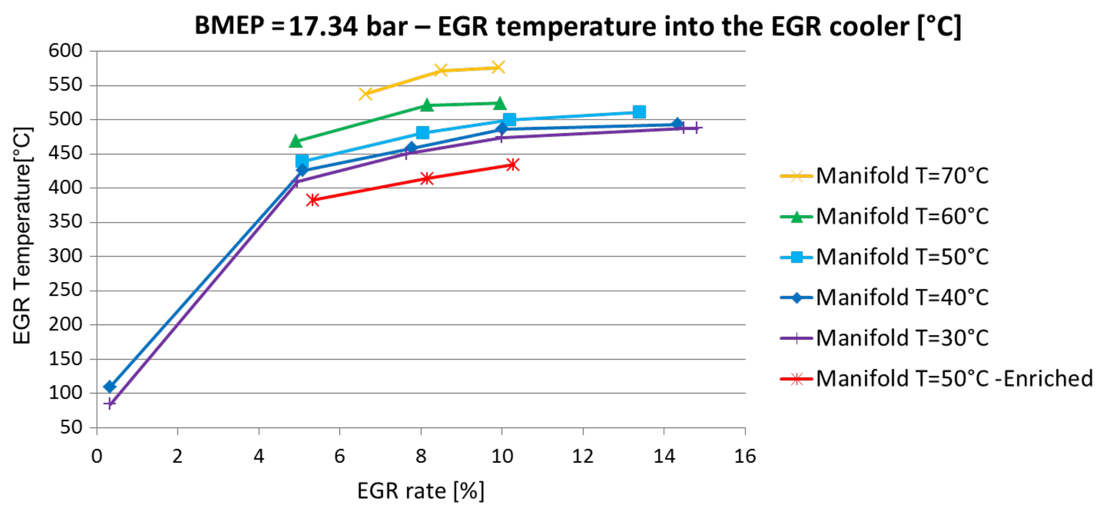


Figure 4.35: *Exhaust gas temperature entering the EGR cooler - BMEP = 17.34 bar*

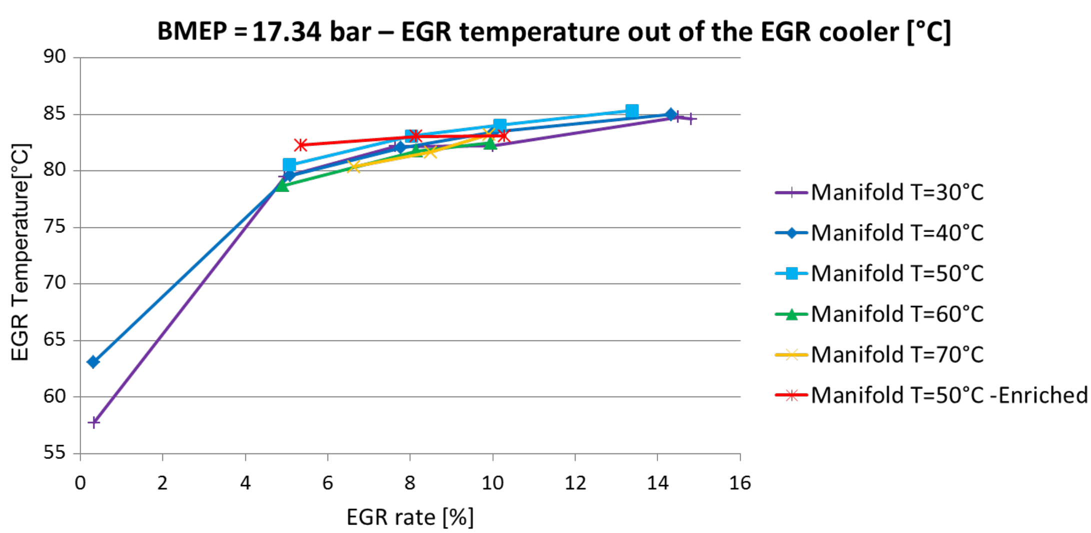


Figure 4.36: *EGR temperature out of the EGR cooler - BMEP = 17.34 bar*

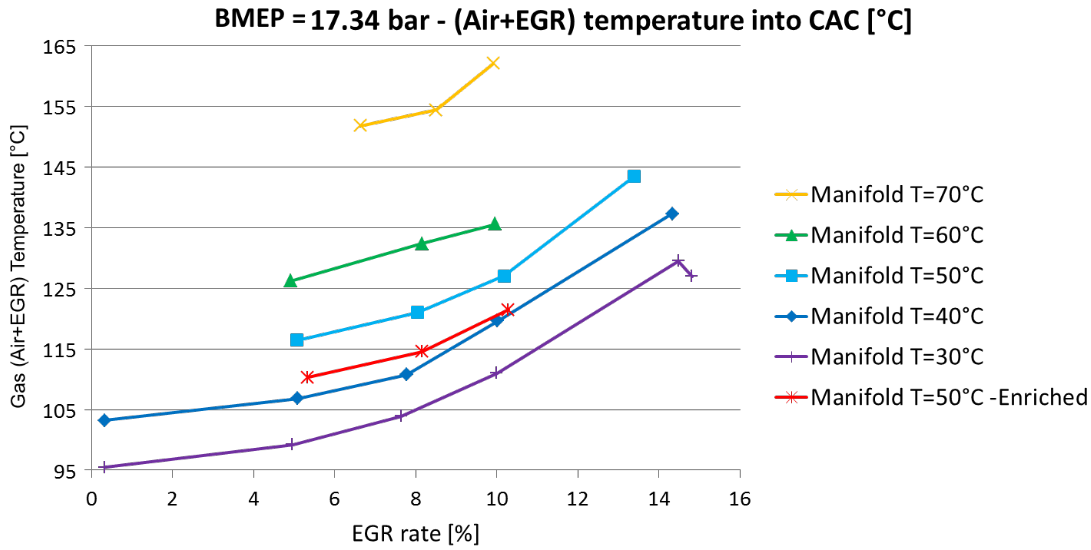


Figure 4.37: Temperature of the mix of fresh air and EGR before entering the charge air cooler - BMEP = 17.34 bar

mixing with fresh air. All these effects combined together lead to the trend showed in Figure 4.37.

Effects of EGR on BSFC

A straightforward reasoning on the BSFC trend is no more valid for this condition since the engine speed is moved from 1500 rpm to 1997 rpm, however, being a very high load condition, the expected range of values should be particularly high. This is confirmed from Figure 4.38, where the BSFC is around 260-230 [g/kWh]; exceptions are found for the 70 °C and for the enrichment case. From this last observation it is clear that, unless strictly necessary, enrichment should be avoided since it contributes a lot in increasing the fuel consumption.

As expected for high load conditions, also in this case lowering the temperature is very effective in reducing the BSFC, however attention should be made because cooling in this condition could be difficult and could lead to a high fuel consumption. The scope of the project will be indeed the investigation of the effects of this additional load on the cooling system.

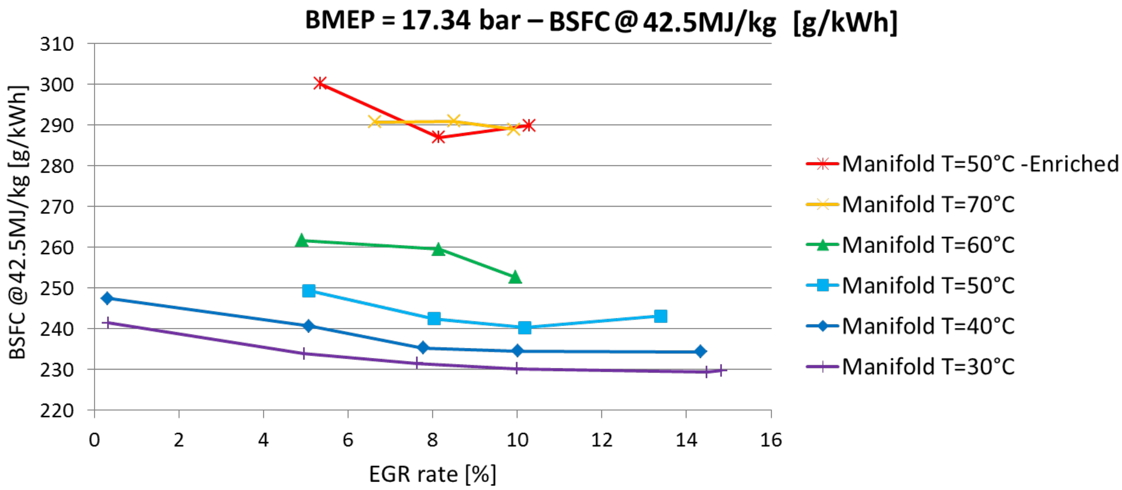


Figure 4.38: *Effects of EGR rate and ACT on the BSFC - BMEP = 17.34 bar*

Effects of EGR on spark advance

As the test ran at 15 bar, the spark should be retarded for most of the conditions when low values of EGR are considered. The positive effect of both low air charge temperature and high EGR rate is present and it is significant that the highest gain in spark timing is obtained when the manifold temperature is set at 30°C and an EGR rate of 15%.

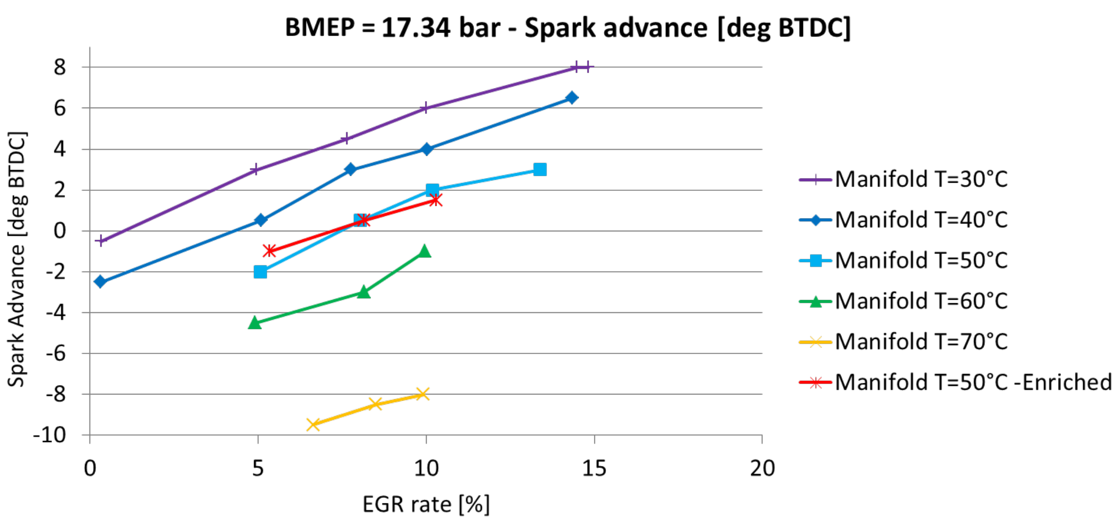


Figure 4.39: *Effects of EGR and ACT on the crank angle spark advance - BMEP = 17.34 bar*

Higher temperature, in high load conditions, are not only worsening the combustion phasing, but are also reducing the maximum admissible amount of EGR rate, as showed in Figure 4.39.

For what concern the spark timing, enrichment is not very effective and does not show any useful improvement, in fact the two curves at 50°C are almost overlapped for all the points, meaning that enrichment is not affecting the spark timing.

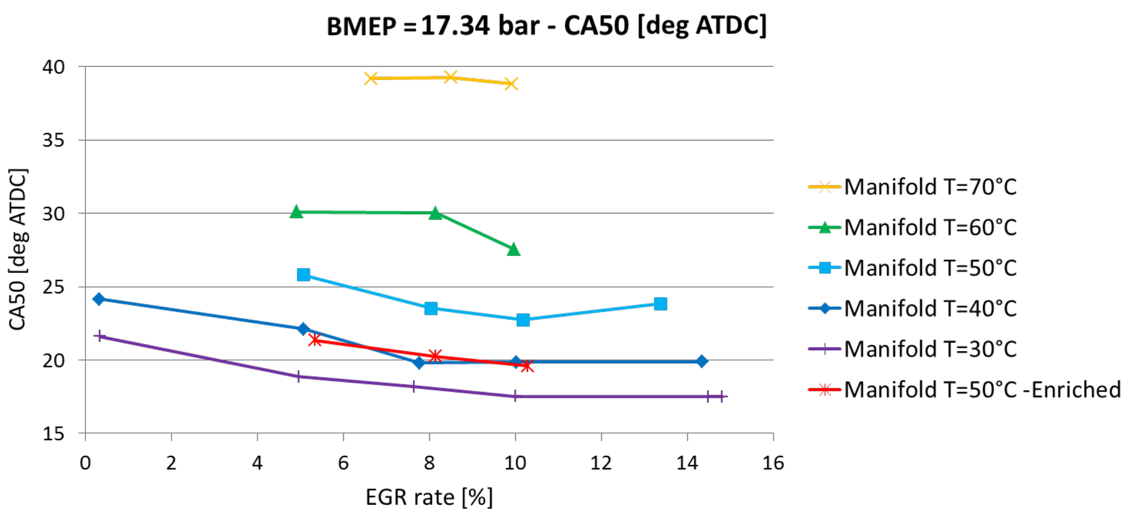


Figure 4.40: *Effects of EGR and ACT on the CA50 - BMEP = 17.34 bar*

The trend for the CA50 is the same of the one showed for the 15 bar condition. It should be noticed that the enrichment (red) and the 40°C (light blue) curves are overlapped; this is because of the assumption made at the beginning: to make a comparison possible between enrichment and the other points, the CA50 was set equal to the one at a manifold temperature of 40°C.

4.2 Pumping work reduction

As general results, it is interesting to highlight the effect of the manifold temperature on the manifold pressure and on the pumping mean effective pressure (PMEP). The graphs presented in the following are obtained from the results of the tests ran at 6.5 bar BMEP. In Appendix B the values of the PMEP for the other loads are reported.

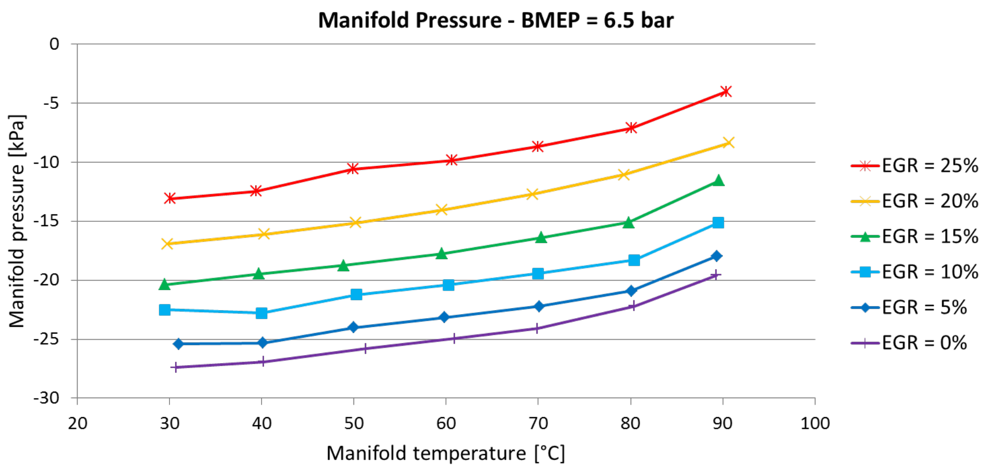


Figure 4.41: *Manifold pressure as a function of manifold temperature at different EGR rates*

This result is interesting because having a higher temperature is directly translated in lower air density. Therefore, in order to maintain constant BMEP, the throttle valve should be opened further as the intake manifold temperature increases.

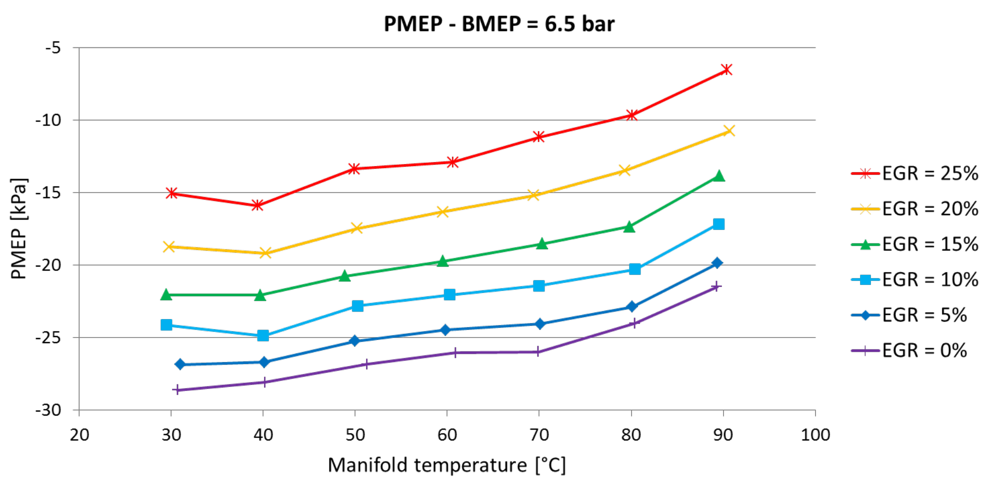


Figure 4.42: *Effects of manifold temperature and EGR rate on PMEP*

The effect of the increased manifold temperature over the pressure is showed in Figure 4.41.

For what concern the PMEP, the trend, similar to the manifold pressure one, is reported in Figure 4.42. As the intake manifold temperature increases, the throttle valve should open up more. This further opening of the throttle valve resulted in reduced pumping loss (i.e., PMEP). This trend of improving PMEP as a function of intake manifold temperature is shown in Figure 4.42. Furthermore, this trend is observed in all 6 different EGR rates tested. Differentiating the contribution of EGR rate and intake manifold temperature on the BSFC improvement is beyond the scope of this study, and will not be discussed further.

4.3 Simulation results analysis

After the extensive description of the results obtained during the tests on the engine dynamometer, the analysis of the results obtained from the simulations on the cooling system will be presented in the followings. This part represents the key point of the whole project since it joins data obtained from the real engine, about the combustion behavior, and data obtained from simulations, about the cooling system.

If so far the main concern was on the engine and on the combustion stability, now the point of view is moved to a general vision; the cooling system is inserted in the equation to evaluate the overall fuel consumption.

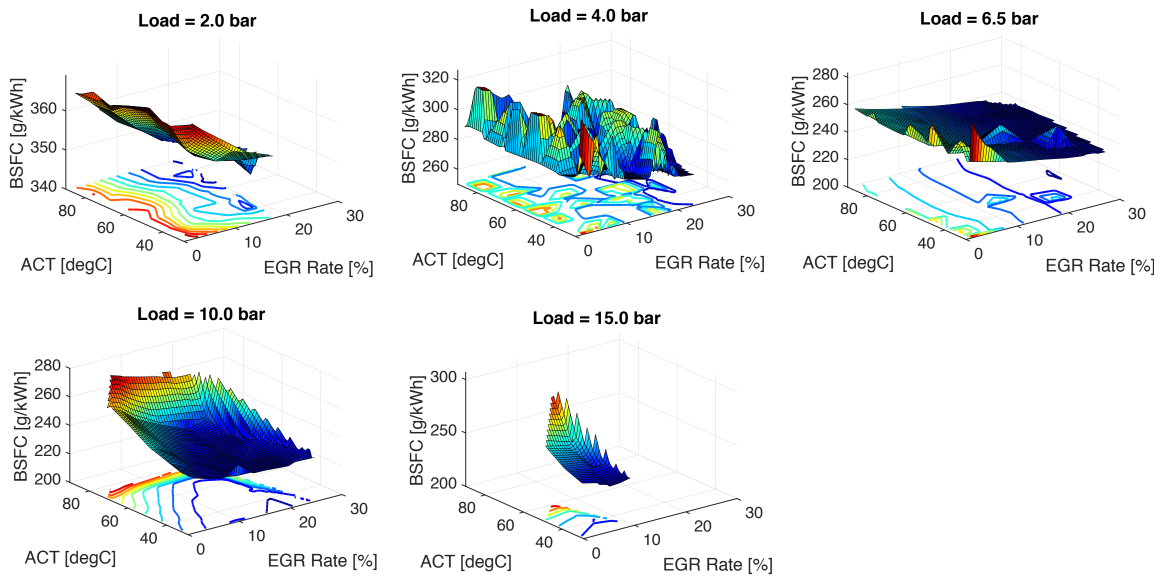
The analysis on the simulation data collected will lead to the identification of the best position of the cooling system actuators to minimize fuel consumption while, at the same time, maintaining combustion stability and engine durability.

Just as a reminder, the procedure followed in this second part could be schematized as:

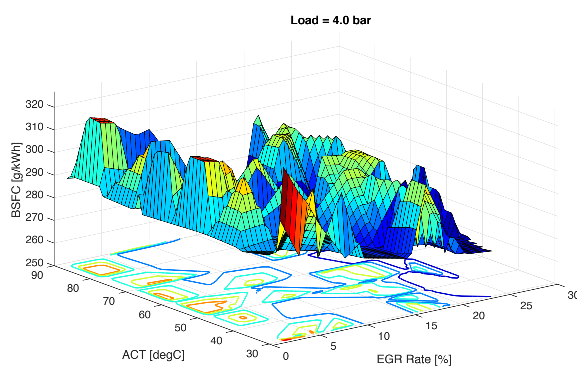
- Preparation of DOE to take into account all different actuators positions
- Simulation of all the positions evaluated using the DOE
- Filtering of the results based on constraints for engine durability
- Analysis of the remaining configurations, after being filtered, to find, among all of them, the one which leads to the lowest fuel consumption.

4.3.1 Interpolation errors

The procedure followed to find the final result is based on the interpolation of data from the first part (engine dyno tests) to obtain the value of BSFC as a function of the input parameters. The main parameter chosen as inputs are: EGR rate, air charge temperature and load condition. In this way the BSFC could be obtain as: $BSFC = f(EGR\ rate, ACT, load)$ and a 3D space is filled with all the data coming from the previous tests. For each set of the three inputs one and only one value of the BSFC is obtained. With this consideration, all the configurations evaluated in the 1D simulations are evaluated, and a value of the BSFC is assigned. From a simple formula it is possible then to retrieve the fuel consumed in both the conditions where the cooling system is considered or not.



(a)



(b) Particular of the 4 bar load

Figure 4.43: *Interpolation of BSFC data at different loads*

The function used to interpolate the data is applying a linear interpolation technique, and the option to not extrapolate data outside the range is chosen as well. With this configuration it is possible to know the value of the BSFC only inside the domain of data collected during the dyno tests. This is a fundamental step since it acts as an additional filtering constraint: all the configurations which are outside the admissible ranges obtained during the tests are not considered.

However, it was found, analyzing the trend of the data at different loads, that the interpolating function is not performing well when considering the data in the whole range of loads.

This effect could be highlighted in Figure 4.43 (a), where it is clearly visible the absurd trend of the 4 bar load, particular in Figure 4.43 (b). These spikes and pits are not absolutely realistic; they are very different from the actual trend, in Appendix A Figure 2, and are generated because of the interpolating technique. In fact, if looking closer in detailed to the loads at 2 and 6.5 bar, it is possible to see a different inclination of the surfaces: the 2 bar is more steep, while the other one is flatter. When interpolating at 4 bar the function is trying to consider values from the three different conditions 2, 4 and 6.5 bar, but in doing so the result is not reliable at all. The 4 bar surface, in fact, is passing through the points collected on the dyno tests, which is correct, but, for the points outside these data (for example 35°C and all the other temperatures different from the tests ones), the values are interpolated considering also the effect of the previous and following surface(load), which are not compatible due to the great difference in the slopes.

This problem is solvable with two different approaches:

- **Proactive:** the number of loads tested on the dyno cell could be increased, in this way the interpolation would be done on more widespread data allowing to simulate more smoothly the change in slope. This clearly would increase the required time for testing as well as the computation time.

- **Reactive:** it was observed that reducing the range over which the data were interpolated the results are representing better the actual BSFC surfaces. For the fuel consumption calculation, in fact, the data were interpolated between the load considered and the following one, for example between 4 and 6.5 bar. It is correct to do this operation because, after analyzing the simulation results, it is found that the increase in the load due to the cooling system is always lower than 1 bar. The results coming from the application of this approach are reported in Appendix A.

Once the interpolation problem was solved, the results obtained are presented in the next sections.

4.3.2 BSFC curves as function of load value

Referring to Figure 4.12, here the same curves are reported. It is important to highlight that these graphs are the result of tests done at 1500 rpm, and, to build them, the points at the different loads are evaluated from the interpolated surfaces. However, the values of the BSFC at the corresponding tests conditions (2, 4, 6.5, 10, and 15 bar) are the correct ones, since are not either interpolated or extrapolated.

Two different ways of representing the $BFSC = f(Load)$ curve are showed in the followings. In Figure 4.44 the graphs are plotted at constant EGR rate, while in Figure 4.45 they are plotted at constant ACT.

Considering Figure 4.44, it is possible to clearly understand the effect of the different ACT on the different loads. What was highlighted from the engine tests is that, for light loads, the effect of the manifold temperature is not very predominant over the effect of EGR rate and so the better compromise is to have high ACT to have the possibility to admit higher EGR rates; for high loads, instead, the effect of different temperature is present and is helping in reducing in a great extent the BSFC, so the best choice, in this cases, would be a lower temperature. This trend is clearly represented in Figure 4.44, in fact, considering, for example the 5% EGR graph, it is possible to see that, for loads lower than 6.5 bar, the BSFC is worsened if a manifold temperature of 30°C (red) is considered, while, for high loads it actually leads to the lowest BSFC value.

Figure 4.44 moreover is showing the different EGR limits and temperature which could be obtained for each loads. For example, considering the graph at 25% EGR, no data are present for loads before 4 bar, meaning that no such amount of exhaust gas is recirculated in these conditions; for the same reason, there are not data for loads greater than 10 bar.

Same considerations could be made for the temperatures, in fact, in all the plots, after 10 bar, no data are present for manifold temperature equal to 80 and 90°C.

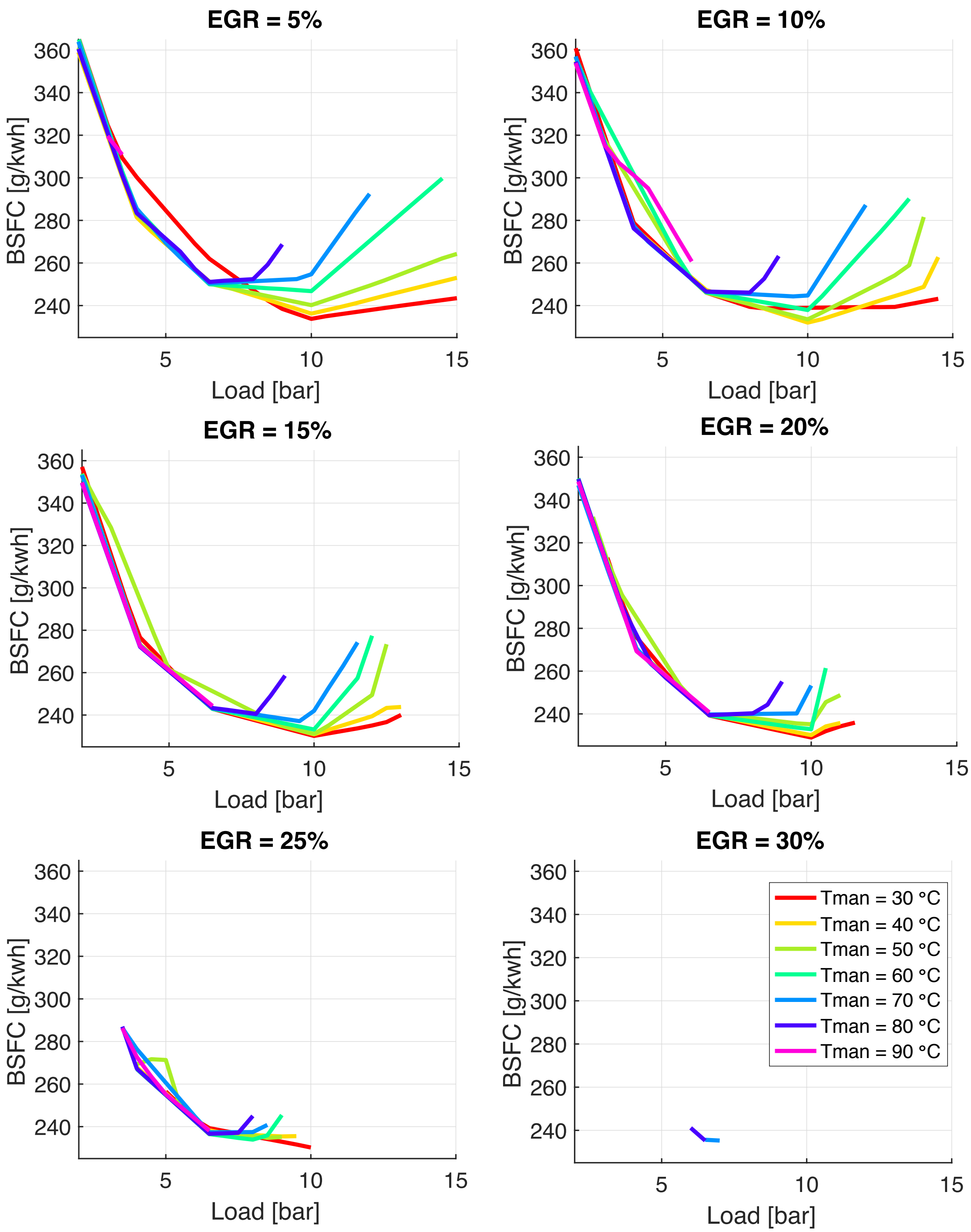


Figure 4.44: *BSFC curves as a function of load at constant EGR rates*

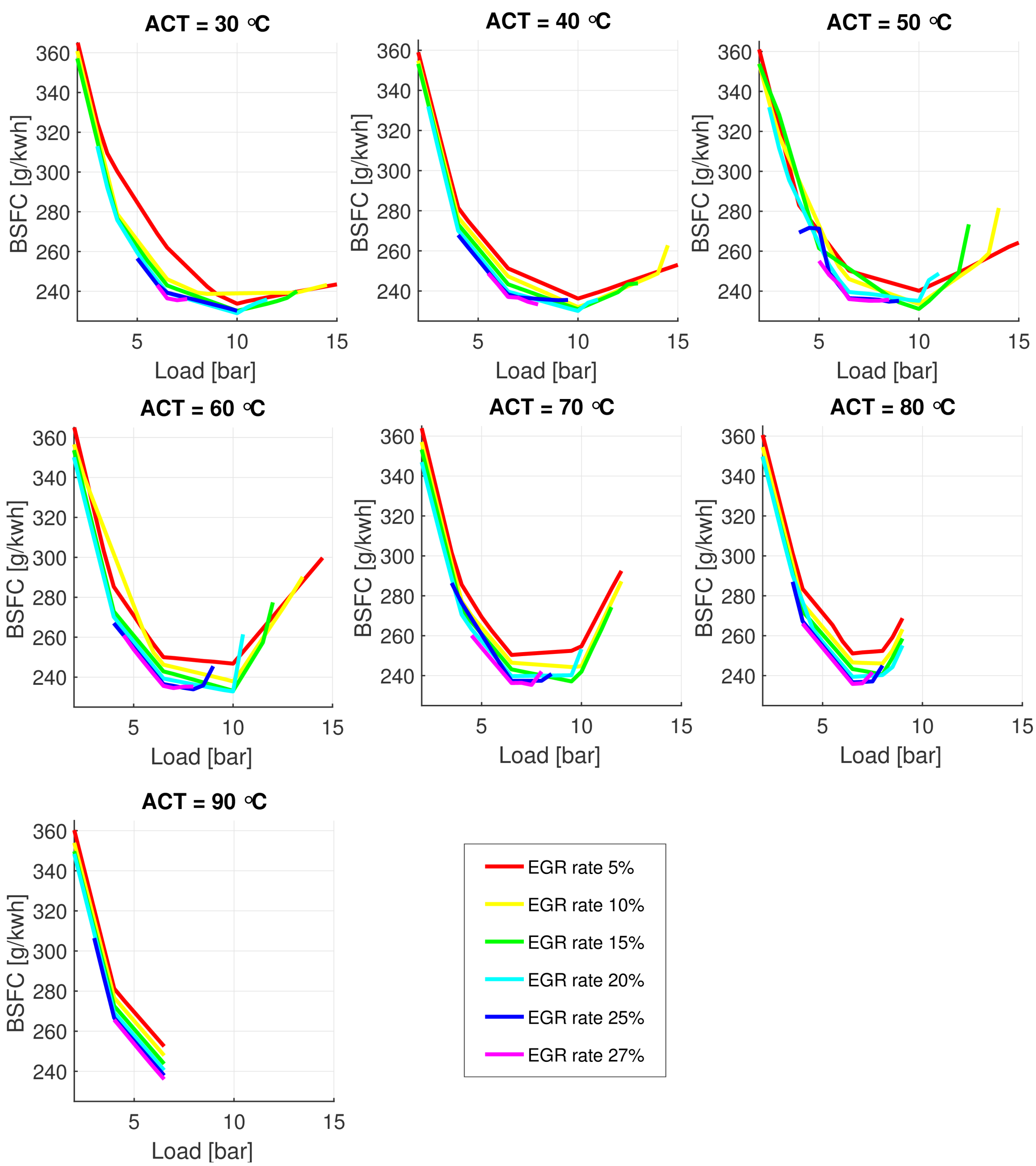


Figure 4.45: *BSFC curves as a function of load at constant manifold temperatures*

Considering Figure 4.45 the same reasoning as above could be done. From this graphs a clearer view on the maximum air charge temperature tolerable is showed. For increasing ACT the lines on the graph are stopped every time at decreasing loads; for the last plot (90°C) data are present only for working points between 2 and 6.5 bar.

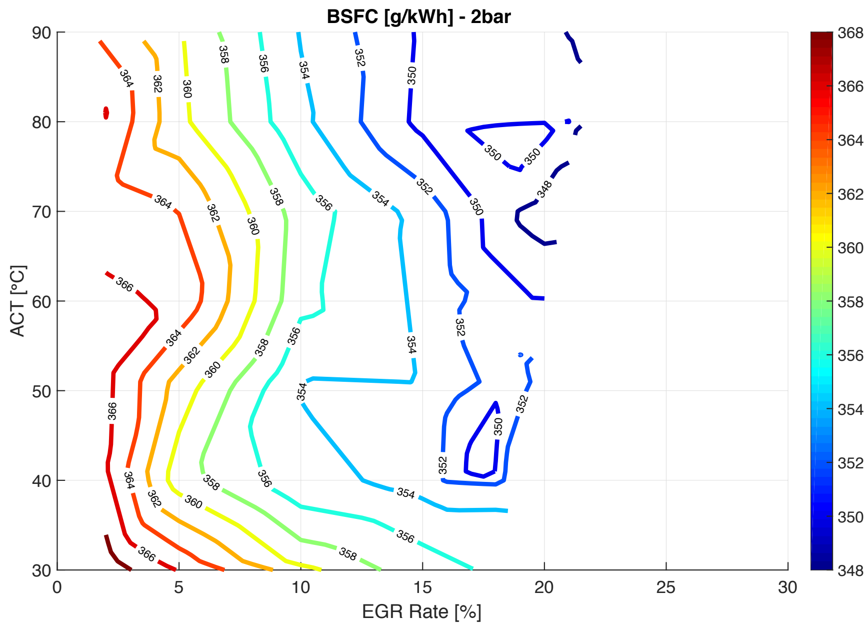
Another clear effect showed in Figure 4.45 is that, for every temperature the lines are almost stacked one over the other, for increasing EGR rates; the 27% line is the one always showing the lowest BSFC values. This trend is particularly interesting since it states that, independently on the ACT, the greatest reduction in BSFC is obtained for the highest EGR rate admissible. Together with this consideration, it should be remembered that, all the data are collected in the limits of a stable combustion process, this assures that, also for the simulation results the actuator positions considered are simulating a possible real life situation.

4.3.3 BSFC contour plot at constant load

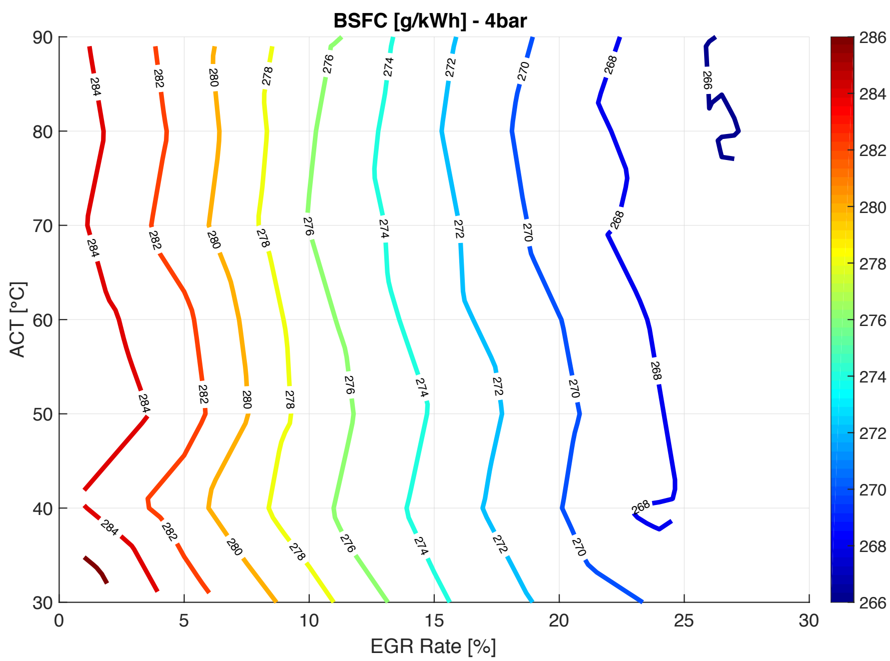
In this section an additional way of representing the BSFC as a function of EGR rate, ACT and load is presented. In these Figures the 3D interpolated space is "sliced" at constant load values and on each plane the effects of EGR rates and ACTs are evaluated.

The first thing that should be noticed is the ranges over which the data are placed. x and y axes are set, for all graphs, with the same limits. In doing so, depending on the load considered, the data will populate the plane accordingly to the maximum EGR admissible at the different temperatures. For low loads the data will occupy a greater part of the plane, while increasing the load, the ranges are reduced. It is fairly interesting that, looking at Figure 4.46 (c) and Figure 4.46 (e), the data are respectively occupying the whole plane or only a temperature range between 30 to 60°C with maximum EGR rate of 10%, this gives a clear explanation of the reduction in EGR rate and manifold temperature changes obtained increasing the load. It is, instead, of great interest the observation of the trend of the iso-BSFC lines as the load is increased. Moving, in fact, from graph (a) to (e) the lines are firstly almost vertical and then the slope is changing to become almost horizontal.

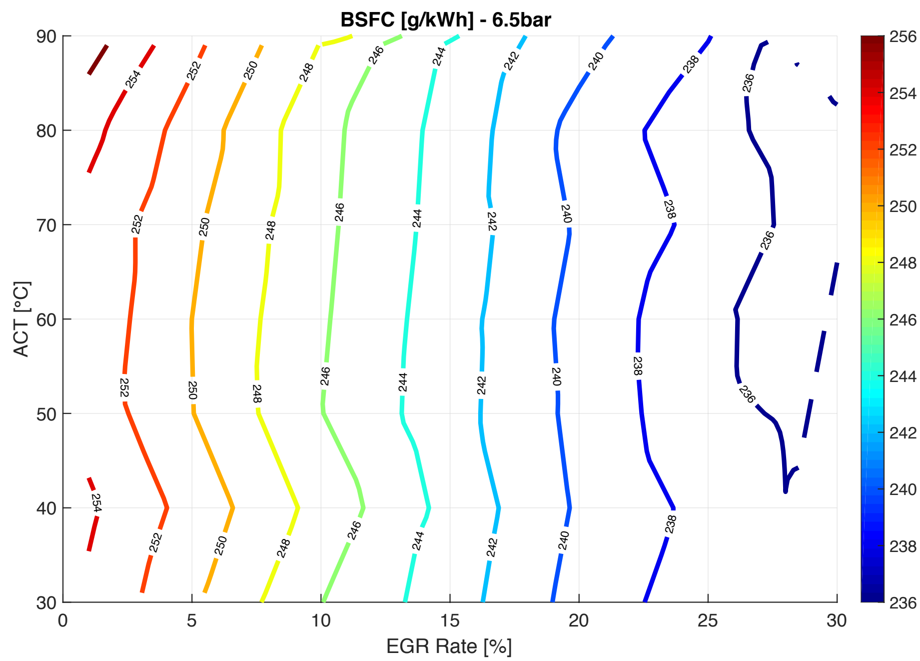
The information coming from this observation are highlighting the effects of the EGR rate and the ACT.



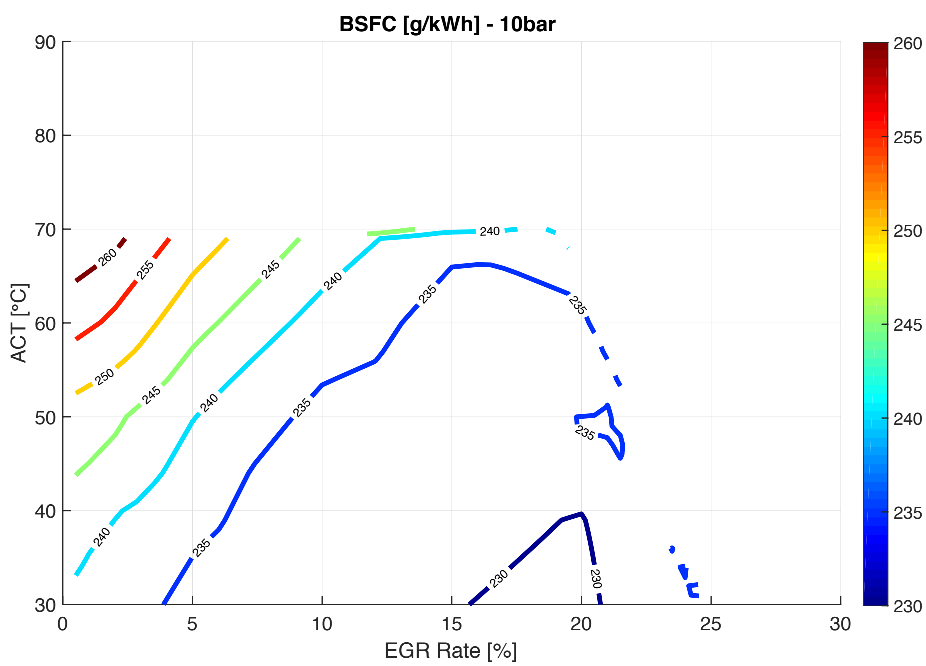
(a) 2 bar load condition



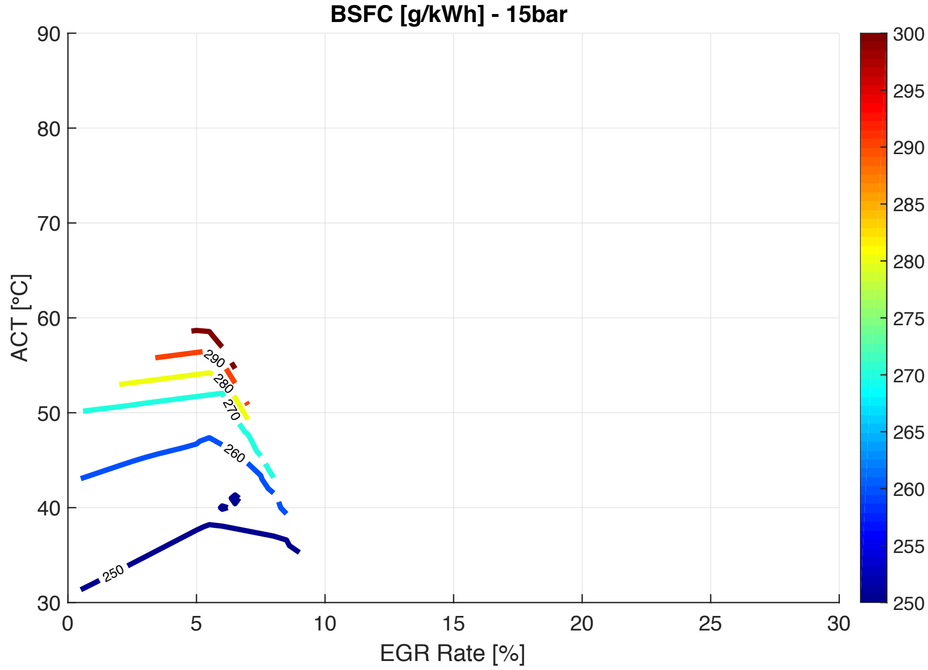
(b) 4 bar load condition



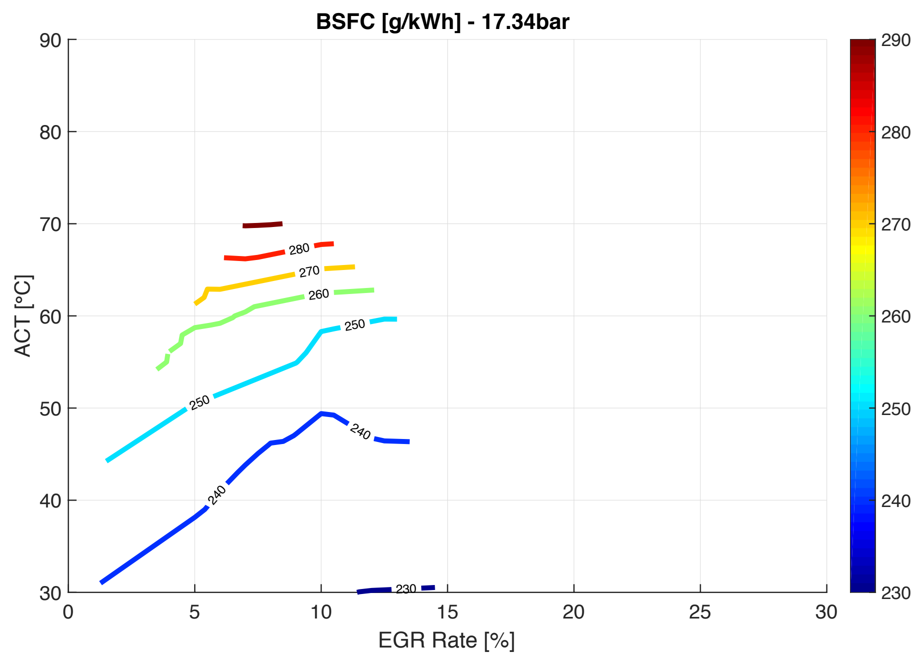
(c) 6.5 bar load condition



(d) 10 bar load condition



(e) 15 bar load condition



(f) 1997 rpm - 17.34 bar load condition

Figure 4.46: *BSFC contour plots at different loads*

For light load, being the lines almost vertical, a small increase in EGR rate permits to reduce the BSFC more than a great decrease in the ACT. For high loads, the lines are almost horizontal, this is translated in a greater effect on the reduction of the BSFC due to lower manifold temperatures rather than variations in EGR rates.

It should also be mentioned that, in Figure 4.46 (a), the data are showing two BSFC islands, this behavior is related to the fact that the BSFC measurements during the engine dyno tests were affected by noise.

4.3.4 Fuel consumption calculations

This section represents the final merge of the results coming from the dyno tests and the 1D heat transfer simulations on the cooling system.

The tests performed on the dyno cell are of paramount importance because of the double function of their results: on one side the BSFC data for wide ranges of EGR rate, ACT and loads have been acquired, on the other side, running the test always considering the combustion stability, has set some constraints on the maximum EGR range and manifold temperature.

The fuel consumption is calculated thus considering this data together with the simulations results. Many actuator positions are analyzed through the DOE technique. Varying these set of positions it was possible to receive as output the additional torque required by the cooling system to acieve different EGR rate and ACT. A very important point to understand the results is that, during the simulations, the cooling system is adjusting itself the manifold temperature depending on the position set for the actuators. The ACT is not forced anymore to predetermined values, but it is obtained, depending on the power drawn by the cooling system. This is translated in the fact that from this point on, when considering ACT values, they are very often different from the previously fixed ones: 30, 40, 50, 70, 80, 90°C.

Since the final goal of the project is to reduce fuel consumption always maintaining combustion stability and engine durability, it is clear the importance covered by the aforementioned results.

After filtering the simulation results depending on the constraints to assure the desired engine durability, and sorting them in ascending order of the total power required to cool (calculated from the simulation results); the first step followed for the final calculation of the fuel consumed is to build a BFSC map for the new load conditions.

The cooling system, in fact, is adding an additional load to the engine, so, to maintain the same power output, the torque developed by the engine itself have to be increased. Considering the previous results, depending on the load condition analyzed, this additional torque will decrease the BSFC, if light loads are considered, or increase it, if dealing with high loads. Recalling that $BSFC = f(EGR\ rate, ACT, load)$ new maps are, at this point, generated for the increased torque points.

The correct procedure to be followed, after generating these maps, is then no more to look at the BSFC values, but to convert all the data in fuel consumption; the unit of measure used for these results is chosen to be [g/s]. The formulas used are the following ones:

$$FC_{Baseline} = \frac{Torque_{Baseline} \cdot rpm \cdot 2\pi}{60 \cdot 1000} \cdot \frac{BSFC(EGR, ACT, Torque_{Baseline})}{3600} \tag{4.3.1}$$

$$FC_{Corrected} = \frac{Torque_{Corrected} \cdot rpm \cdot 2\pi}{60 \cdot 1000} \cdot \frac{BSFC(EGR, ACT, Torque_{Corrected})}{3600} \tag{4.3.2}$$

- $FC_{Baseline}$ is the fuel consumption calculated when the cooling system is not considered; it is obtained directly from the engine dyno test. The only difference is that in this case the EGR rate and ACT are values directly obtained from the simulations, meaning that they could be whichever point inside the ranges indicated in Figure 4.46. It is measured in [g/s].

- Both $Torque_{Baseline}$ and $Torque_{Corrected}$ are expressed in Nm. The first one is directly retrieved from the load condition considered and it is equal to the one set during the dyno tests. The second one instead is the one obtained from the 1D simulations and, for that reason, the following relationship is always verified: $Torque_{Corrected} > Torque_{Baseline}$

- rpm is the engine speed, except for the 17.34 bar load, where the engine speed was set to 1997 rpm, its value is always 1500 rpm.

- The BSFC values, in both equations, are measured in [g/kWh]. The main difference between the two is the third input: the load condition. In fact, it is determining whether an increase or a decrease in the BSFC value is achieved. EGR rate and ACT are obtained from the simulations and are the same for the baseline or corrected fuel consumption calculations, because what is of major importance is the difference in the torque due to the additional resistance of cooling system components.

One fundamental concept that should be always kept in mind is that the BSFC values for $Torque_{Baseline}$ are placed on the surfaces interpolated from engine dyno tests; the BSFC values for $Torque_{Corrected}$ are obtained through interpolation, between $load_n$ and $load_{n+1}$, and consequently they are placed on different surfaces.

From the application of the aforementioned formulas the fuel consumed for the different set of actuators position is calculated and fuel consumption surfaces are built for every load considered. As will be showed in the next pages, the surfaces representing the baseline fuel consumption are simply the same BSFC presented in Appendix A, but scaled by a $factor = \frac{Torque_{Baseline} \cdot rpm \cdot 2\pi}{60 \cdot 1000 \cdot 3600}$, while the ones representing the corrected fuel consumption are more complex since they depend both on $Torque_{Corrected}$ and on $BSFC_{Corrected} = f(EGR, ACT, Torque_{Corrected})$.

To facilitate the understanding of the results presented in the followings, could be important to stress again on the difference between $Fuel\ consumption_{Baseline}$ and $Fuel\ consumption_{Corrected}$. The first is calculated when no cooling system is attached on the engine; same configuration of the dyno cell. The second is related to the simulations and to the added resistance of the cooling system components. When comparing the two clearly the Baseline will be always lower than the corrected one.

Before going further in the analysis of the results it is necessary to recall that during the simulations also the vehicle speed was considered, and the values of this parameter were extrapolated from analysis on the FTP cycles: city and highway. A summary table is reported in Table 4.4; where only one speed value is present it means that the vehicle speed during highway and city traveling, at that engine load, is considered to be the same.

Table 4.4: *Vehicle speed for each load and FTP cycle considered*

<table>
<tr><th></th><th>2 bar</th><th>4 bar</th><th>6.5 bar</th><th>10 bar</th><th>15 bar</th><th>17 bar</th></tr>
<tr><td>City</td><td>25 mph</td><td>27 mph</td><td>45 mph</td><td rowspan="2">54 mph</td><td rowspan="2">52 mph</td><td rowspan="2">45 mph</td></tr>
<tr><td>Highway</td><td>45 mph</td><td>52 mph</td><td>54 mph</td></tr>
</table>

4.3.5 Fuel consumption at load = 2 bar

The analysis will be divided in two parts, one related to the city and the other to the highway.

City

In Figure 4.47 the trend of the fuel consumption over different EGR rates and ACT is reported. The green surface represents the fuel consumed in the baseline condition, where the effect of the cooling circuit is not considered. This surface, as already mentioned, is obtained directly from the data collected on the dyno tests and it is similar to Figure 1 in Appendix A, but scaled.

The red curve, instead, is the one calculated considering the torque corrected to take into account the additional load due to the cooling system. It is, as previously stated, obtained from the corrected torque and the BSFC associated to it. It shows some peaks between the manifold temperatures of 40-45°C and 60-65°C; these temperature should then be avoided because the final fuel consumed would be fairly high.

Figure 4.48 is particularly interesting because is showing many information all in a glance. It is worth to spend some time in describing this graph.

The first thing to consider is that the data used in this plot are sorted in ascending order, from the lowest to the highest power required by the cooling system; this is also noticed looking at the general trend of the blue curve. The cooling system power is not reported in the graph only to help the reading. The values to look at are the squares or the dots, the line between them are just added to help the reading since they could help

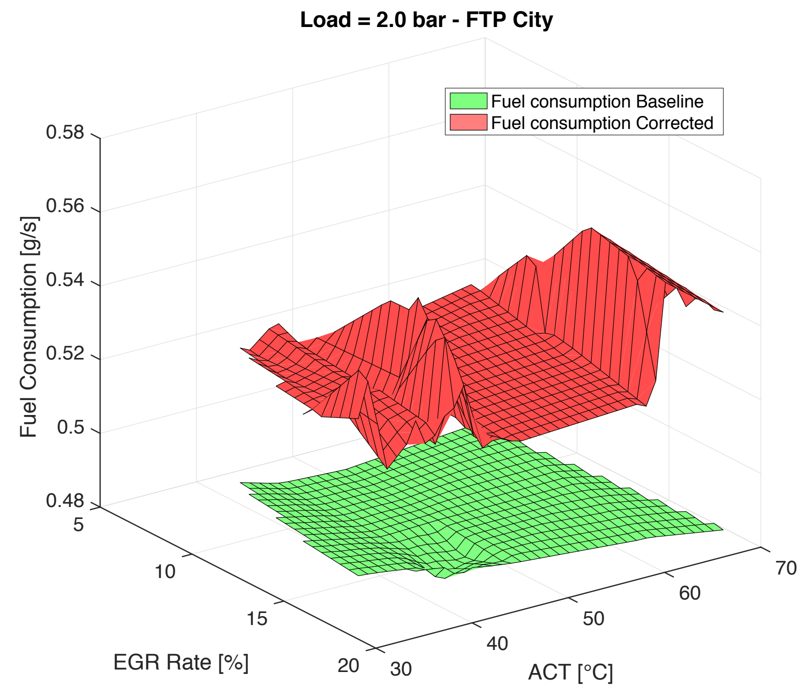


Figure 4.47: *Fuel consumption surfaces - 2 bar City*

in distinguish the different conditions. It must be clear that the graph is showing discrete points and it is wrong to read results on the lines.

The red curve represents the baseline fuel consumption in [g/s] and it ranges between 0.483 and 0.496 g/s. On the x axis the number of the configuration leading to the particular value of fuel consumption is reported and, looking in closer detail, the lowest fuel consumption when the cooling system is not considered is found for the configuration number 67. Following the reasoning that, for the baseline configuration the fuel consumption is equal to the BSFC surface but scaled, the minimum fuel consumption point is found to be also the lowest BSFC value.

Without the study made in this project, configuration 67 would have been chosen as the preferred working point to minimize the fuel consumption, however, as will be explained, this approach is wrong when the cooling system is considered, even though sometimes both the minimum coincide.

The blue curve represents the percentage increase in fuel consumption in respect to the baseline when the cooling system is considered. It is calculated as:

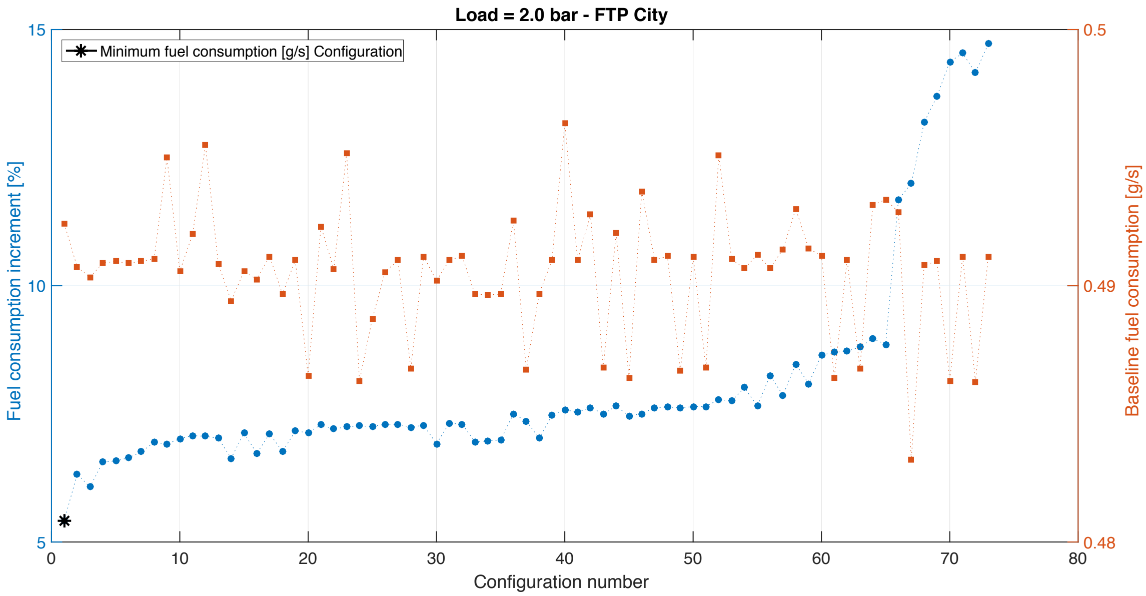


Figure 4.48: *Percentage increment of fuel consumption (blue) in respect to the baseline configurations (red) - 2 bar City*

$$Percentage\ increment = \frac{FC_{Corrected} - FC_{Baseline}}{FC_{Baseline}} \cdot 100 \qquad (4.3.3)$$

From the baseline fuel consumption and its percentage increase the corrected fuel consumption could be easily evaluated, so, in a certain way, in Figure 4.48, even if not directly, also this result is plotted; it is not showed on the same graph not to burden the reading. The range over which the percentage is varied is between 5 to 15%. On this curve it is also highlighted the best configuration for what concern the fuel consumption when the cooling system is considered. In this particular case it is found to be the configuration 1 (black cross in the graph), which is also the one with the lowest percentage increase. It should be observed, however, that, looking directly at the baseline fuel consumption it is not at its minimum, but actually is above the average. This is particularly interesting because the best configuration is not found when the baseline BSFC is at its minimum but when both the baseline fuel consumption and the percentage increment are at low values. When observing other results these considerations will be clearer.

In Figure 4.49 fuel consumption contour graphs are plotted. It should be noticed that

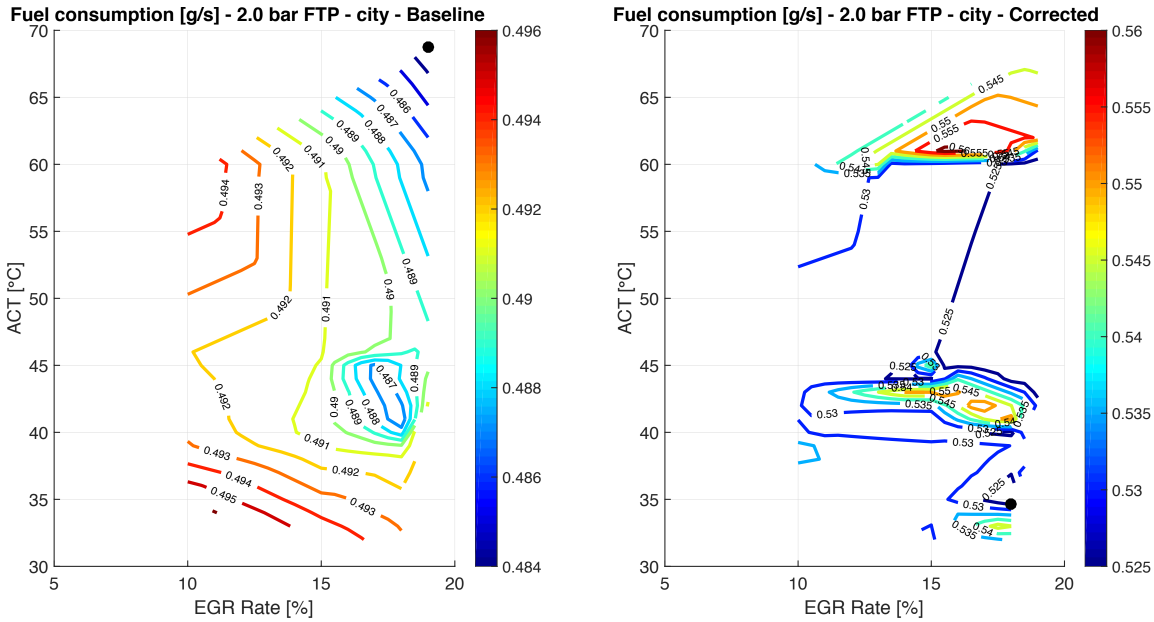


Figure 4.49: *Best configurations to reduce fuel consumption: baseline on the left and corrected on the right - 2 bar City*

the graph on the left, representing the fuel consumption of the baseline configuration, is simply the same of Figure 4.46 (a) but scaled and limited to a smaller region; the EGR values range from 5 to 20% and the y axis between 30 and 70°C. This difference is the result of the implementation of the cooling system, in fact only the temperatures and EGR rates admissible when the cooling system is simulated are considered. On this graph, with a black dot, it is highlighted the best fuel consumption point for the baseline configuration. Then it is increased (considering the percentage increase of this configuration) to obtain the Non-optimized point and its main values are present in Table 4.5.

The plot on the right is representing the fuel consumption region when the cooling system is considered. Recalling the red surface in Figure 4.47 it is possible to notice here that two high fuel consumption islands are present, these are representing the two peaks on the surface. Also for this condition the black dot is indicating the best fuel consumption point, and the first thing that is showed is that the temperature is almost halfed. This result is particularly interesting because, when the cooling system considered, the best fuel consumption point position is completely opposite in respect to the initial belief coming from engine dyno tests that at low loads it is better to have a high manifold temperature. The results for this configuration are presented in Table 4.5.

Table 4.5: *Fuel consumption results for baseline and corrected configurations - 2 bar City*

2 bar City	**ACT** [°C]	**EGR** [%]	**Cooling Power** [W]	**Fuel consumption** [g/s]
Non-optimized	68.71	19	818.27	0.541
Optimized	34.66	18	360.03	0.519
		FC reduction [%]	4.07%	

The fuel consumption reduction in percentage is calculated considering how the previous studies were made in respect to the new approach. The *Non-optimized row* is representing the fuel consumed if the old method for the analysis is used. In fact, before this project, the procedure followed was looking at the lowest BSFC baseline value, consider the actuators configuration which leads to that point, calculate the additional power required by the cooling system and finally the fuel consumed (in this case the Non-optimized one). The new approach is no more looking at the lowest baseline BSFC, but at the lowest corrected BSFC. In doing so the values for the fuel consumption are slightly different, there is only one case where they are exactly the same point: 4 bar highway. This is a further explanation of the reasons behind having chosen, for this load, configuration 1 rather than configuration 67. The final reduction in fuel consumption is then found to be around 4.07%, obtained only changing the working point in a possible range favorable both for the cooling system (low power required) and both for the engine (combustion stability and engine durability are respected). The important difference that should always kept in mind is that the Baseline fuel consumption is not considering the cooling system connected to the engine. When this system is considered, the fuel consumption is consequently increased and the Non-optimized value is found. The Non-optimized value is, in fact, taking into account the cooling system and it is the results that would have been obtained from the old procedure.

Highway

The analysis for the highway test is identical to the one conducted for the city. Similar data charts will be presented and the attention will be focused on the particularly interesting results.

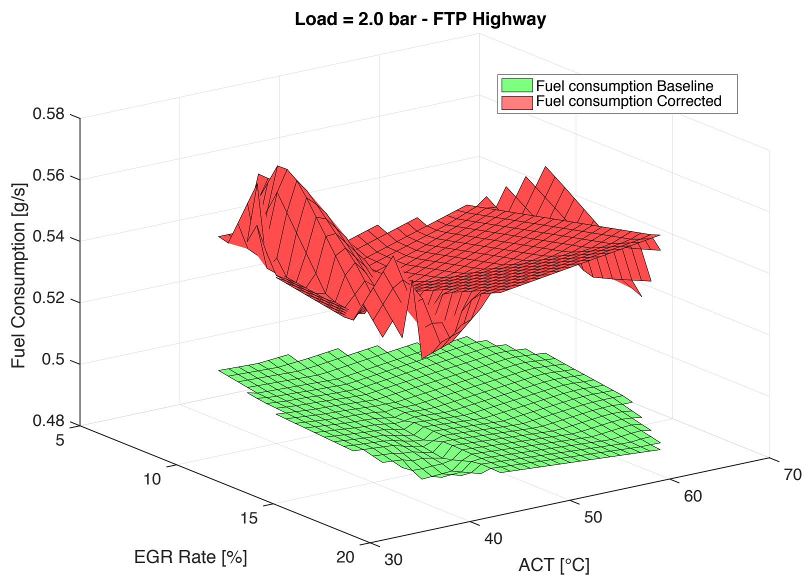


Figure 4.50: *Fuel consumption surfaces - 2 bar Highway*

In Figure 4.50 the fuel consumption surfaces are presented, the reasoning and the color codes are the same applied above.

Figure 4.51 is showing in a better way the idea behind this project. In this case, differently than Figure 4.48, the best configuration for the lowest corrected fuel consumption is found for configuration 7, which does not show neither the minimum value of the baseline fuel consumption (and consequently of baseline BSFC) or the minimum percentage increment. This fact is of particular interest since it is proving that is not only the power required by the cooling system to influence the fuel consumption, but a combined effect of that power together with the baseline fuel consumption, meaning that the engine and the cooling system has to be analyzed as a unique system and not in a separate way.

It should be noticed that the number of configurations analyzed for this load condition are less than half of the previous ones. This should be referred to the fact that the 1D model is having some difficulties to reach the steady state. The possible reasons are related to the

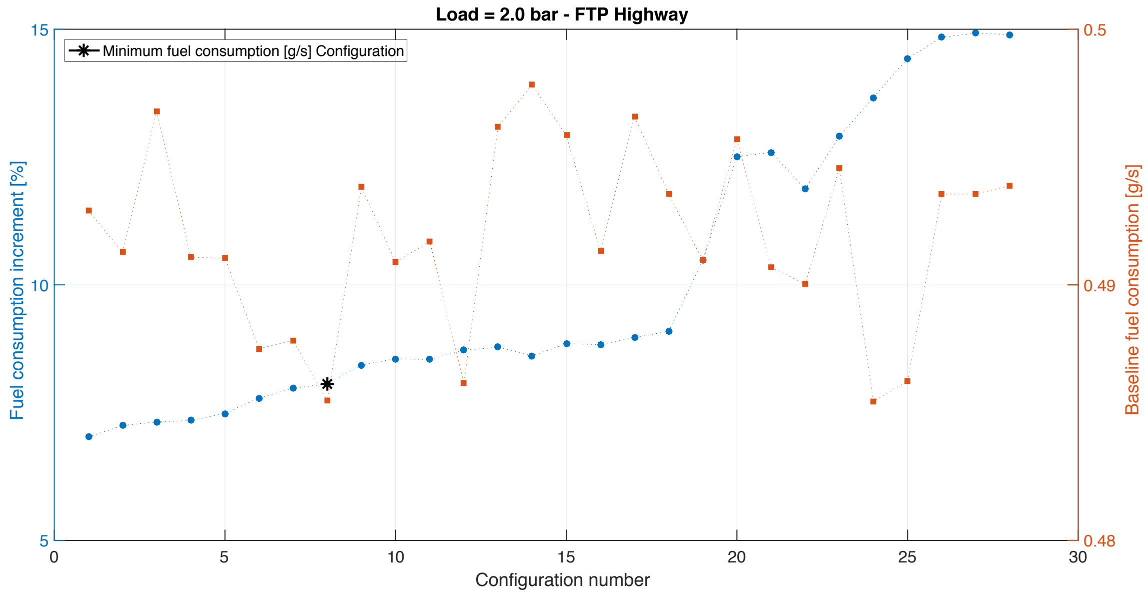


Figure 4.51: Percentage increment of fuel consumption (blue) in respect to the baseline configurations (red) - 2 bar Highway

low amount of coolant flow in conjunction with a high air flow rate due to the high vehicle speed and it represents one of the limitations of the steady state analysis over a transient one. The model is probably tuned better for higher loads, where the behavior will be better, but despite this fact good results has been found also for the light load conditions.

In Figure 4.52 the fuel consumption contour maps are reported for the baseline and corrected configurations. The black dots are highlighting that the EGR rate and ACT to minimize the fuel consumption almost coincide, as it is reported also in Table 4.6. With this consideration no or very small fuel reduction should be expected, but actually the fuel consumption reduction is around 4.9%. Recalling that this evaluation is obtained comparing the previous method with the new one, this great reduction could be explained considering that the cooling power of the two points is different, because not only the actuators affecting the EGR rate and the ACT are considered, but the system itself as a whole.

As a general reasoning, looking at the corrected fuel consumption contour it is possible to individuate different low value islands, this could be a good region to reduce the fuel consumption, but the best position is still the one highlighted by the black dot.

Figure 4.53 is showing the baseline BSFC map over which the different points for the

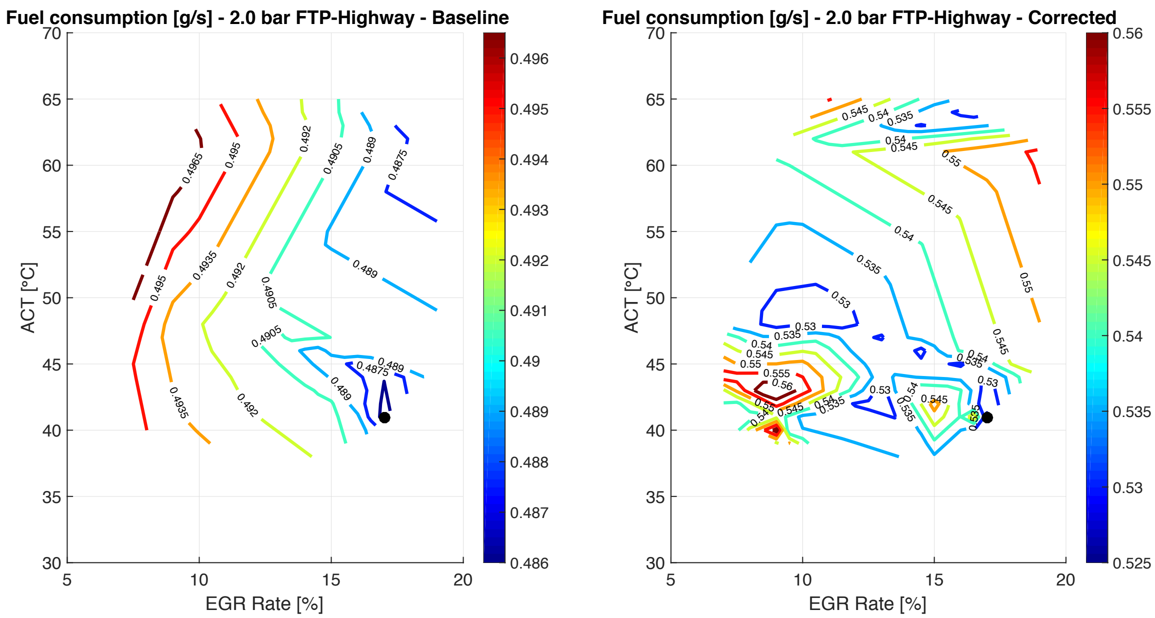


Figure 4.52: Best configurations to reduce fuel consumption: baseline on the left and corrected on the right - 2 bar Highway

Table 4.6: Fuel consumption results for baseline and corrected configurations - 2 bar Highway

2 bar Higway	ACT [°C]	EGR [%]	Cooling Power [W]	Fuel consumption [g/s]
Non-optimized	40.80	17	939.35	0.552
Optmized	40.97	17	540.69	0.525
			FC reduction [%]	4.93%

lowest fuel consumption are placed. Considering that 2 bar is placed in the descending part of the BSFC/load curve, the corrected points are showing a lower BSFC value than the baseline ones, and this is reported also looking at Figure 4.53(the perspective is impairing the vision).

4.3.6 Fuel consumption at load = 4 bar

The analysis for the 4 bar load is similar to the one at 2 bar, the only exception is found in the values for the vehicle speeds, both in city and highway tests, as reported in Table 4.4.

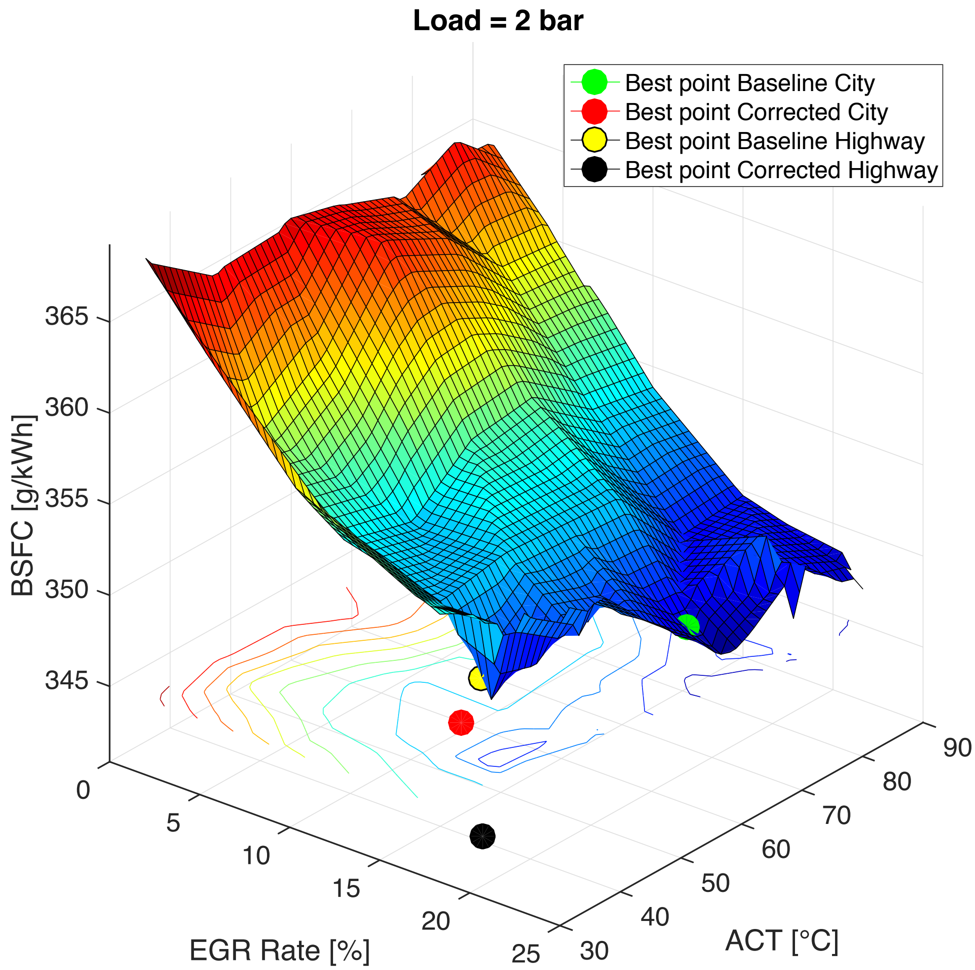


Figure 4.53: *Position of the best actuators configurations over the baseline BSFC surface - 2 bar*

City

The analysis performed on fuel consumption surfaces, shown in Figure 4.54, is identical to the previous ones; nothing particular should be highlighted.

Also for what concern the baseline fuel consumption and its percentage increment, Figure 4.55, the results are similar to the previous cases. Also in this case it has not been chosen the lowest baseline BSFC point, but one point with both low values of percentage increase and baseline BSFC. The configuration which leads to the lowest corrected fuel consumption is the number 7, while the one with the lowest baseline one is the number 23. As it is reported in Table 4.7, the fuel reduction obtained, analyzing the results, is found

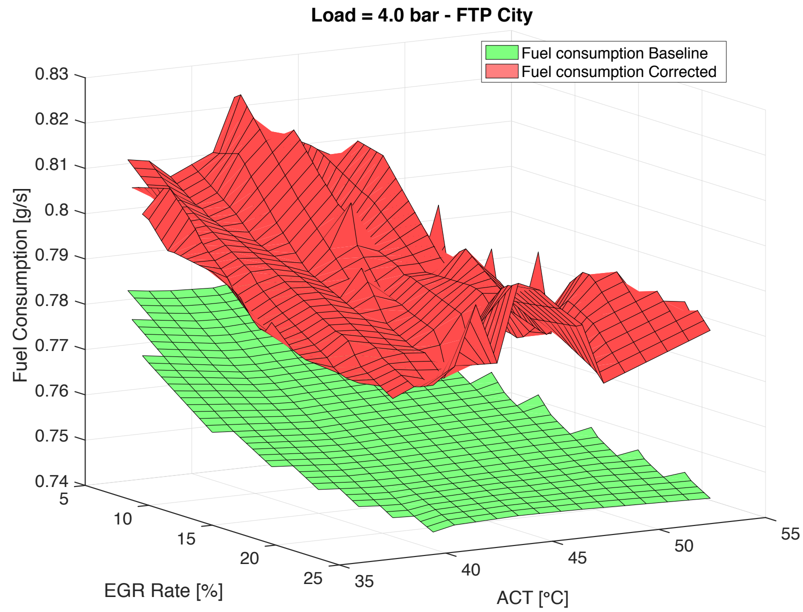


Figure 4.54: *Fuel consumption surfaces - 4 bar City*

to be around 0.33%.

In Figure 4.56 the fuel consumption maps are reported; as expected, the baseline one (on the left) has the same trend as Figure 4.46(b) and it is clearly showed the major effect of the EGR rate over the ACT to reduce the fuel consumed. The corrected one, as expected from Figure 4.54, is showing many different island at different values, this is the results of the spiky trend of the fuel consumption surface.

The black dots are in line with the aforementioned reasoning on the effect of the EGR rate, in fact are both placed on the right part of the graphs. They differ in the temperature values, because increasing the manifold temperature, the cooling system is let to work at high temperatures and consequently the load is reduced.

Table 4.7: *Fuel consumption results for baseline and corrected configurations - 4 bar City*

4 bar City	ACT [°C]	EGR [%]	Cooling Power [W]	Fuel consumption [g/s]
Non-optimized	39.70	24	444.24	0.770
Optimized	41.62	24	379.28	0.768
		FC reduction [%]	0.33%	

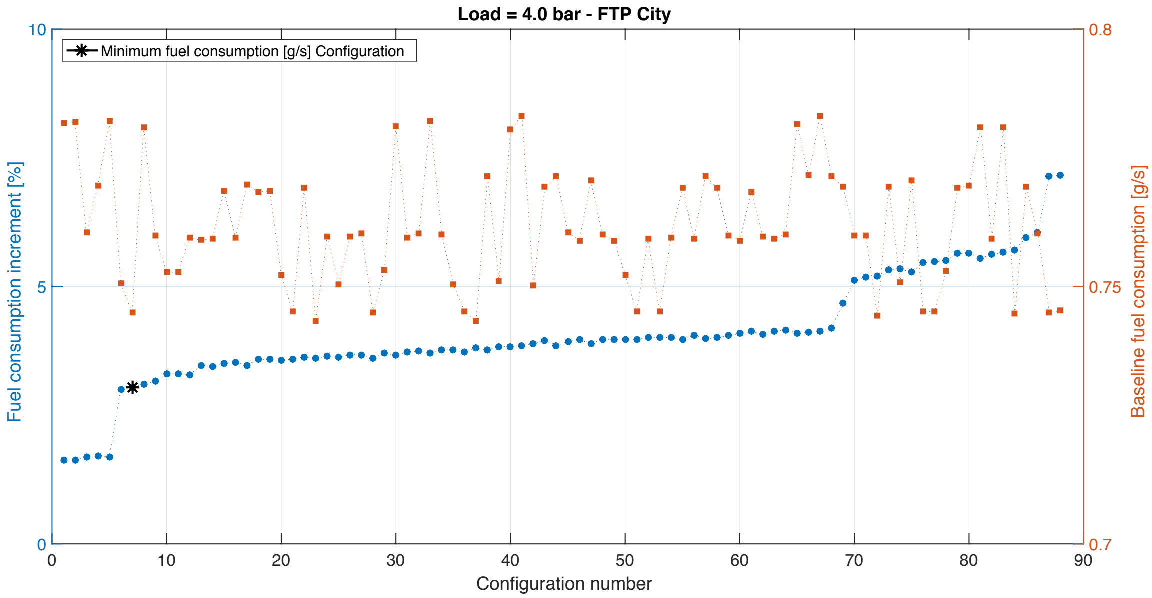


Figure 4.55: *Percentage increment of fuel consumption (blue) in respect to the baseline configurations (red) - 4 bar*

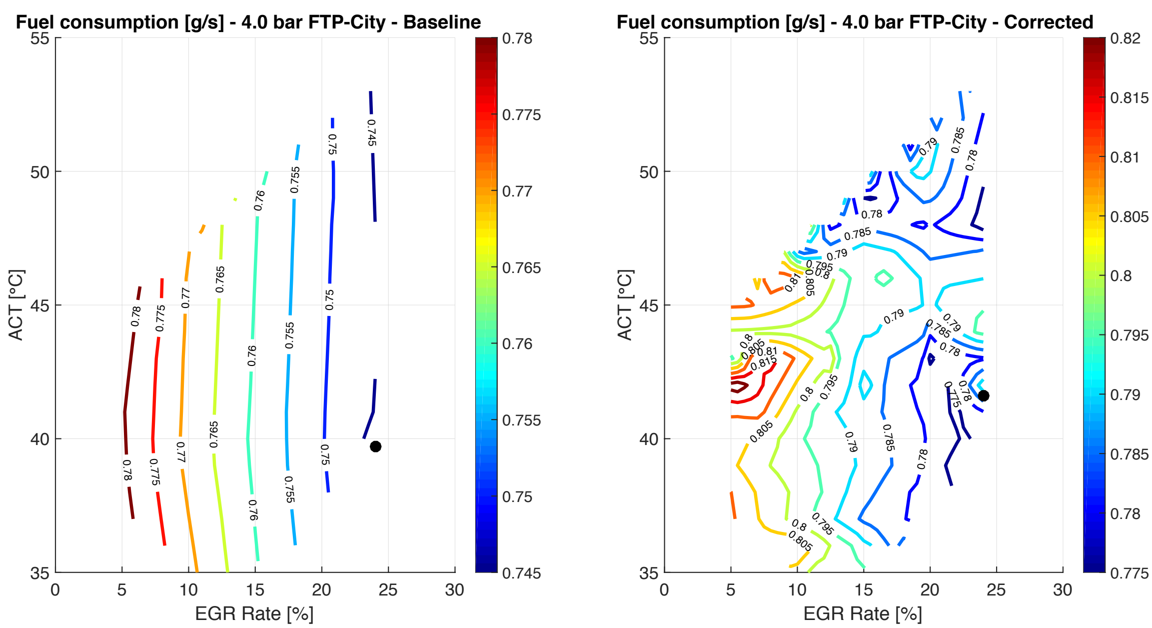


Figure 4.56: *Best configurations to reduce fuel consumption: baseline on the left and corrected on the right - 4 bar City*

In Table 4.7 the values of the EGR rate, ACT, cooling power and fuel consumption are reported for the two configurations. As it is already stated, the fuel consumption reduction is low compared to the 2 bar cases. This result is expected if the variation of cooling power is considered: the two values are very close one to the other, while before the differences were fairly high.

Highway

The fuel consumption surfaces obtained, Figure 4.57, for this working point are in line with what expected and nothing particular should be mentioned.

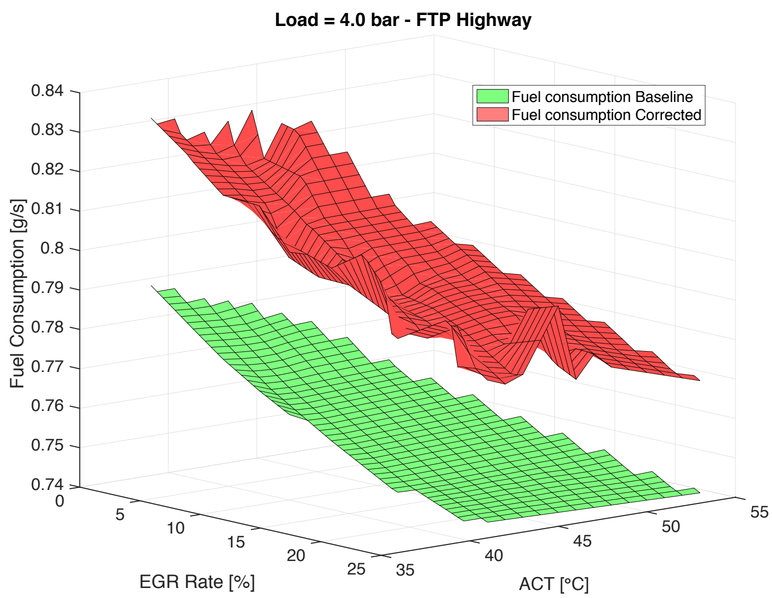


Figure 4.57: *Fuel consumption surfaces - 4 bar Highway*

What should be mentioned instead is that, for this set of load and vehicle speed, the lowest fuel consumption for the baseline and corrected configuration are coincident, meaning that no real decrease in fuel consumption would have been found using the approach developed in the project.

Looking at Figure 4.58 it is important to highlight that the lowest fuel consumption is not found for the configuration 1, which shows the lowest cooling power value, but it is found for the configuration 2, which has a slightly higher power value. This additionally confirms what explained so far about the cooling power: it is not the parameter to be minimized, but its interaction with the baseline fuel consumption.

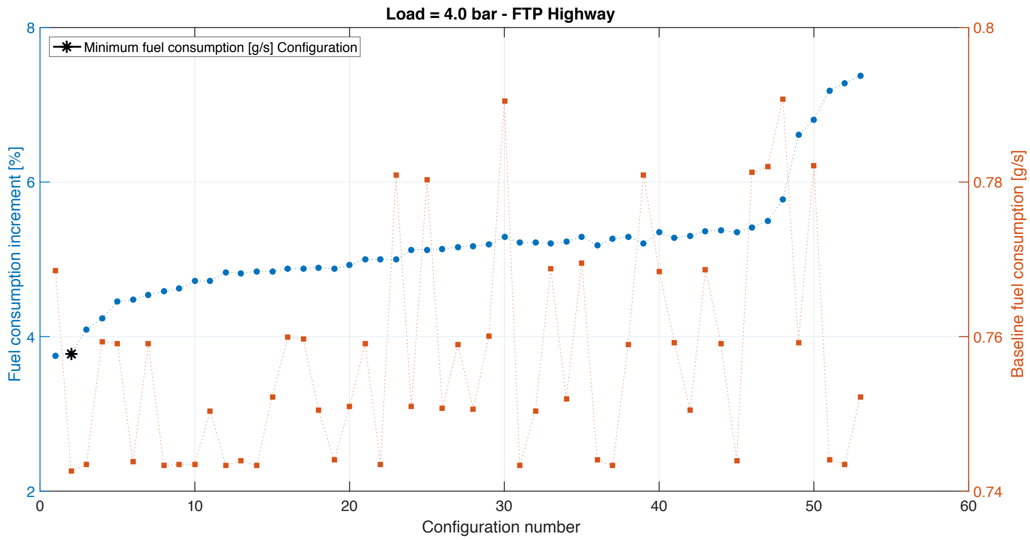


Figure 4.58: Percentage increment of fuel consumption (blue) in respect to the baseline configurations (red) - 4 bar Highway

In Figure 4.59 it is possible to notice that also the corrected fuel consumption is starting to show a trend similar to the baseline one, this is because the red surface in Figure 4.57 is more flat and smooth in respect to the previous cases. The black dot on both graphs is placed in the same position, this is confirmed also by Table 4.8, where only one line of results is present.

Table 4.8: Fuel consumption results for baseline and corrected configurations - 4 bar Highway

4 bar Higway	ACT [°C]	EGR [%]	Cooling Power [W]	Fuel consumption [g/s]
	53.30	25	464.83	0.771
		FC reduction [%]	0%	

To conclude, Figure 4.60, is showing, over the 4 bar BSFC curve, the different best point discovered. In accordance with what was found in the previous sections, when the cooling system is considered, since it adds additional load, the BSFC values for these configurations are lowered in respect to the baseline. It should be noticed also that the green and yellow dots are placed on the surface, while the red and black ones are respectively beneath them, this is a consequence of having the same EGR rate and ACT but at a higher load.

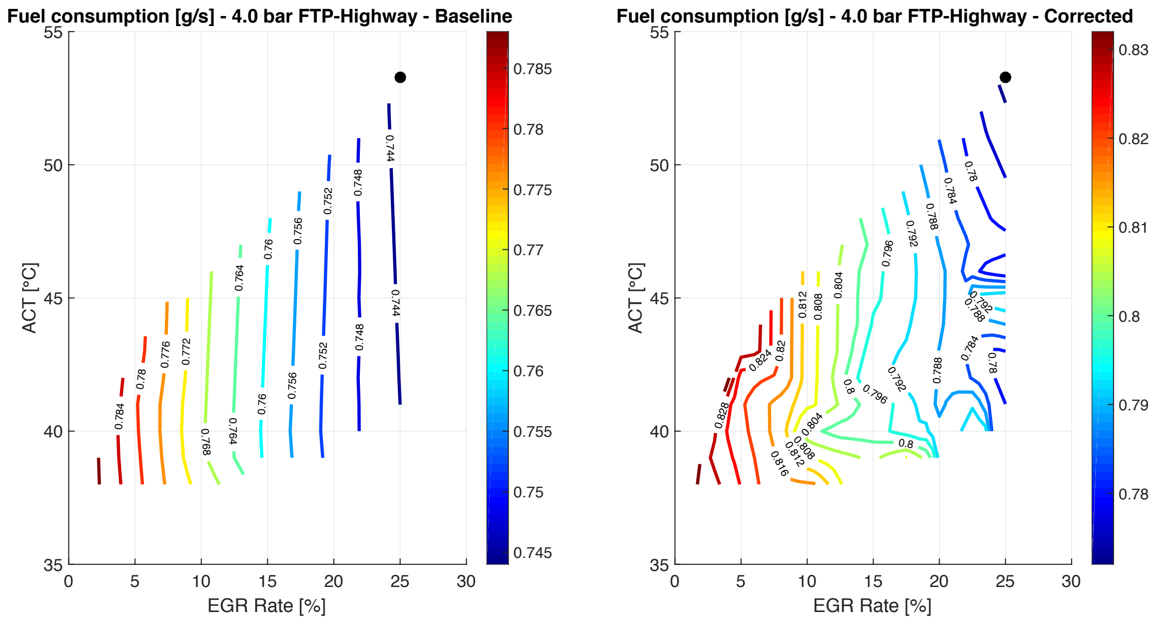


Figure 4.59: *Best configurations to reduce fuel consumption: baseline on the left and corrected on the right - 4 bar Highway*

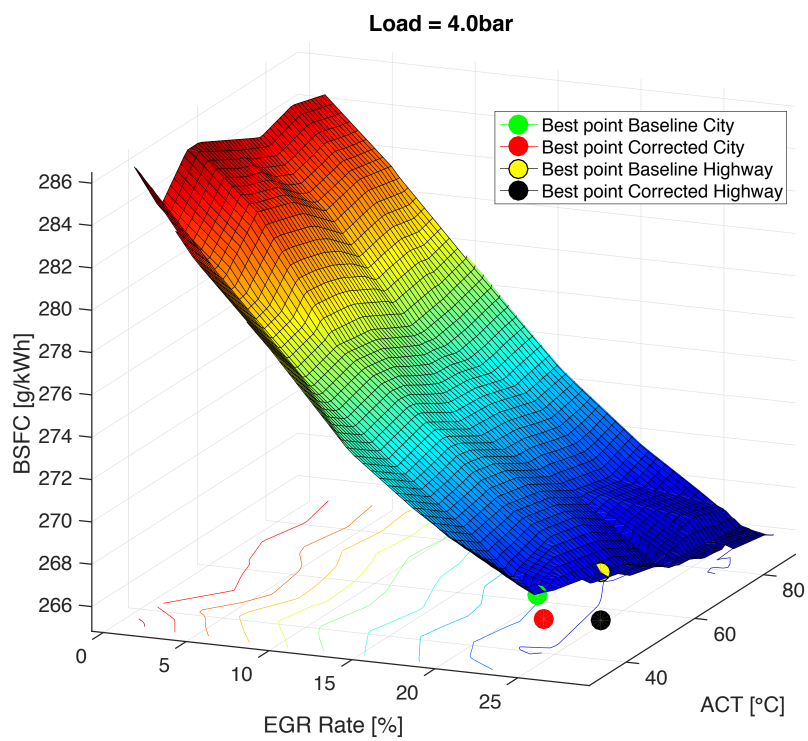


Figure 4.60: *Position of the best actuators configurations over the baseline BSFC surface - 4 bar*

4.3.7 Fuel consumption at load = 6.5 bar

A complete explanation on how to read the graphs and on their trends has already been provided. From this point on only the results worth to mention will be highlighted.

City

Looking at the surfaces shown in Appendix A, at higher loads the surfaces tends to be more smooth and this is reflected also in Figure 4.61.

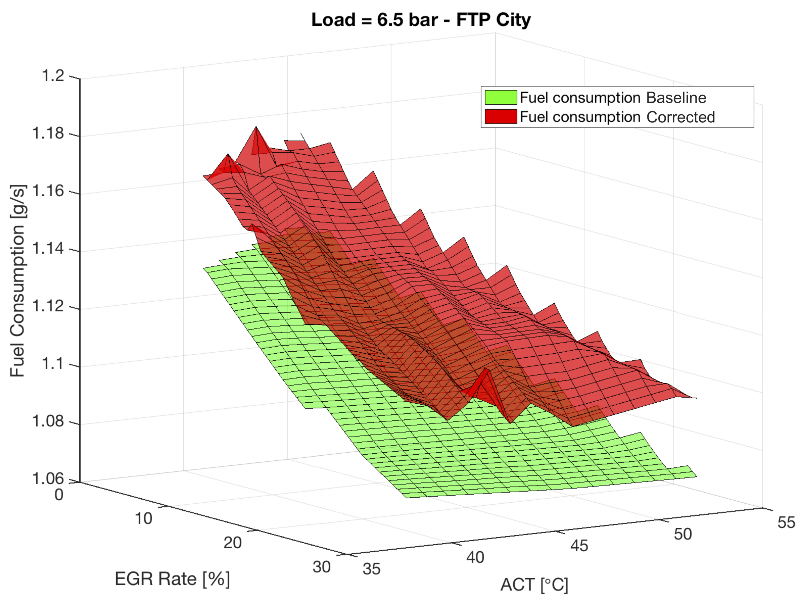


Figure 4.61: *Fuel consumption surfaces - 6.5 bar City*

Table 4.9: *Fuel consumption results for baseline and corrected configurations - 6.5 bar City*

6.5 bar City	ACT [°C]	EGR [%]	Cooling Power [W]	Fuel consumption [g/s]
Non-optimized	53.63	25	469.98	1.097
Optimized	47.76	25	303.21	1.088
		FC reduction [%]	0.83%	

What is emerging from Figure 4.62 is that the model is performing better, in fact the number of configurations analyzed is higher than all the previous tests. In this case the

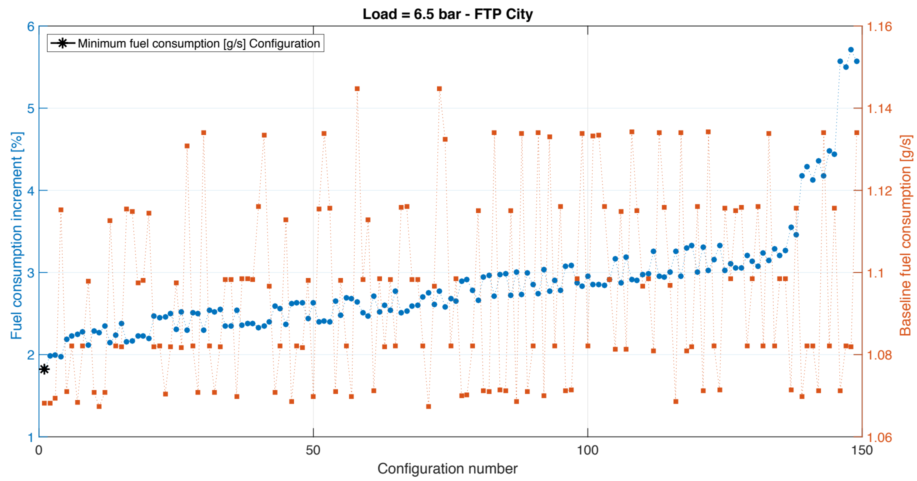


Figure 4.62: *Percentage increment of fuel consumption (blue) in respect to the baseline configurations (red) - 6.5 bar City*

best configuration is found to be the number 1, which is also the lowest cooling power configuration.

In Figure 4.63, the corrected fuel consumption contour plot is showing a trend more similar to the baseline one, this is the effect of the smoothening of the surfaces at higher loads.

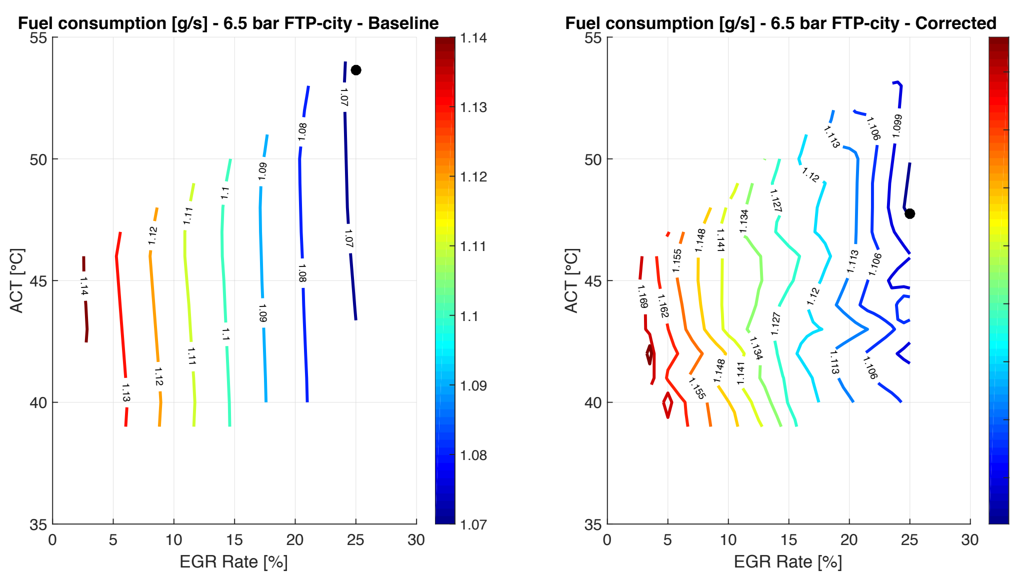


Figure 4.63: *Best configurations to reduce fuel consumption: baseline on the left and corrected on the right - 6.5 bar City*

Highway

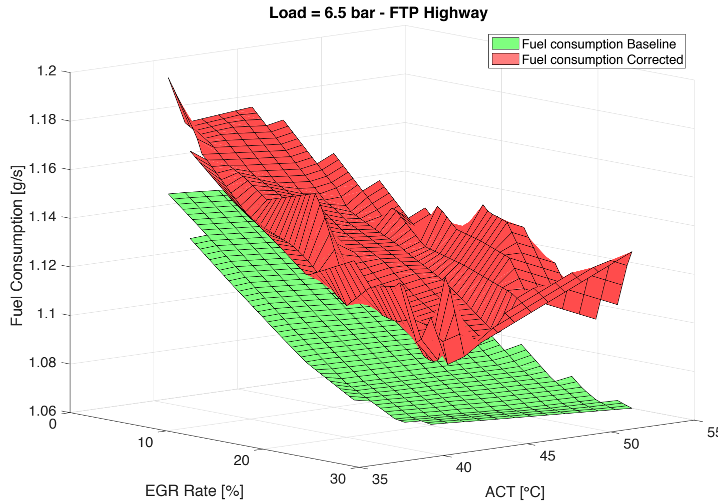


Figure 4.64: *Fuel consumption surfaces - 6.5 bar Highway*

One fact that should be noticed when increasing the load is that the ranges in which the data are plotted are reduced; particular attention should be paid on the values of x and y axes, which are, to give a better resolution, not all set at the same ranges.

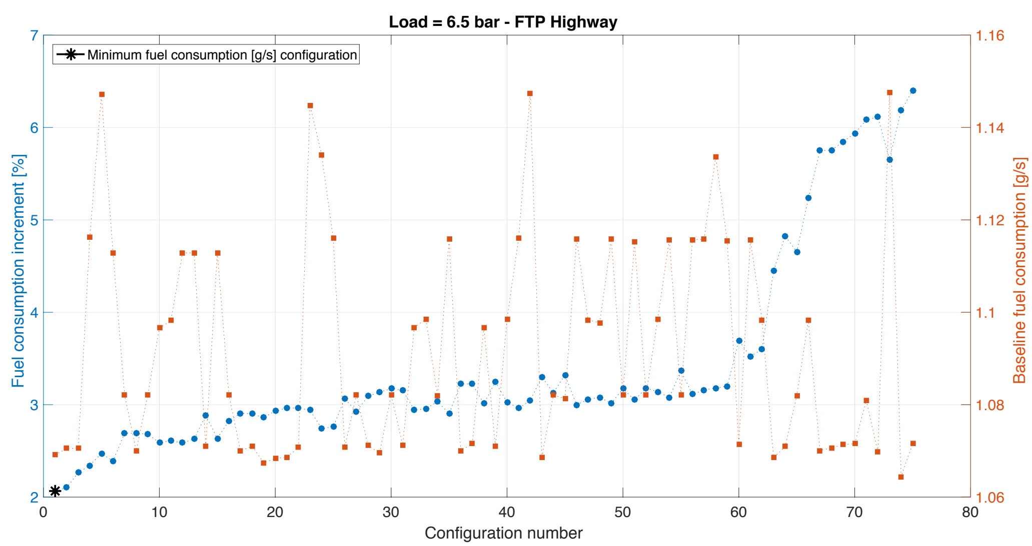


Figure 4.65: *Percentage increment of fuel consumption (blue) in respect to the baseline configurations (red) - 6.5 bar Highway*

In Table 4.10 the results obtained for the best configurations are reported, in this case the reduction is quite significant and similar to lower loads; it is important to recall that it could be achieved only changing the control strategy over the ACT and EGR rate, meaning that no hardware modifications should be made.

Table 4.10: *Fuel consumption results for baseline and corrected configurations - 6.5 bar Highway*

6.5 bar Highway	ACT [°C]	EGR [%]	Cooling Power [W]	Fuel consumption [g/s]
Non-optimized	53.38	27	1019.03	1.130
Optimized	42.15	27	344.57	1.091
		FC reduction [%]	3.45%	

An interesting trend, in line with all the previous considerations, could be individuated looking at Figure 4.67. Remembering that at 6.5 bar the BSFC curves were starting to bend upwards, the BSFC points for the corrected configurations are now slightly above the baseline ones; this effect will be even more prominent for higher loads.

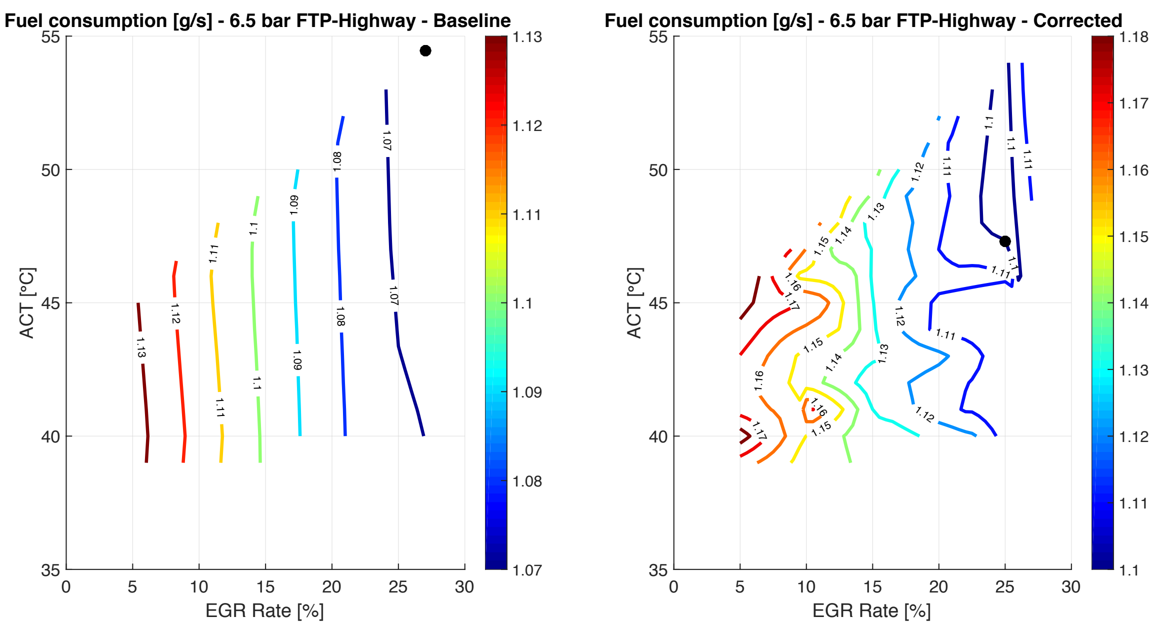


Figure 4.66: *Best configurations to reduce fuel consumption: baseline on the left and corrected on the right - 6.5 bar Highway*

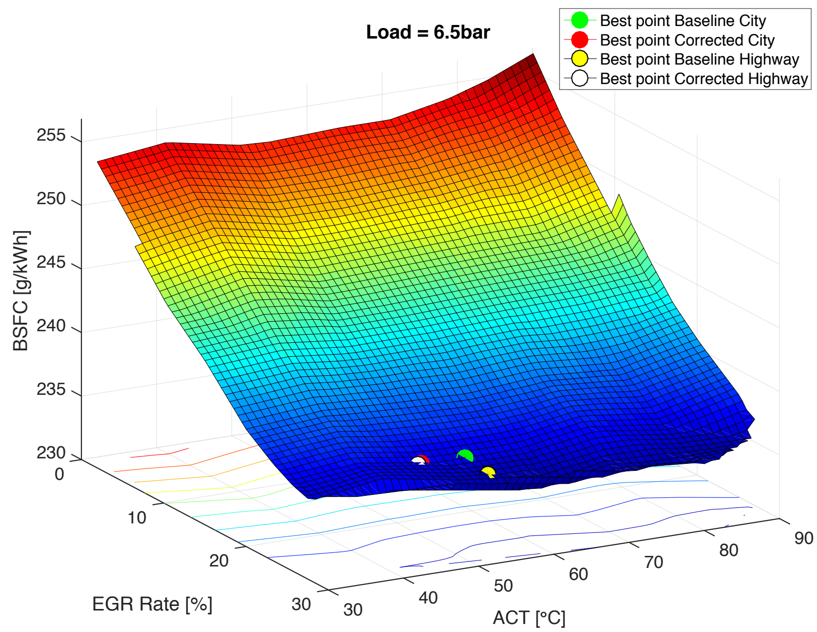


Figure 4.67: *Position of the best actuators configurations over the baseline BSFC surface - 6.5 bar*

4.3.8 Fuel consumption at load = 10 bar

From this working point on, a distinction between the City and Highway tests will not be present.

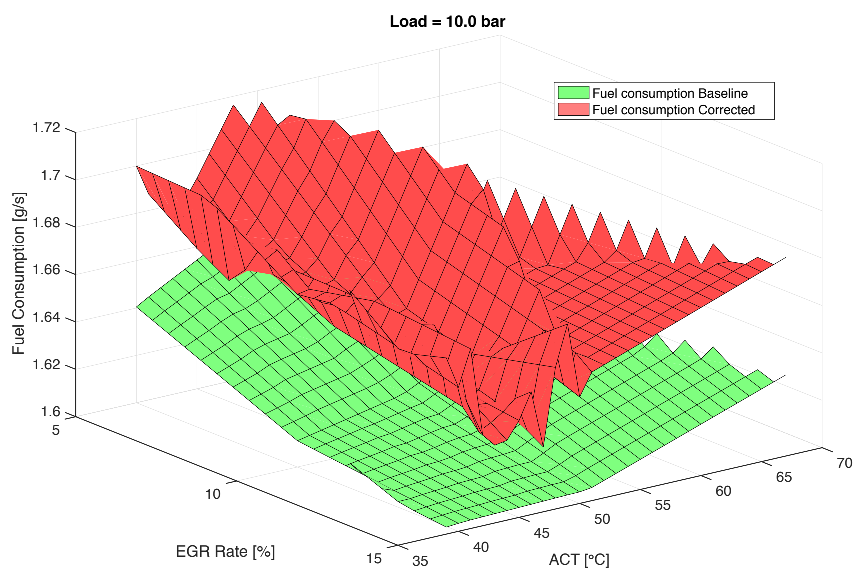


Figure 4.68: *Fuel consumption surface - 10 bar*

For this load condition the corrected fuel consumption surface, Figure 4.68, is starting to show some irregularities. This could be addressed to the fact that the next load point on which the BSFC values were interpolated is at 15 bar. With such a high gap between the two loads the results could be impaired, so they should be taken with a certain uncertainty.

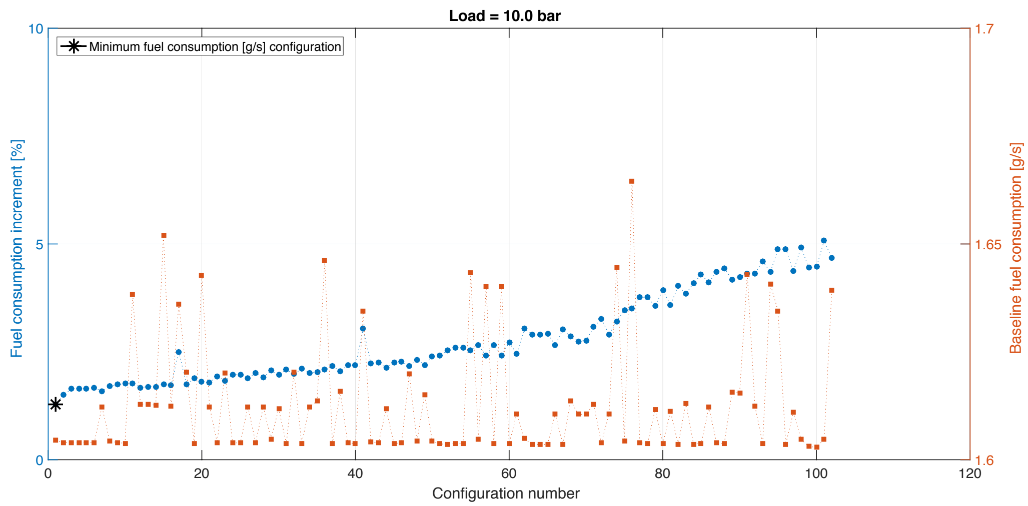


Figure 4.69: Percentage increment of fuel consumption (blue) in respect to the baseline configurations (red) - 10 bar

In Figure 4.70 the effect of the ACT in respect to the EGR rate is starting to increase. The best configuration from the corrected fuel consumption calculation has moved to a higher manifold temperature, this could be explained again considering the additional load required by the cooling system to cool a lot the mixture. With the usage of simulations it has been highlighted, in fact, that it is better to let the engine run at a higher temperature decreasing the BSFC, but reducing at the same time and in a greater extent the cooling system power request.

Table 4.11: Fuel consumption results for baseline and corrected configurations - 10 bar

10 bar	ACT [°C]	EGR [%]	Cooling Power [W]	Fuel consumption [g/s]
Non-optimized	38.83	15	1006.48	1.674
Optimized	47.21	15	256.98	1.625
		FC reduction [%]	2.95%	

In Figure 4.71 the point representing the best configuration for the lowest corrected fuel consumption is again placed slightly above the surface, further demonstrating that the region now analyzed is where the BSFC is increased for increased loads.

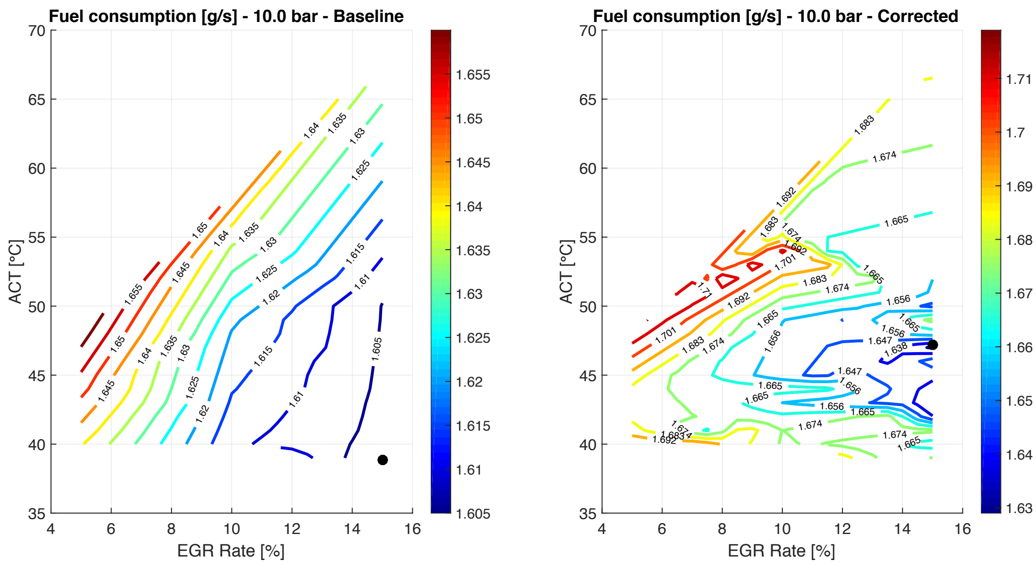


__Figure 4.70:__ Best configurations to reduce fuel consumption: baseline on the left and corrected on the right - 10 bar

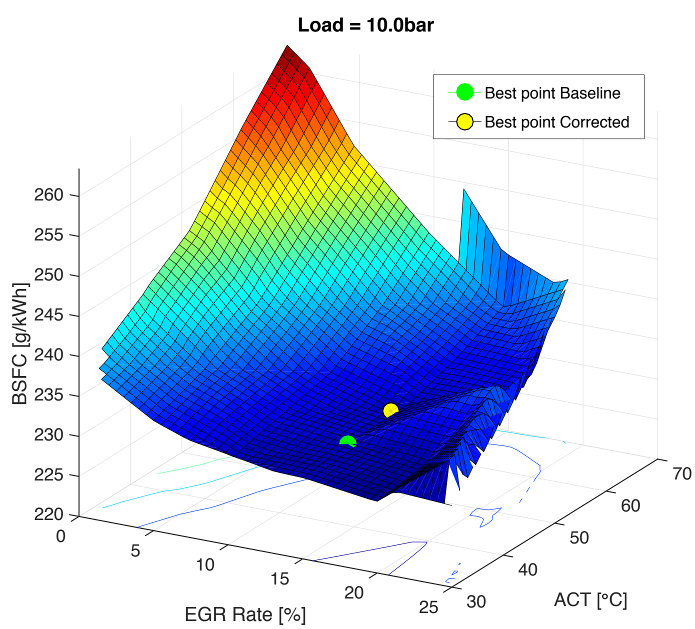


__Figure 4.71:__ Position of the best actuators configurations over the baseline BSFC surface - 10 bar

4.3.9 Fuel consumption at load = 15 bar

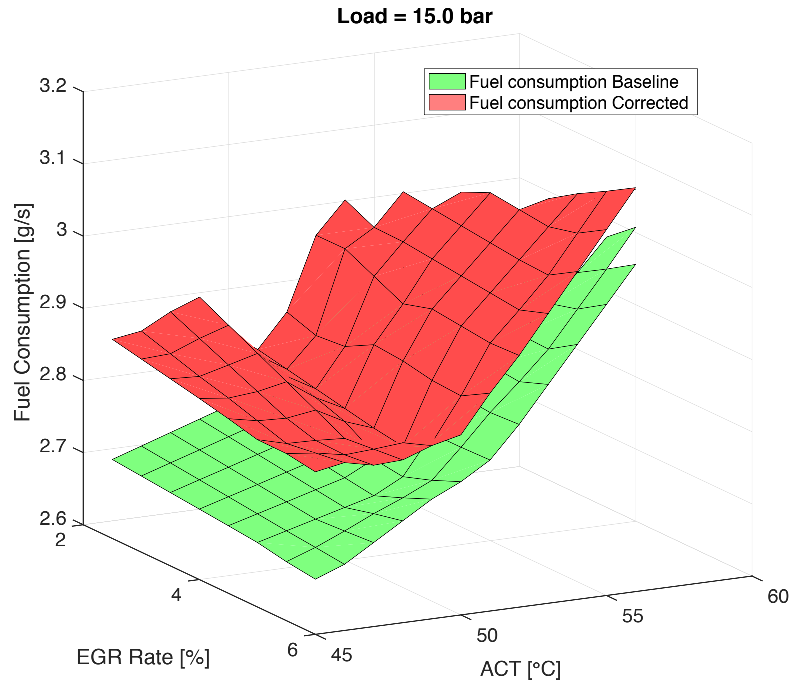


Figure 4.72: *Fuel consumption surface - 15 bar*

The results obtained for this load have to be considered carefully since the surface used to calculate the BSFC for loads above 15 bar is extrapolated from the previous points. In doing so the results could not be accurate, however it should be mentioned that these values are not automatically generated by the software, but are calculated manually. This operation is necessary since the extrapolation technique performed by the software itself is creating surfaces that are not representing at all a possible real trend; abrupt changes from point to point are observed and so it has been decided to proceed manually.

Looking at Figure 4.72 it is possible to notice that, despite the extrapolation, the surfaces do not present strange behavior, meaning that the results, always considered with the required uncertainty, are fairly representative of the fuel consumed by the cooling system.

In Figure 4.74 the preponderant effect of the ACT is shown, and it is highlighted, despite the extrapolation made, on the graph on the right. For this load condition the two best configurations not only differ on the ACT value, but also on the EGR rate. This is meaningful since now also the amount of recirculated gas is accounted: cooling a higher

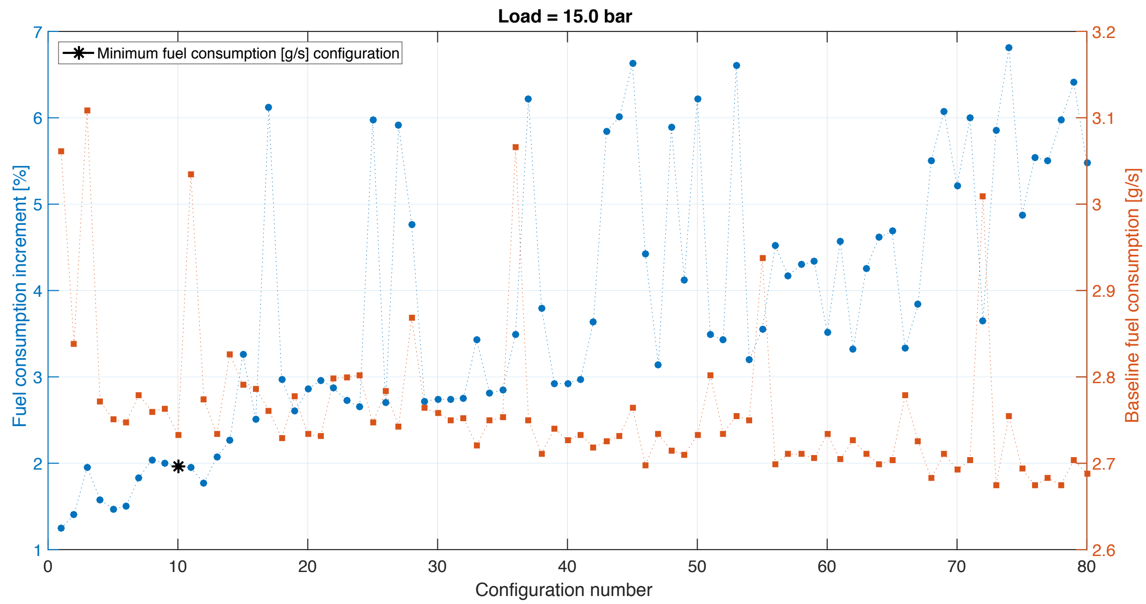


Figure 4.73: *Percentage increment of fuel consumption (blue) in respect to the baseline configurations (red) - 15 bar*

Table 4.12: *Fuel consumption results for baseline and corrected configurations - 15 bar*

15 bar	**ACT [°C]**	**EGR [%]**	**Cooling Power [W]**	**Fuel consumption [g/s]**
Non-optimized	44.99	6.2	1530.73	2.822
Optimized	49.03	5	929.46	2.786
		FC reduction [%]	1.28%	

flow rate requires a higher cooling power.

The results for each configuration are reported in Table 4.12 and when considering the cooling power a reduction in more than a half is achieved, but it is not directly translated in a great decrease in fuel consumption, limited to the value of 1.28%.

Finally the position of the best actuators configuration over the BSFC curve at 15 bar is reported in Figure 4.75 and it is quite interesting the fact that the surface is occupying a very narrow part of the space: EGR between 0 to 10% and ACT between 30 and 60°C.

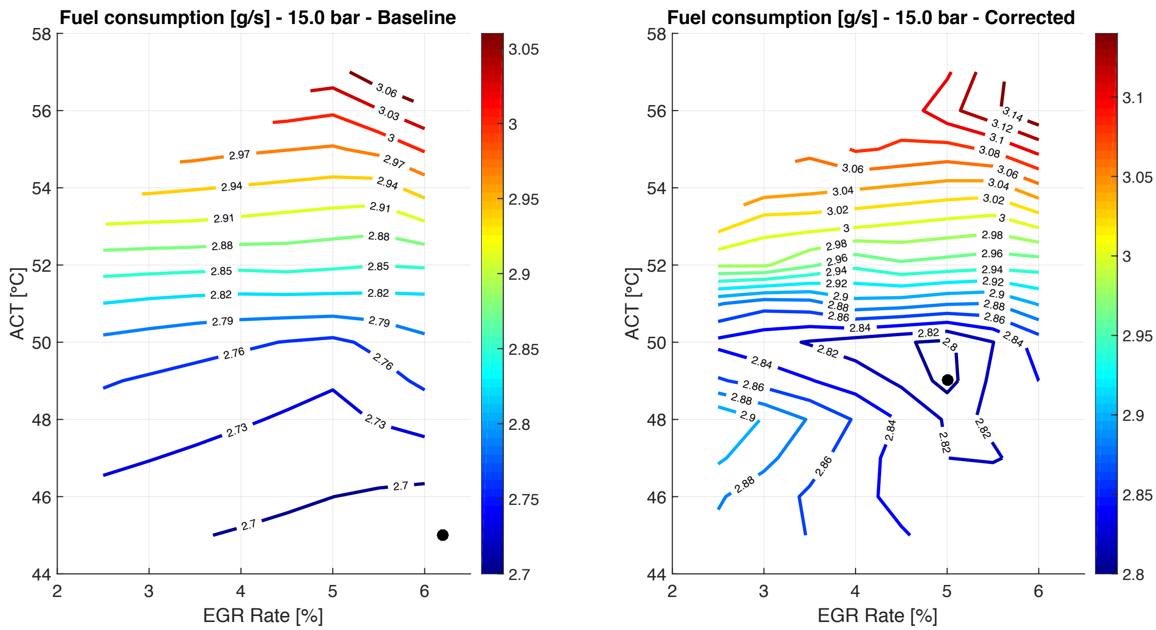


Figure 4.74: *Best configurations to reduce fuel consumption: baseline on the left and corrected on the right - 15 bar*

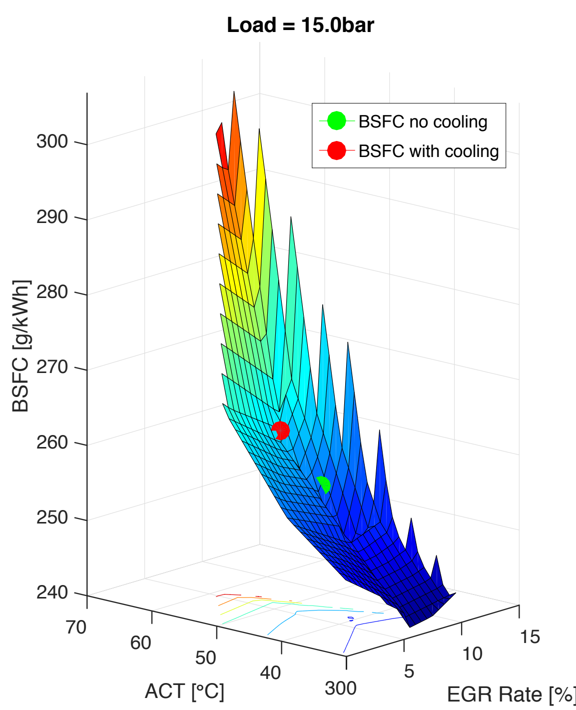


Figure 4.75: *Position of the best actuators configurations over the baseline BSFC surface - 15 bar*

4.3.10 Fuel consumption at load = 17.34 bar

This load condition is particularly severe for the engine and the cooling system since it is run at high speed, high load and 36°C external temperature. From a point of view of the combustion the engine should admit the mixture at the lowest possible manifold temperature, but at the same time the load required on the cooling system would be rather high, and so a tradeoff must be found.

In order to relief the duty of the cooling system some tests with mixture enrichment, to reduce in-chamber temperature, were performed. The tests were run always bearing in mind that this technique is directly affecting in a great extent the fuel consumption, but the final outcome from this test point is to discover if it is better to cool the engine using the cooling system or enriching.

The results presented for this working point are only reported as a general study, since they are obtained after strong assumptions:

1. Only one load condition has been run, at 17.34 bar. With solely the data at this point it is impossible to interpolate or to extrapolate the BSFC values for other loads.

2. For the calculations of the corrected fuel consumption the data used are, considering the above point, the baseline BSFC and the corrected torque. In doing so it is supposed that at higher loads the BSFC value is the same of the baseline one. This is unrealistic and lead to underestimation in the final result of fuel consumption since, for loads higher than 17.34 bar, the BSFC should further increased.

3. To compare the effect of enrichment a "corrected" BSFC has been supposed, and an extreme case has been analyzed, in fact the "corrected" BSFC value, used to compare the fuel consumed in enriched condition, is increased by 20% from the baseline one.

Bearing in mind the above points it is possible to proceed analyzing the results.

In Figure 4.76 the fuel consumption surfaces are presented, it is fundamental to notice that the red surface is nothing more than the green curve but shifted to higher values: the reason is explained in point 2 above.

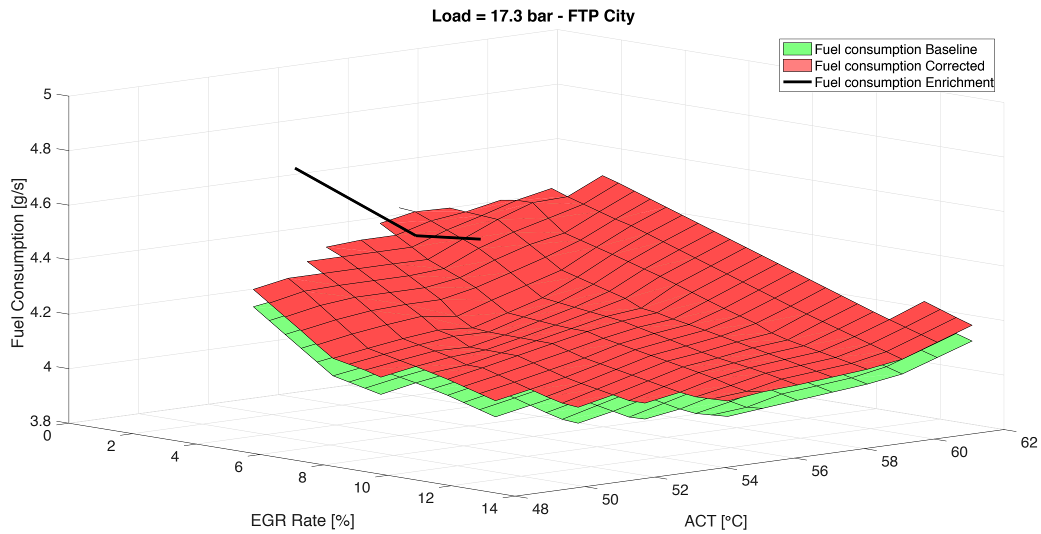


Figure 4.76: *Fuel consumption surface - 17.34 bar*

Always considering the assumptions made it is easy to understand that the best configuration for the lowest fuel consumption is found to be the same in both cases. This fact is shown in Figure 4.77 and in Table 4.13.

It is of particular interest, instead, the iso-fuel consumption plot in Figure 4.78 where the corrected fuel consumption contour lines are plotted together with the fuel consumption points of the enrichment technique. The data of the fuel consumed having a richer mixture are directly retrieved from the dyno tests since are parameters actually measured.

The first idea coming from this graph is that enriching the mixture is worsening a lot the fuel economy, but, this cannot be taken as strictly true due to the approximation processes made in the calculations.

Table 4.13: *Fuel consumption results for baseline and corrected configurations - 17.34 bar Highway*

17.34 bar	**ACT [°C]**	**EGR [%]**	**Cooling Power [W]**	**Fuel consumption [g/s]**
	52.37	10	1450.44	3.999
		FC reduction [%]	0%	

A final conclusion could be instead draw looking at Figure 4.79 where the fuel consumption calculated with BSFC values 20% higher than the baseline is plotted together with the

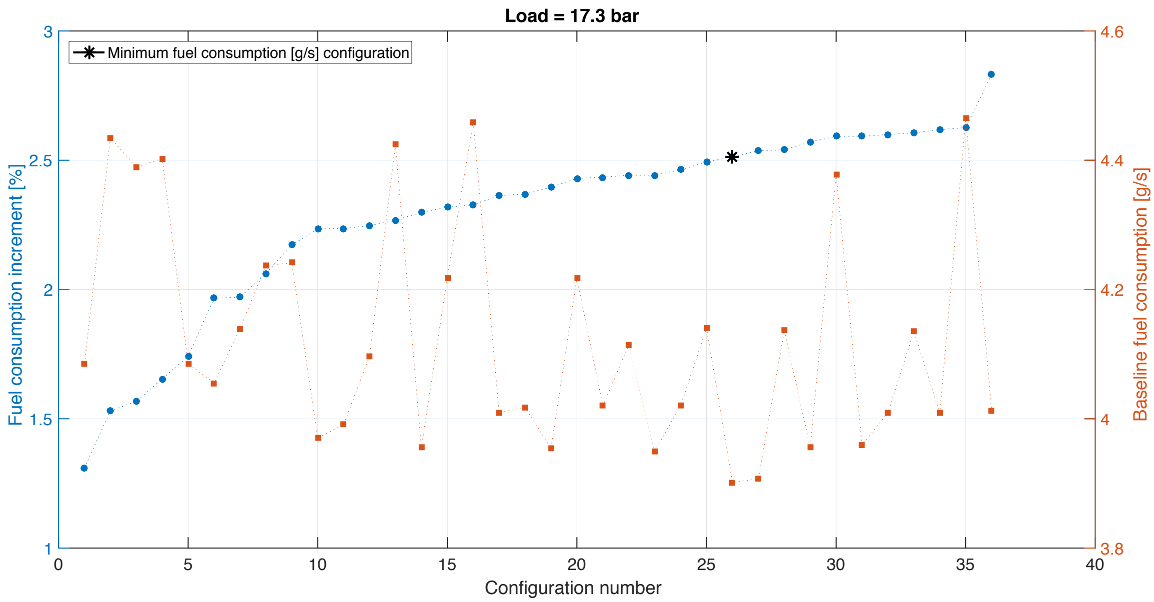


Figure 4.77: Percentage increment of fuel consumption (blue) in respect to the baseline configurations (red) - 17.34 bar

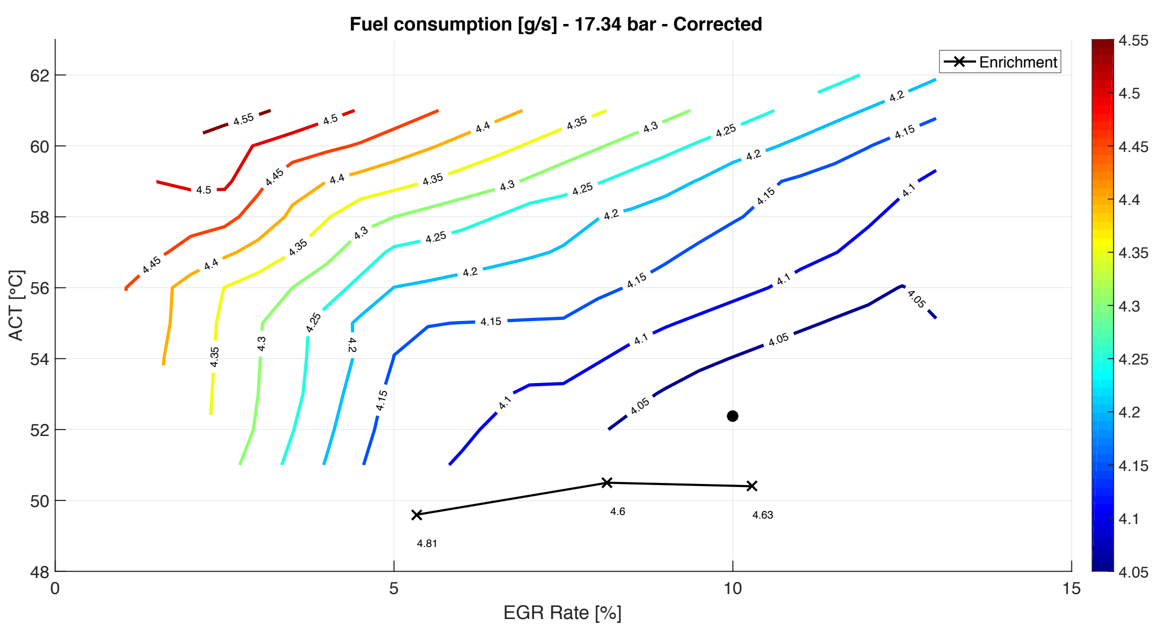


Figure 4.78: Best configurations to reduce fuel consumption - 17.34 bar

enrichment ones. It must be considered that an increase in 20% in not a realistic value, but it is exaggerated and it is actually producing overestimated fuel consumption results. The increase in load due to the cooling system is no more than 0.4 bar, meaning that the increase in the BSFC could not be as high as 20%.

Accordingly to the very high values of the BSFC considered, enrichment is showing a lower fuel consumption rather than the corrected one. Nevertheless if more data on the BSFC would have been considered, cooling the engine using the cooling system allows to save fuel in respect to the enrichment of the mixture.

The savings discovered only avoiding the usage of mixture enrichment are in the range of 14 to 18%, when the initial values of BSFC are considered.

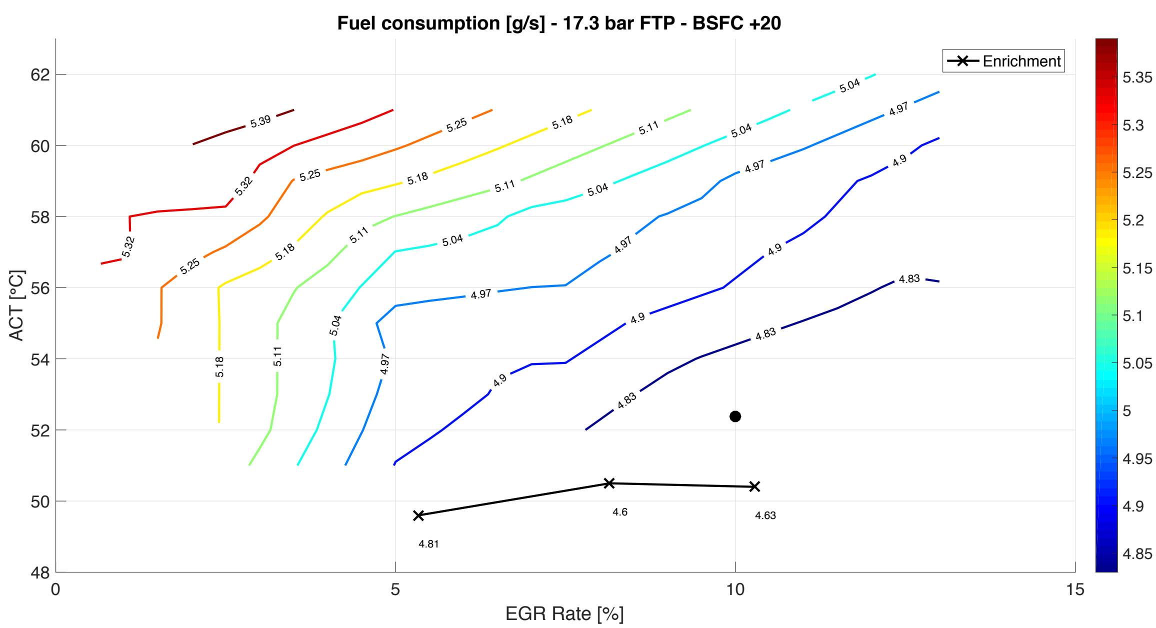


Figure 4.79: *Best configurations to reduce fuel consumption with BSFC increased by 20% - 17.34 bar*

For this configuration, however, another control strategy could be considered. Since it is a condition in which high power output from the engine is required, the value to look at, to calculate the best actuators position is no more the one with the lowest fuel consumption, but the one with the lowest cooling power request. In this case, as shown in Figure 4.77, they don't coincide. With this consideration, in Table 4.14 the results which minimize the power and the one which minimize the fuel consumption are presented. Even when the minimum resistance torque is considered (first row) the fuel consumption is still well below the one with enrichment. However, it should be noticed, that the difference in torque between the best configuration for the minimum power subtracted and the best configuration for the minimum fuel consumption differ in torque only by 1.19% while the difference in fuel consumption is of 3.37%. Looking at these data a probable final decision would bear the small increase in torque to privilege the reduction in the fuel consumption.

Table 4.14: *Fuel consumption results for lowest power request by the cooling system and lowest fuel consumption - 17 bar*

17 bar	**ACT [°C]**	**EGR [%]**	**Cooling Power [W]**	**Fuel consumption [g/s]**
Minimum Power	54.545	5	756.03	4.139
Minimum Fuel	52.366	10	1450.44	3.999

Chapter 5

Conclusions

In the chapter dedicated to the results, for each load and vehicle speed, the best actuators configurations have been individuated. In the results collected from the simulations conducted on the cooling system model, the optimal position to be set for each single component, such as pump speed, thermostat opening position etc., is stored and can be retrieved to be implemented in real driving conditions. These optimal positions, for clear reasons of company secrets, cannot be published in this study.

Some final conclusions could be nonetheless drawn observing the various results obtained.

The first concept that should be always kept in mind is that the reduction in fuel consumption calculated during this project is achieved only considering a different approach to the problem. The hardware configuration has not been changed in any of its parts and this is of paramount importance since the fuel consumption reduction is obtainable only changing the control strategy: increasing or decreasing the EGR rate or the ACT depending on the situation considered.

From the tests on the engine dynamometer some preliminary results are obtained and a sort of control strategy could be implemented. However, the problem of this approach is that it is looking deep in detail to what is better for the combustion and the engine itself without considering the auxiliary components attached to it. This represents a major drawback of the tests performed in these conditions. On the other hand, simulating the engine behavior with all the cooling module would require additional tests and instrumentations, increasing rapidly the related costs.

In the studies conducted so far the engine and the cooling system were observed separately, but with the procedure followed in the project, their mutual effects are taken into account, with clear benefits from the fuel consumption point of view.

The effects of EGR and ACT observed in the first part of the project are in line with what is expected from studies in literature. The knocking suppression coming from the lower ACT is helping a lot in improving the combustion phasing, obtaining more power output from the engine burning less fuel.

At the same time EGR has been found to have the same effect on the combustion phasing, especially giving the possibility to further advance the spark event at high EGR rates. Its positive effect in reducing pumping losses has been analyzed as well, and it has been found to be one of the reasons of the great reduction in fuel consumption.

When considering low manifold temperature, it must be taken into account also the additional power required by the cooling system. This point has been stressed many times, since it represents the key of the project. The analysis on the best temperature and EGR rate would be meaningless if the whole system is not considered.

In order to create a general guideline, what is obtained in the results could be summarized in the following:

- For *light loads*, running the engine at a higher temperature is beneficial since pumping losses are reduced. In these conditions the main effects in fuel reduction are obtained varying the EGR rate rather than changing manifold temperature. The low influence of this parameter is playing a fundamental role since the power request on the cooling system could be reduced allowing higher charge temperatures.

- For *high loads* the best fuel economy is achieved for lower manifold temperatures. In this case the EGR effect is not as influent as in the previous condition, but, as far as the combustion stability is maintained, it is better to work with high amounts of recirculated gas. In this way it is taken advantage of the positive effects of the two solutions. Lowering too much the manifold temperature however could produce side effects of increased fuel consumption due to the higher additional load created by cooling system.

- For *severe conditions*, similar to the one analyzed at 17.34 bar, enrichment is found to require a very high amount of fuel. To correctly compare the enrichment of the mixture to the usage of the cooling system, it has been looked for the same CA50 of the 40°C manifold condition. The final conclusion is that, despite the severe conditions, it is still better to cool the engine using the cooling system.

In conclusion the development of a general procedure for all the working points is not possible since each point should be analyzed carefully. However a common trend has been observed, and few generic assumptions to control the engine and the cooling system to have the lowest fuel consumption can be presented:

- Operate the engine at the highest EGR rate for the load condition observed,
- Let the manifold temperature take values around 40 and 50°C
- Avoid enrichment whenever is possible since it impairs in a great extent the fuel economy.

These presented above are general rules and each load condition should be analyzed more in detail to take real advantage from the outcome of the project.

The major disadvantage of this project is its impossibility to analyze transient conditions, but it is only related to steady state conditions.

This fact could affect the control strategy during abrupt accelerations when the pedal is completely pushed down. In this phase the engine, maybe, would require a lower operating temperature and so the control on the cooling system should be changed. The problem is that this change, since is related to cooling fluid in a liquid state, could require a certain time to reach the desired manifold temperature. An ideal strategy would change this temperature instantaneously, but it is clearly impossible. The different operating temperatures in steady state and in transient conditions should be analyzed with great attention. A small temperature difference between the two should be individuated, in such a way the change could be done in a lower amount of time.

In conclusion, it should be mentioned that:

- Most of the gas temperatures and flow rates are directly implemented in the 1D model after being collected during the dyno tests, further improving the simulations reliability.

- The 1D cooling system model could take into account also the effect of the air conditioning system, but for the purpose of this project, no analysis were run with the A/C switched on.

- The attention is only focused on the engine and the cooling system without considering emissions. A deeper analysis should also take into account emissions and their interaction with EGR rates and ACT in order to comply with the stricter regulations.

- The best fuel consumption point is found analyzing many important parameters at the same time: combustion stability, engine durability, cooling system power requirements and their close interaction.

At the end, to summarize, the contribution of this project for the company will result in an improvement over the past method of analysis. The old procedure, in fact, was considering an optimization or of the engine or of the cooling system, considered separately. The drawback of this previous method was that, when considered together, the optimal configuration for the cooling system would result in a configuration worse from the engine point of view. The innovative element of this project is that both the engine and cooling system are considered at the same time. This is fairly important because it leads to find the optimized configuration when the systems are considered acting together.

Chapter 6

Recommendations

For future studies some points should be improved in order to have higher reliability on the results. Here the main modifications and improvements are briefly summarized and presented.

- Perform additional dyno tests at different load conditions, in doing so, having a higher amount of data it is possible to have better interpolation of the BSFC values and obtain more reliable results as well.

- Test the engine also at different engine speeds. This will increase the amount of available results affecting directly the complexity of the analysis, but at the same time a higher resolution on the working conditions is obtained.

- Improve the capabilities of the 1D model to simulate low load conditions, which for some actuators positions did not reach a steady state result and consequently failed.

- Build the control strategy for the cooling system, maybe with the usage of look-up tables that, depending on the engine load, gear engaged and vehicle speed, set the different actuators to the correct position to minimize fuel consumption.

Bibliography

[1] *http://www.gasbuddy.com/Charts* as May 2016

[2] *https://turbo.honeywell.com/wp-content/uploads/2011/12/charts-31.gif* as April 2016

[3] *https://en.wikipedia.org/wiki/Brake-specific-fuel-consumption/media/File:Brake-specific-fuel-consumption.svg* as May 2016

[4] R. Golloch, G. P. Merker, *Internal Combustion Engine Downsizing. Fundamentals, State of Art and Future Concepts*, MTZ Worldwide, Vol.66/2, pp. 20-22, doi: 10.1007/BF03227737

[5] S. K. Chen, P. Flynn, *Development of a Compression Ignition Research Engine*, SAE Technical Paper 650733, 1965, doi:10.4271/650733

[6] G. Police, S. Diana, V. Giglio, B. Iorio, N. Rispoli, *Downsizing of SI Engines by Turbocharging*, Proceedings of ESDA2006, 8^{th} Biennial ASME Conference on Engineering Systems Design and Analysis. Torino, Italy, July 4-7, 2006

[7] J. B. Heywood, *Internal Combustion Engine Fundamentals*, McGraw-Hill International Editions, Singapore 1989

[8] A. D. J. Fraser, *How Low can we go? Challenges and opportunities of engine downsizing to reduce CO2 emissions* Seminar Proceedings IMechE, pp. 1-9, London, UK, 2011

[9] *http://s4wiki.com/wiki/Wastegate_bypass_regulator_valve* as May 2016

[10] C. Cantemir, *Twin Turbo Strategy Operation*, SAE Technical Paper 2001-01-0666, 2001, doi:10.4271/2001-01-0666

[11] K. Osako, T. Yokoyama, T. Yoshida, T. Hoshi, M. Ebisu, T. Shiraishi, *Development of Twin-scroll Turbine for Automotive Turbochargers using Unsteady Numerical Simulation*, Mitsubishi Heavy Industries Technical Review, Vol. 50/1, pp. 23-31 March 2013

[12] *http://image.superstreetonline.com/f/16322612+w+h+q80+re0+cr1+ar0+st0/modp-0906-03-o-twin-scroll-diagram.jpg* as May 2016

[13] *http://www.fastmotoring.com/index.php/2010/10/variable-nozzle-turbine-vnt-or-variable-geometry-turbo-vgt/* as May 2016

[14] *https://simanaitissays.files.wordpress.com/2014/08/schematic1.jpg* as May 2016

[15] U. Spicier, F. Sarikoc, *Engine Downsizing by Gasoline Direct Injection and Turbocharging*, International Symposium on Transport Phenomena and Dynamics of Rotating, pp. 595-602, Red Hook, NY, 2010

[16] D. Petitjean, L. Bernardini, C. Middlemass, S. M. Shahed, *Advanced Gasoline Engine Turbocharging Technology for Fuel Economy Improvements*, SAE Technical Paper, 2004-01-0988, 2004, doi:10.4271/2004-01-0988

[17] S. Park, J. Lee, K. Kim, S. Park, H. Kim, *Experimental characterization of cooled EGR in a gasoline direct injection engine for reducing fuel consumption and nitrogen oxide emission*, Heat and Mass Transfer, Vol. 51, No. 11, pp. 1639-1651, 2015, doi:10.1007/s00231-015-1633-0

[18] L. Francqueville, J. Michel, *On the Effects of EGR on Spark-Ignited Gasoline Combustion at High Load*, SAE Int. J. Engines 7(4):1808-1823, 2014, doi:10.4271/2014-01-2628

[19] Y. L. Zu, Z. X. Zhou, H. Y. Yu, *Characteristics Study of Knocking, Fuel Consumption and Emissions with Cooled External EGR on SIDI Turbo Charged Gasoline Engine*, Applied Mechanics and Materials, Vols. 66-68, pp. 102-107, 2011, doi: 10.4028/www.scientific.net/AMM.66-68.102

[20] *https://gsf165.files.wordpress.com/2013/06/cc-fig-2.gif* as May 2016

[21] *http://www.cradle-cfd.com/images/tec/column01/fig4-4.jpg* as May 2016

[22] S. d'Ambrosio, *Engine Cooling System*, Combustion Engines and their Application to Vehicles, Course Slides, Politecnico di Torino, 2015

[23] *Supercharger Charge Coolers - A System Perspective*, Vortech Engineering LLC, Channel Island, CA, 2004.

[24] *http://image.ec21.com/company/d/dk/dk6/dk6052/upimg/p7-1.jpg*, as May 2016.

[25] *https://www.dieselnet.com/tech/images/diesel/air/cool_low_dp.gif*, as May 2016.

[26] C. Malvicino, F. Di Sciullo, W. Ferraris, F. Vestrelli, F. Beltramelli, *Advanced Dual Level Vehicle Heat Rejection System for Passenger Cars*, SAE Int. J. Engines 5(3):1260-1267, 2012, doi:10.4271/2012-01-1204.

[27] J. Stehlig, R. Dingelstadt, J. Ehrmanntraut, R. Mueller, J. Taylor, *Air Intake Modules with Integrated Indirect Charge Air Coolers*, Proceedings of the FISITA 2012 World Automotive Congress: Volume 2: Advanced Internal Combustion Engines (II), pp. 1379-1387, 2013, doi:10.1007/978-3-642-33750-5-46.

[28] *http://www.valeo.de/medias/upload/2014/07/76008/module-admission-air-diaporama.jpg*, as May 2016

[29] G. H. Abd-Alla, *Using exhaust gas recirculation in internal combustion engines: a review*, Energy Conversion and Management, Vol. 43, Issue 8, pp. 1027-1042, May 2002, doi:10.1016/S0196-8904(01)00091-7.

[30] D. Takaki, H. Tsuchida, T. Kobara, M. Akagi, T. Tsuyuki, M. Nagamine, *Study of an EGR System for Downsizing Turbocharged Gasoline Engine to Improve Fuel Economy* SAE Technical Paper 2014-01-1199, 2014, doi:10.4271/2014-01-1199.

[31] S. Potteau, P. Lutz, S. Leroux, S. Moroz, E. Tomas, *Cooled EGR for a Turbo SI Engine to Reduce Knocking and Fuel Consumption*, SAE Technical Paper 2007-01-3978, 2007, doi:10.4271/2007-01-3978.

[32] H. Wei, T. Zhu, G. Shu, L. Tan, Y. Wang, *Gasoline engine exhaust gas recirculation – A review*, Applied Energy, Vol. 99, pp. 534-544, November 2012, doi:10.1016/j.apenergy.2012.05.011.

[33] J. Su, M. Xu, T. Li, Y. Gao, J. Wang, *Combined effects of cooled EGR and a higher geometric compression ratio on thermal efficiency improvement of a downsized boosted spark-ignition direct-injection engine*, Energy Conversion and Management, Vol. 78, pp. 65-73, 2014, doi:10.1016/j.enconman.2013.10.041.

[34] J. M. Lujàn, H. Climent, R. Novella, M. E. Rivas-Perea, *Influence of a low pressure EGR loop on a gasoline turbocharged direct injection engine*, Applied Thermal Engineering, Vol. 8, pp. 432-443, 5 October 2015, doi:10.1016/j.applthermaleng.2015.06.039.

[35] T. Alger, J. Gingrich, C. Roberts, B. Mangold, *Cooled exhaust-gas recirculation for fuel economy and emissions improvement in gasoline engines*, International Journal of Engine Research, Vol. 12 No. 3, pp. 252-264, June 2011, doi: 10.1177/1468087411402442.

[36] M. Kaiser, U. Krueger, R. Harris, L. Cruff, *"Doing More with Less"- The Fuel Economy Benefits of Cooled EGR on a Direct Injected Spark Ignited Boosted Engine*, SAE Technical Paper 2010-01-0589, doi:10.4271/2010-01-0589.

[37] E. Galloni, G. Fontana, R. Palmaccio, *Effects of exhaust gas recycle in a downsized gasoline engine*, Applied Energy, 105, issue C, p. 99-107, 2013, doi:10.1016/j.apenergy.2012.12.046.

[38] T. Lattimore, C. Wang, H. Xu, M. L. Wyszynski, S. Shuai, *Investigation of EGR Effect on Combustion and PM Emissions in a DISI Engine*, Applied Energy, Vol. 161, pp. 256-267, 1 January 2016, doi:10.1016/j.apenergy.2015.09.080.

[39] A. Cairns, H. Blaxill, G. Irlam, *Exhaust Gas Recirculation for Improved Part and Full Load Fuel Economy in a Turbocharged Gasoline Engine*, SAE Technical Paper 2006-01-0047, 2006, doi:10.4271/2006-01-0047.

[40] A. Cairns, Hugh Blaxill, *A Comparison of Knock Reduction Strategies in a Turbocharged DI Gasoline Engine*, GLOBAL POWERTRAIN CONGRESS. Detroit, Michigan, 2006.

[41] K. Kumano, S. Yamaoka, *Analysis of Knocking Suppression Effect of Cooled EGR in Turbo-Charged Gasoline Engine*, SAE Technical Paper 2014-01-1217, 2014, doi:10.4271/2014-01-1217.

Appendix A: Interpolated BSFC surfaces

In this appendix the BSFC surfaces at different loads are reported. These figure are explicative of how the surfaces should be when no interpolation errors are present. Comparing Figure 2 to Figure 4.43 (b) it is possible to understand that the interpolation made by the software when, all the data are considered, is not correct at all.

It should be also highlighted that in Figure 6 over the BSFC surface it is plotted also the BSFC for the enrichment condition.

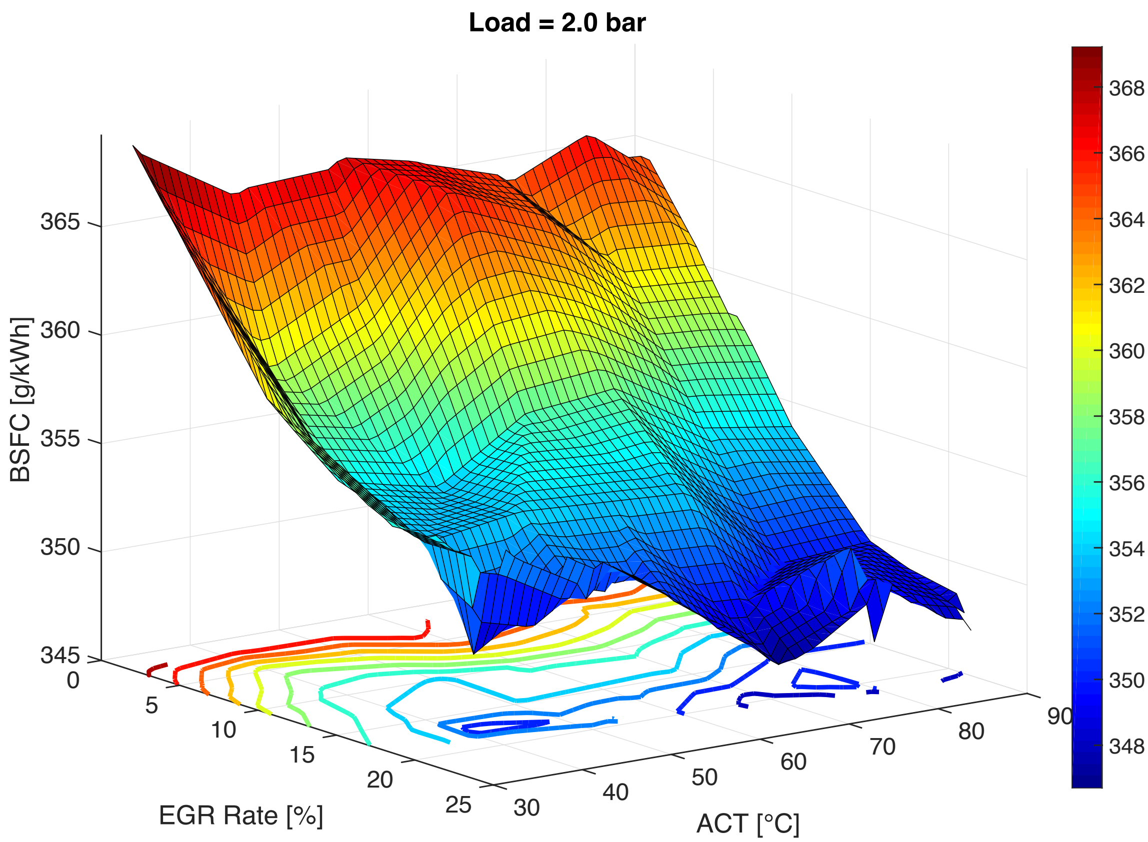


Figure 1: *BSFC surface at 2 bar load*

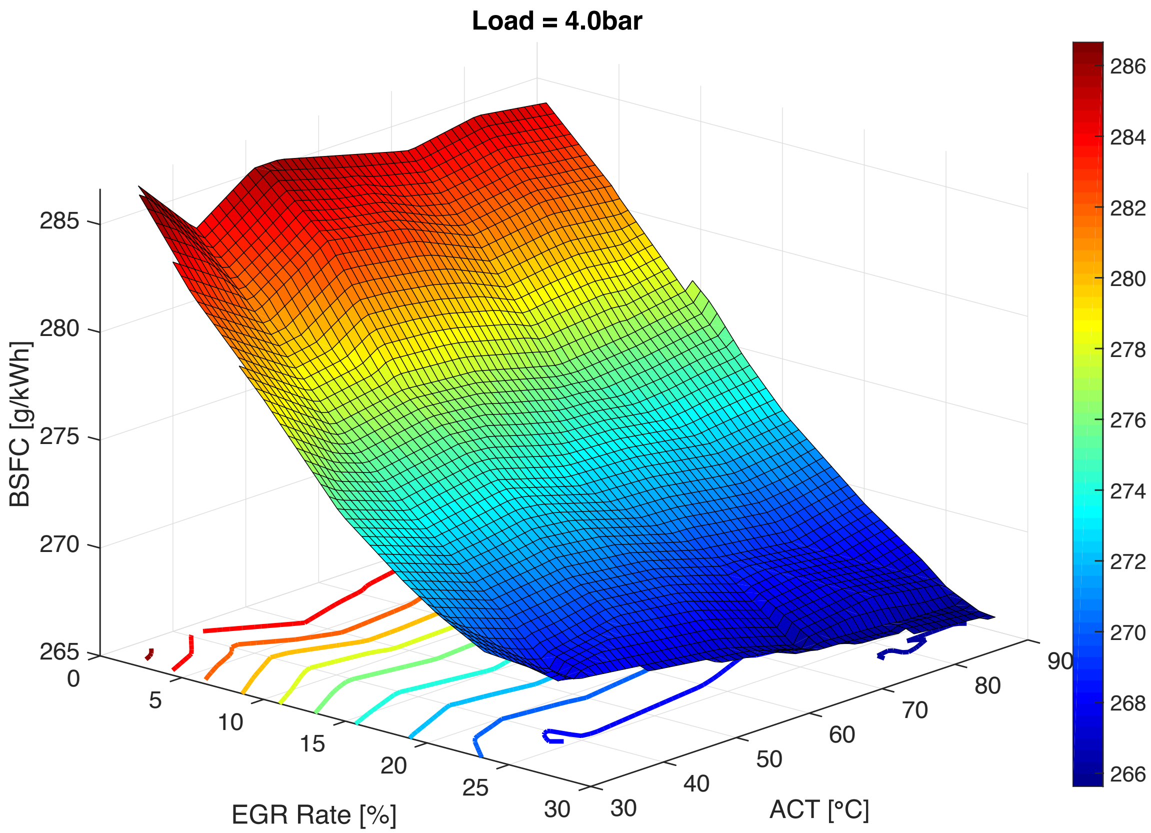


Figure 2: *BSFC surface at 4 bar load*

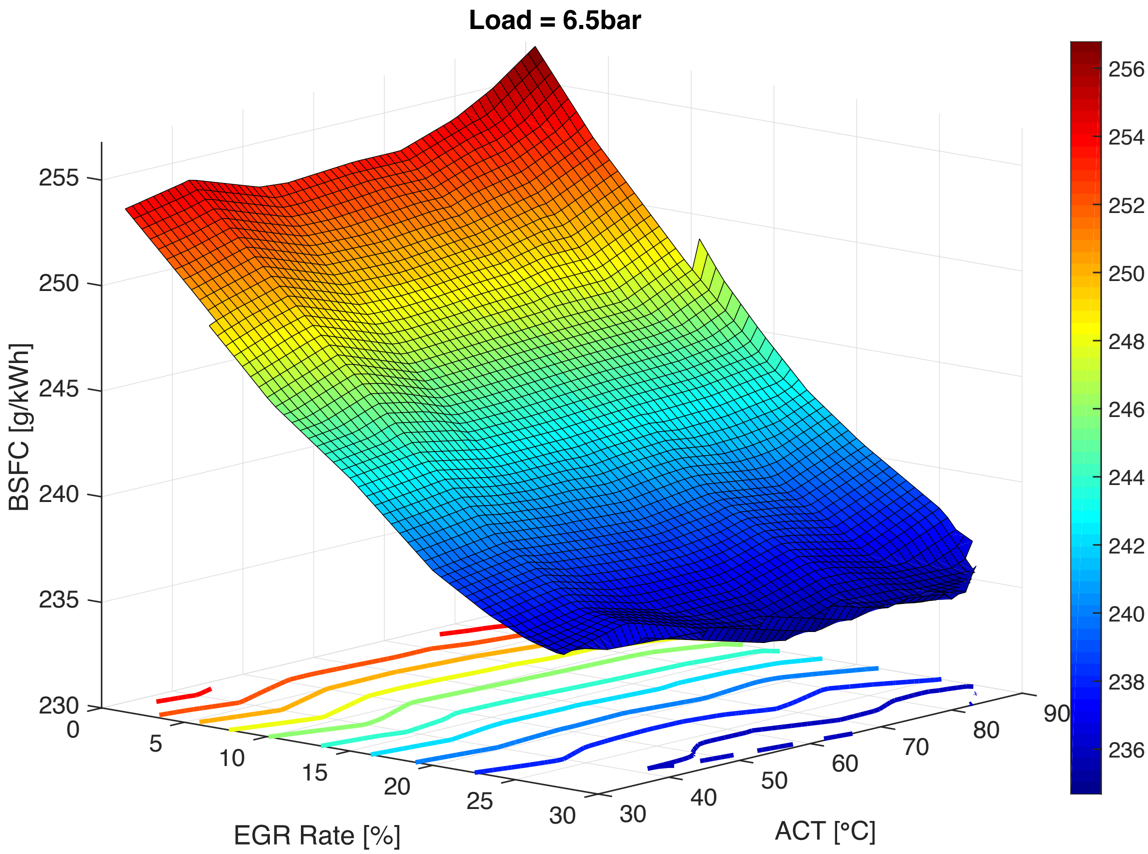


Figure 3: *BSFC surface at 6.5 bar load*

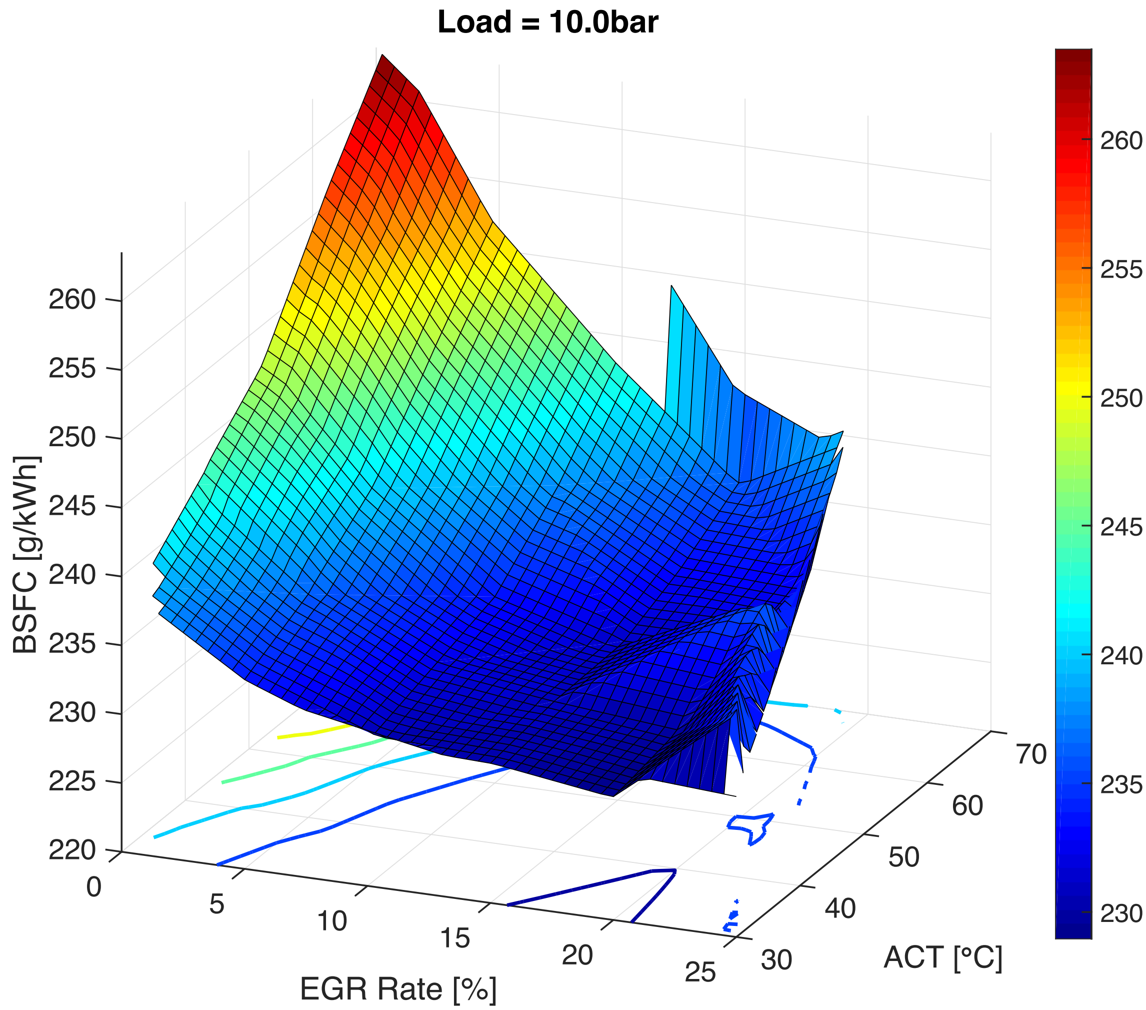


Figure 4: *BSFC surface at 10 bar load*

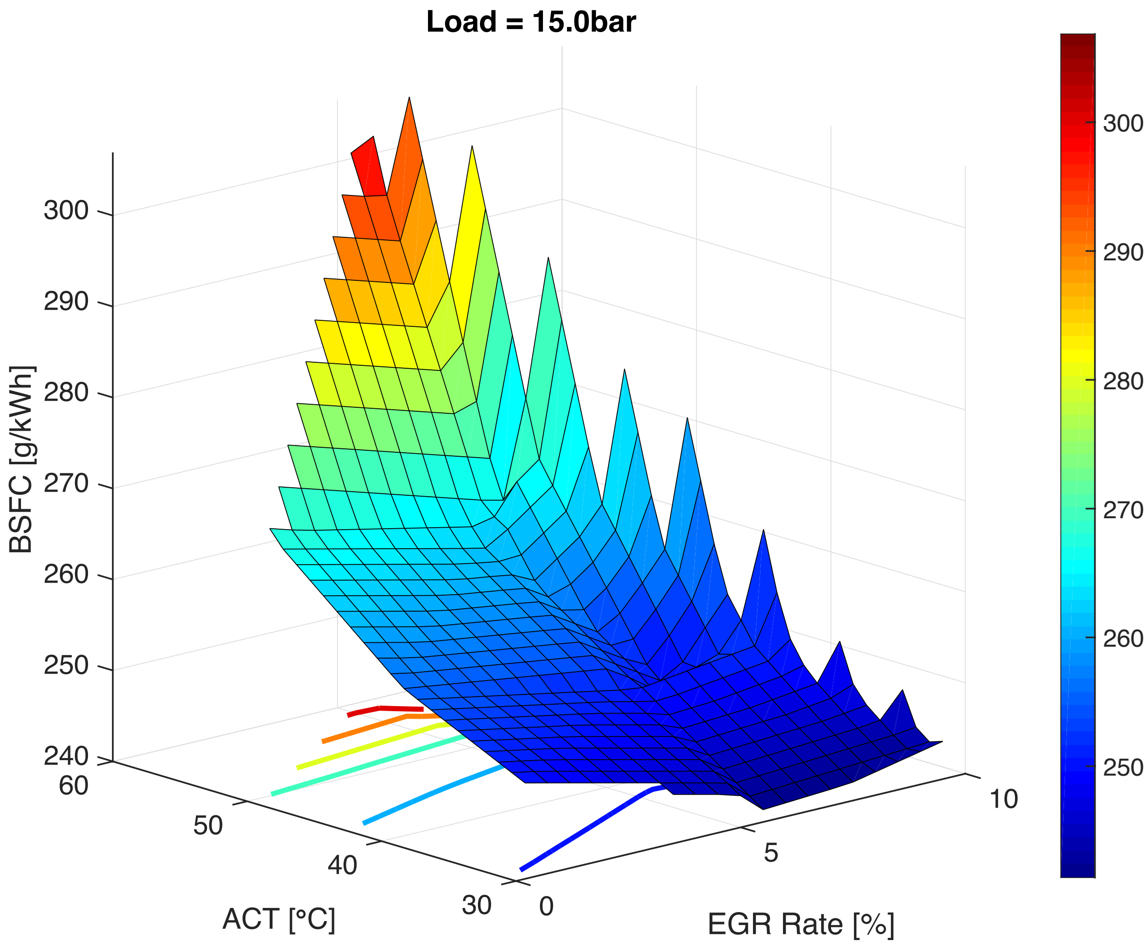


Figure 5: *BSFC surface at 15 bar load*

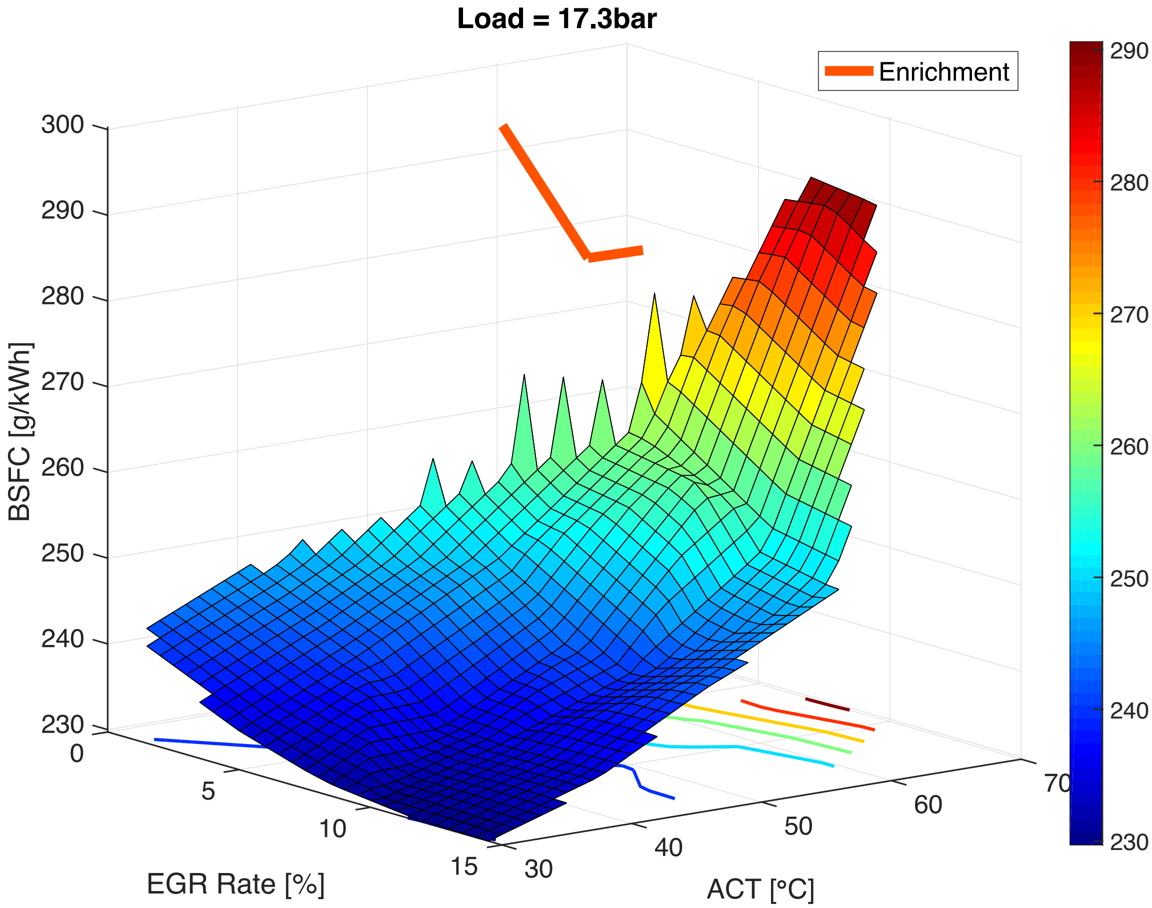


Figure 6: *BSFC surface at 17.34 bar load*

Appendix B:

PMEP data at different loads

In this Appendix, the data regarding the Pumping Mean Effective Pressure are presented in the case of the load conditions not presented in the main body. For the 2 and 4 bar BMEP conditions nothing remarkable should be mentioned; the trend is in line with the 6.5 bar one.

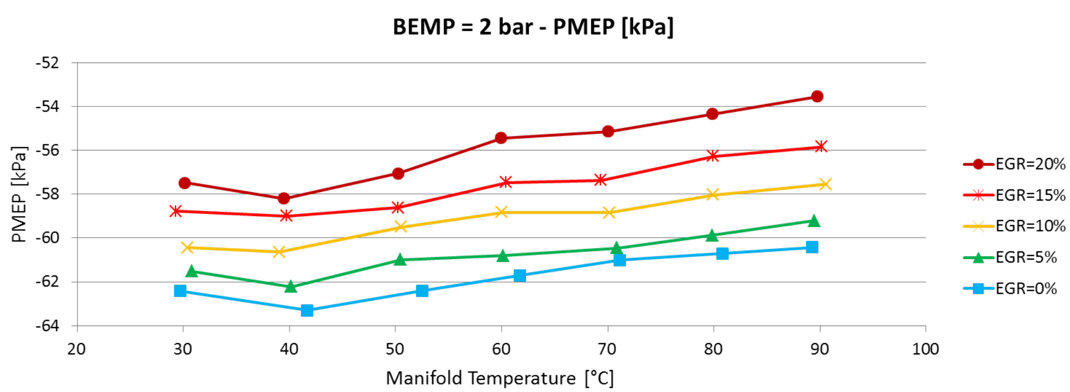


Figure 7: *Effects of manifold temperature and EGR rate on PMEP - BMEP = 2 bar*

Having higher manifold temperatures and higher EGR rates is directly translated in the need of open up the intake valve more. In doing so the pumping loop, in Figure 2.21, is reduced and the BSFC is reduced as well. So, as a final result, this great reduction in PMEP is producing the reduction observed in BSFC.

For the conditions at BMEP 10 and 15 bar, high amount of boost levels are present. This is directly influencing the pumping mean effective pressure. As it is shown in Figure 9 and

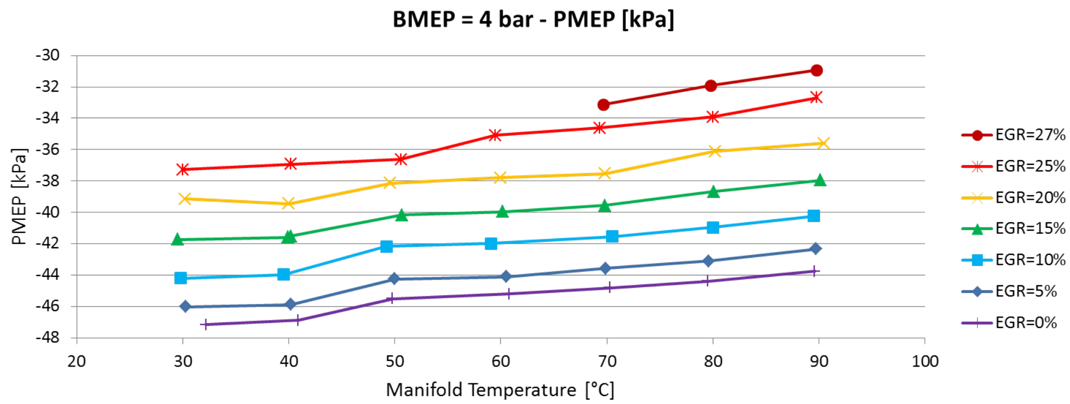


Figure 8: *Effects of manifold temperature and EGR rate on PMEP - BMEP = 4 bar*

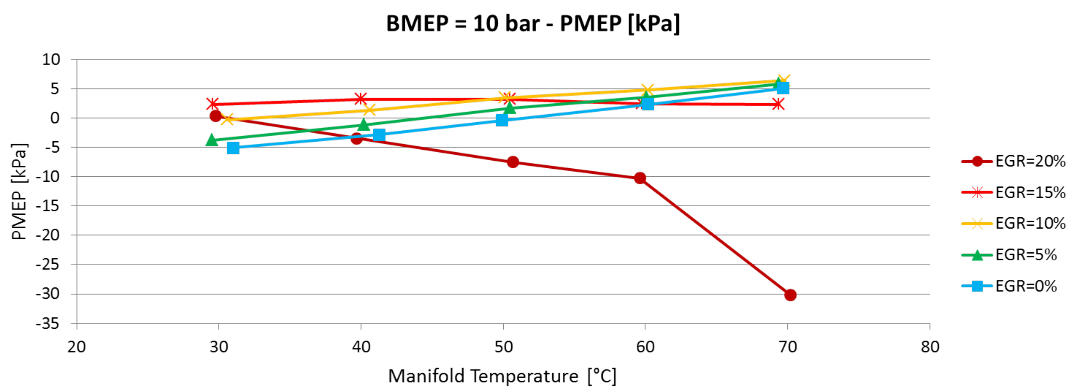


Figure 9: *Effects of manifold temperature and EGR rate on PMEP - BMEP = 10 bar*

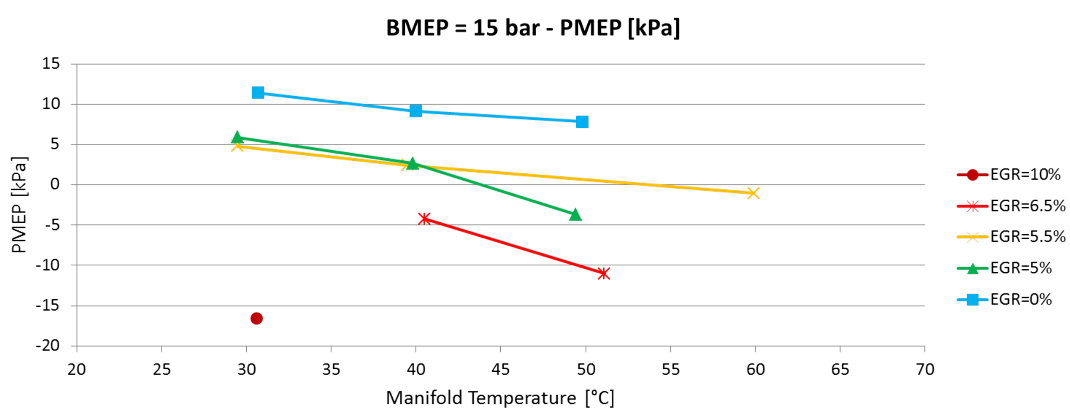


Figure 10: *Effects of manifold temperature and EGR rate on PMEP - BMEP = 15 bar*

Figure 10, the trend is no more highlighting a reduction of the BMEP as the temperature or EGR is increase, but the trend is actually opposite. This could be explained considering that, to have such an high amount of boost, the turbine is creating a big amount of back pressure. This will result in an increase of the pumping loop area. However, it must be said that, despite this increase in the PMEP, the positive effect of having a turbocharged engine are still appreciable in terms of power output and fuel consumption reduction.

Appendix C:
Combustion phasing characteristics

The combustion phasing for the different load conditions are presented in the followings. The main parameters plotted are the CA50, for those load conditions not present in the main body, the ignition delay and the combustion duration.

Ignition delay is defined as the crank angle degrees between the ignition event and the start of the combustion. To be more precise, the parameter CA10 is used. It is the crank angle position where the 10% of the total heat release occurs. To determine the ignition delay the difference between the two crank angle positions CA00 and CA10 is used.

Combustion duration, similarly, is defined as the difference between CA10 and CA90, where CA90 is the crank angle position where the 90% of the total heat release occurs. Again, the values of this parameter, as well as the previous one, are not the exact crank angle positions, but are representing a crank angle variation. This is important, because knowing the engine speed it is possible to calculate the duration in time of these two events.

A deeper analysis of the effects of EGR rates and ACTs on the combustion phasing is beyond the scope of the project, and will not performed further. The data will be simply presented according to the different load conditions.

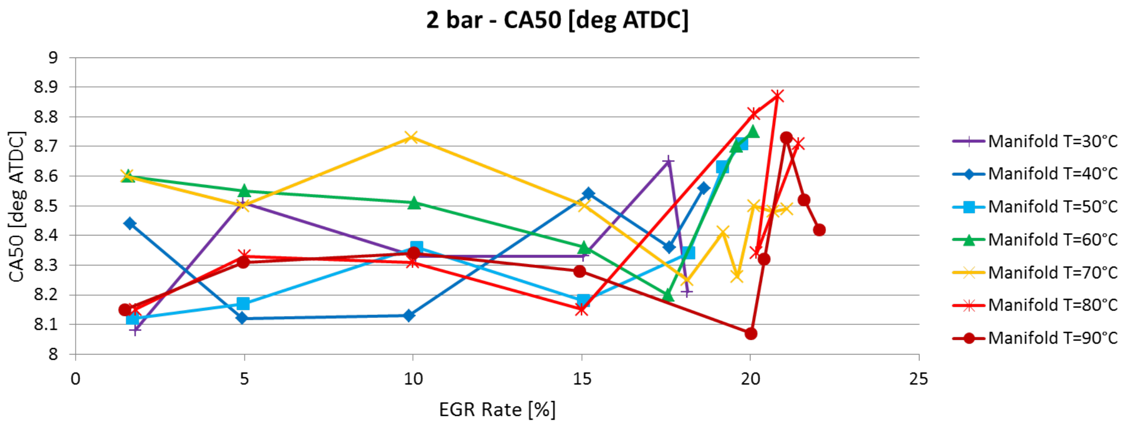


Figure 11: *Effects of EGR rate and ACT on CA50 - BMEP = 2 bar*

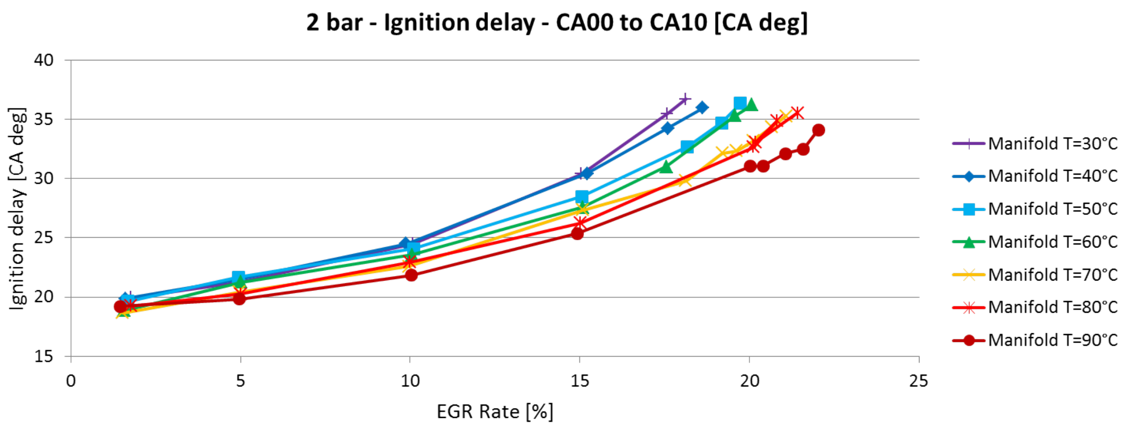


Figure 12: *Effects of EGR rate and ACT on ignition delay - BMEP = 2 bar*

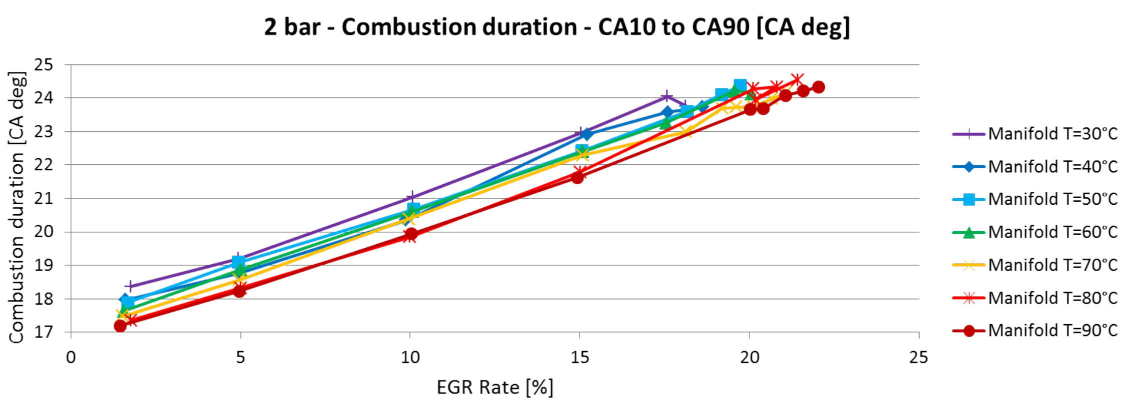


Figure 13: *Effects of EGR rate and ACT on combustion duration - BMEP = 2 bar*

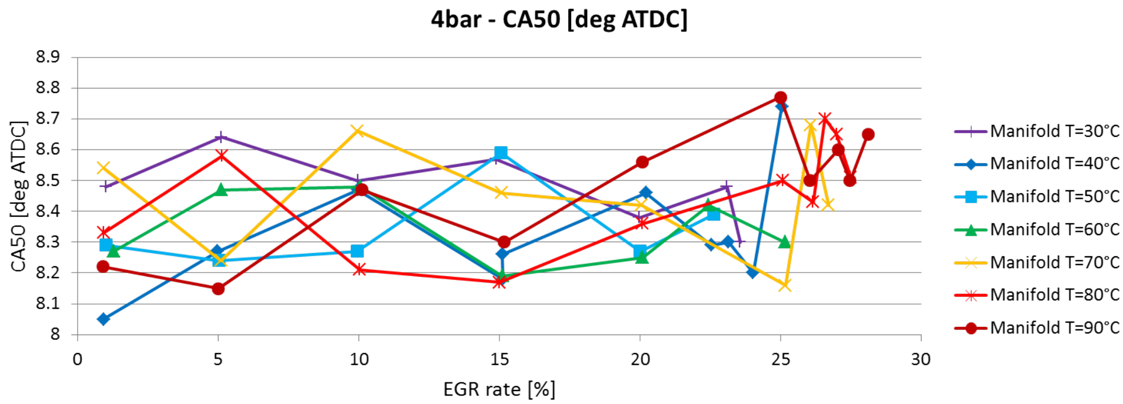


Figure 14: *Effects of EGR rate and ACT on CA50 - BMEP = 4 bar*

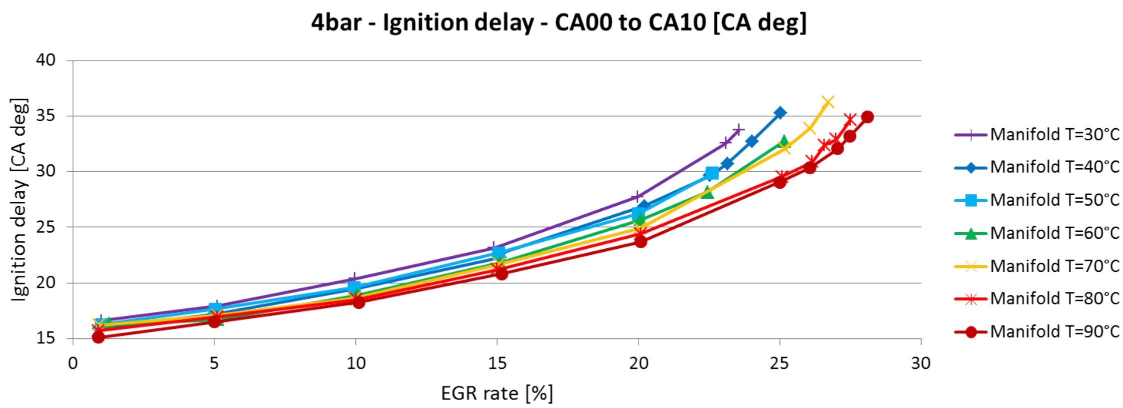


Figure 15: *Effects of EGR rate and ACT on ignition delay - BMEP = 4 bar*

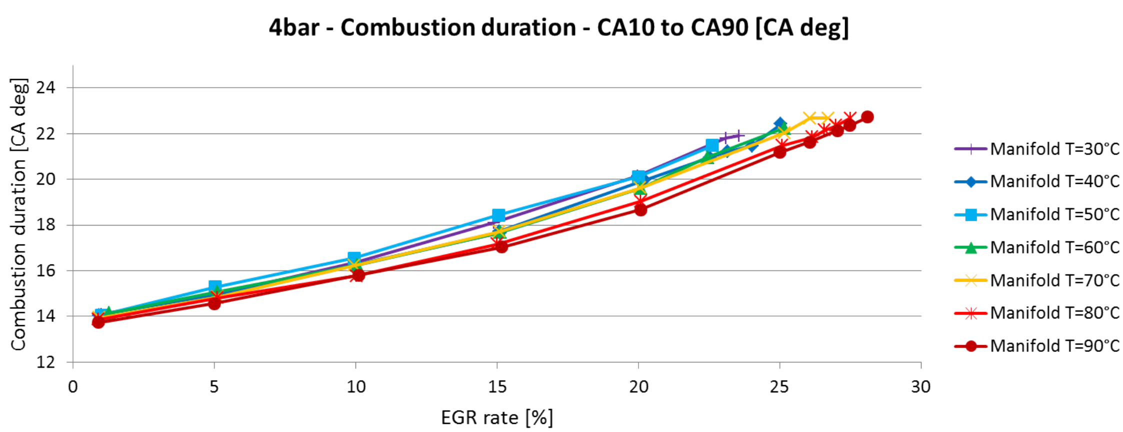


Figure 16: *Effects of EGR rate and ACT on combustion duration - BMEP = 4 bar*

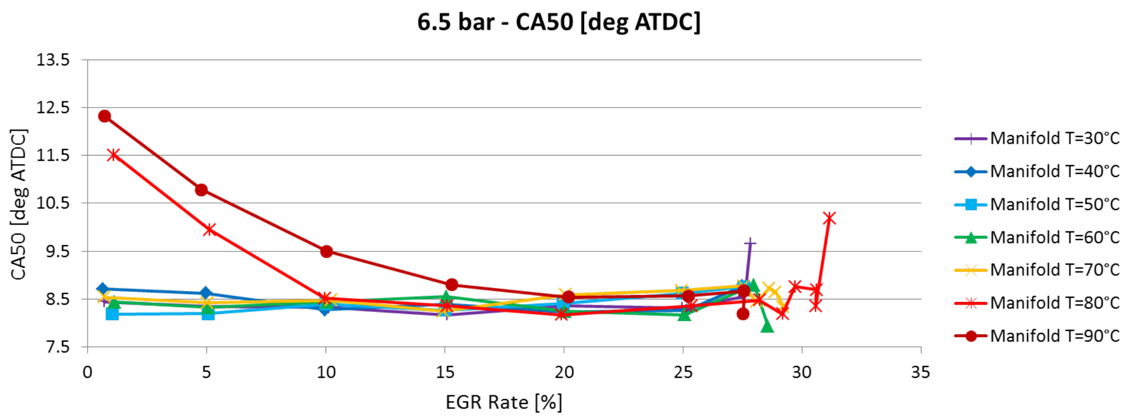


Figure 17: *Effects of EGR rate and ACT on CA50 - BMEP = 6.5 bar*

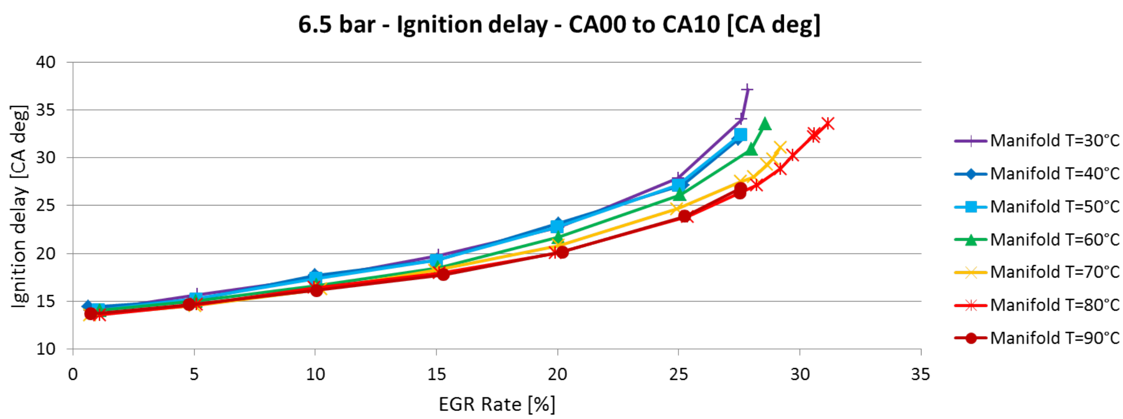


Figure 18: *Effects of EGR rate and ACT on ignition delay - BMEP = 6.5 bar*

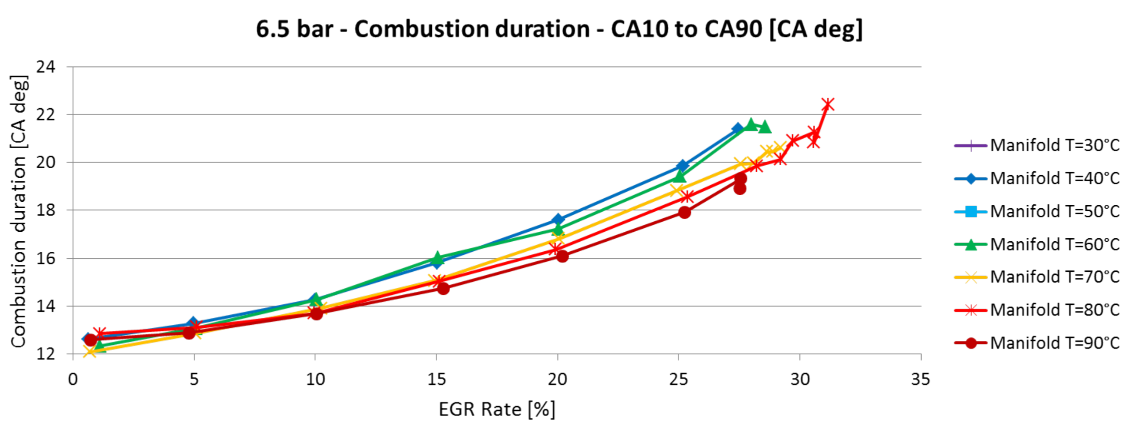


Figure 19: *Effects of EGR rate and ACT on combustion duration - BMEP = 6.5 bar*

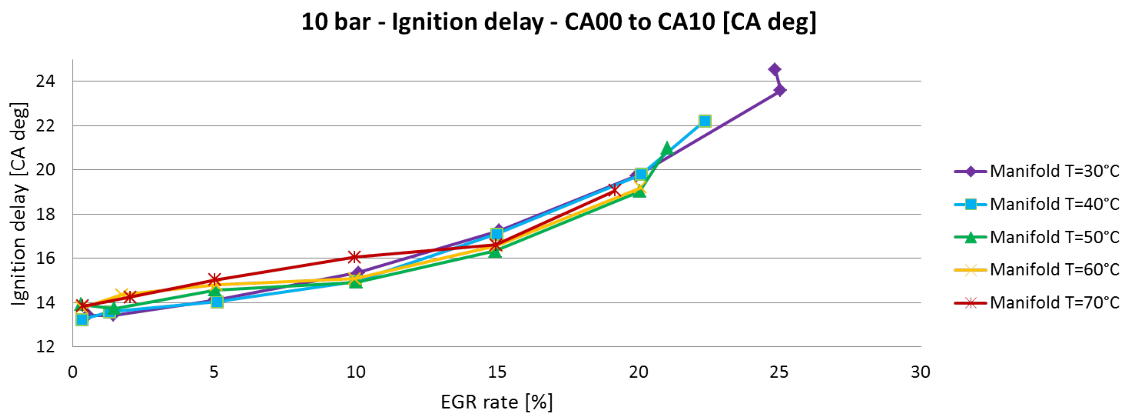


Figure 20: *Effects of EGR rate and ACT on ignition delay - BMEP = 10 bar*

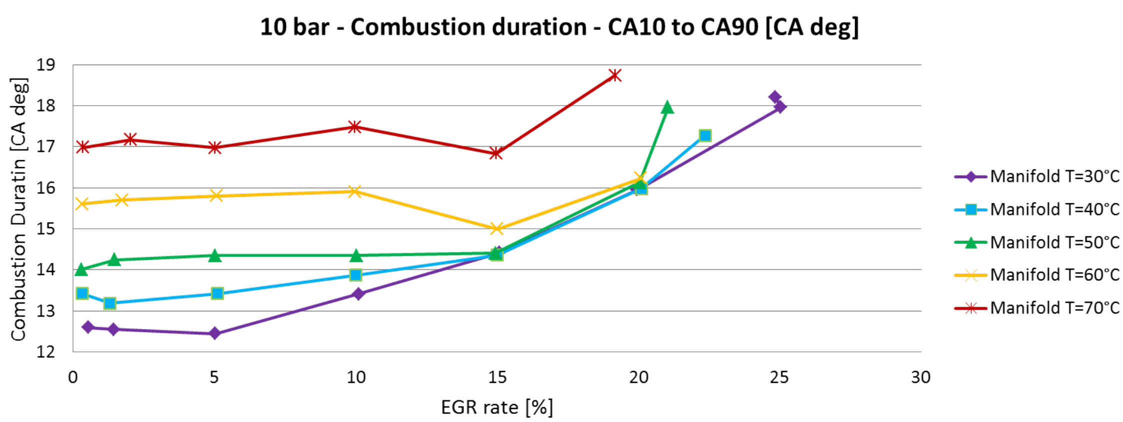


Figure 21: *Effects of EGR rate and ACT on combustion duration - BMEP = 10 bar*

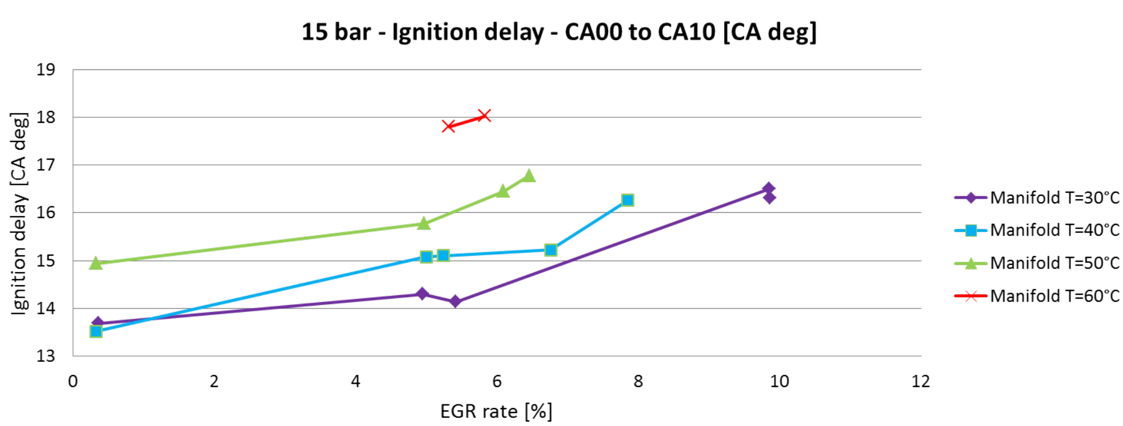


Figure 22: *Effects of EGR rate and ACT on ignition delay - BMEP = 15 bar*

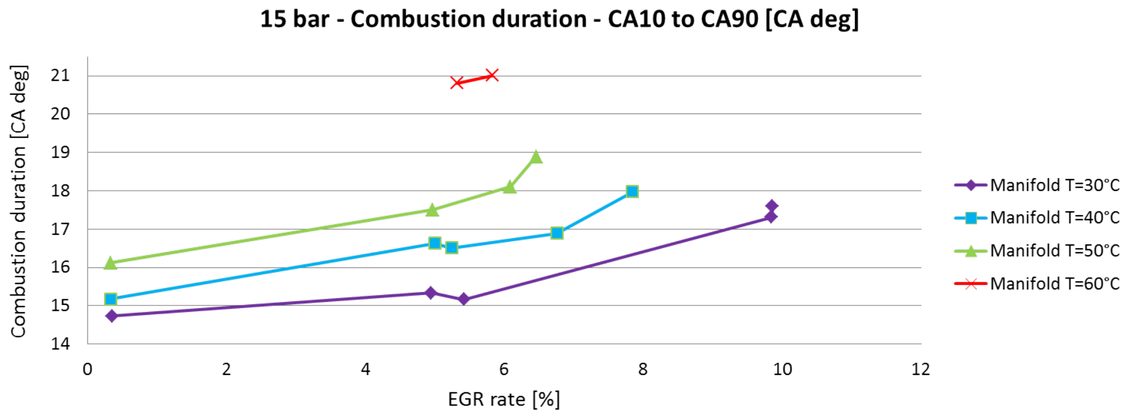


Figure 23: *Effects of EGR rate and ACT on combustion duration - BMEP = 15 bar*

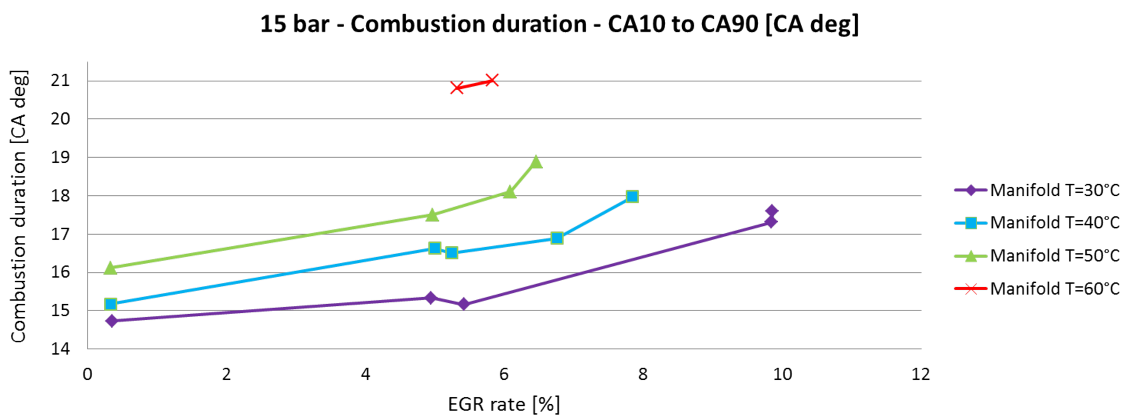


Figure 24: *Effects of EGR rate and ACT on ignition delay - BMEP = 17.34 bar*

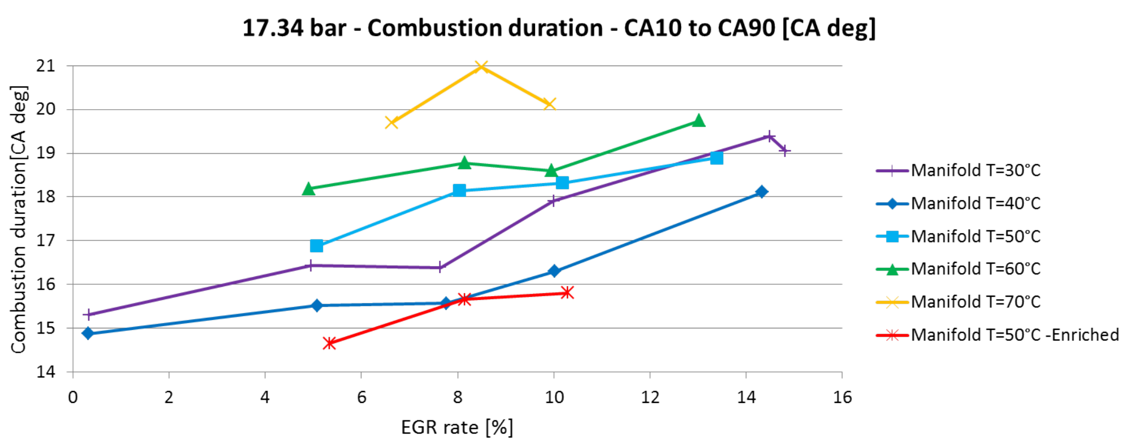


Figure 25: *Effects of EGR rate and ACT on combustion duration - BMEP = 17.34 bar*

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