Effect of boosting system architecture and thermomechanical limits on diesel engine performance. Part-I: Steady State Operation

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Abstract

Internal combustion engines developments are focused on efficiency optimization and emission reduction. To achieve these, downsized or downspeeded engines are required which can reduce fuel consumption and CO2 emission. However, these technologies ask for efficient charging system. This paper consists of study of different boosting architectures (single stage and two stage) with combination of different charging system like superchargers, e-boosters etc. A parametric study is carried out with a 0D engine model to analyze and compare different architectures on same base engine. The impact of thermomechanical limits, turbo sizes and other engine development options characterizations are proposed to improve Fuel consumption, maximum power and performance of the downsized/downspeeded diesel engines.

1 Introduction

The potential of new emerging turbocharging architectures to enhance the performance of downsized and down speeded engines has taken a crucial part. Upcoming new emissions test cycles are much more demanding with high EGR rates and transients. Moreover, turbocharger size, thermomechanical limits have also important consequences on engine performance and their impact have to be characterized to quantify possible benefits modifying their values. This paper focuses on a comprehensive study with the 0D engine model to respond to these specific objectives. A sophisticated model that includes a 0D phenomenological combustion model (combustion process) (*1*) and a 0D filling and emptying model (multi-cylinders and manifolds) was then developed to achieve the model complexity required by this study.

This study is divided into two papers, first part consist of the analysis of the engine and boosting systems performance under steady-state operations along with the hypotheses that have been assumed accordingly. (The results obtained with the main turbocharger will thus be reported before those obtained in two-stage operations). Following the future needs in charger development, the operating ranges required by downsized-down speeded engines will be confronted to conservative supercharger, compressor and turbines characteristics maps. At last, the transient aspects will be considered with an analysis of the boosting architectures performance on different downsized engines during cold transient test cycles.

2 Modelling and Methodology

The OD model has been created with Matlab considering several degrees of engine downsizing. So the engine scaling process based on a similarity approach is carried out. Finally, the other hypotheses made on the input data relating to the gas path elements, injections settings and EGR systems. Three passenger car Diesel engines have been involved in the characteri- zation and validation work. The first two engines (referred as Engine A and B) have been designed by the French manufacturer PSA under the Euro IV emissions regulations, while the third one (Engine C) has been more recently designed by Renault and respect the Euro V regulations.

3 Boosting Architectures

It has been highlighted the most promising boosting systems to increase the performance of automotive downsized-downspeeded engines are sequential serial two-stage turbocharging, me-

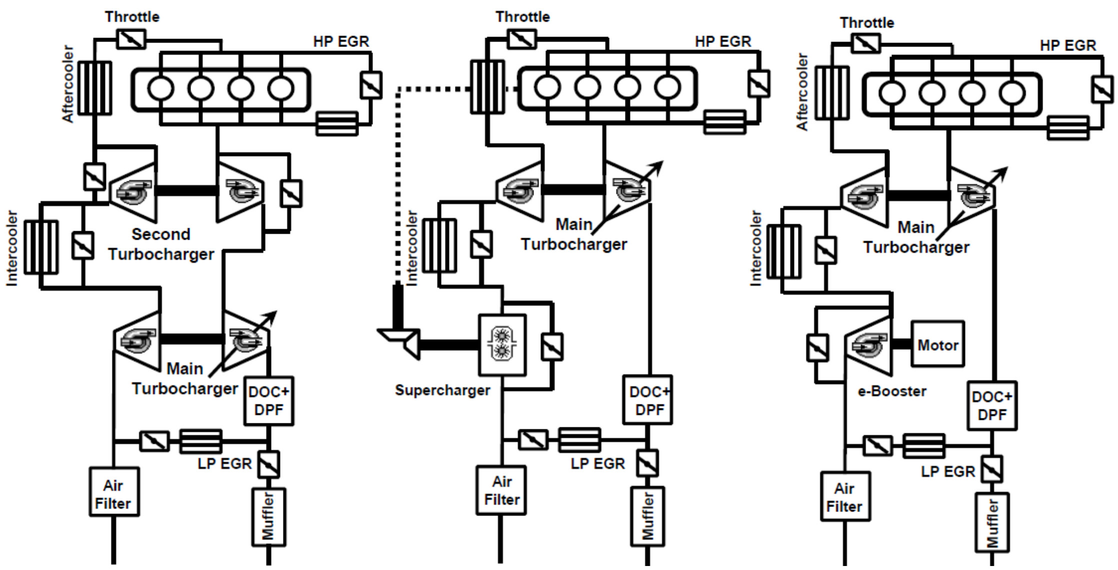

Figure 1: Two-Stage Boosting architectures

chanical auxiliary supercharging and electric booster. (*17*) These architectures have thus been analyzed in this chapter and a schematic of each one of them can be observed in figure 1. All architectures are composed of a main turbocharger fitted with a variable geometry turbine, a HP and LP EGR circuit equipped with their corresponding valves and cooler, an intake throttle to forced HP EGR mass flows when necessary, an air filter, an after treatment system and a muffler. To cool the intake gas, an aftercooler is positioned before the intake manifold. An additional intercooler can also be employed between both stages to perform an extra cooling through the control of a bypass valve. In the serial two-stage turbocharging system, the second turbocharger is fitted in the HP stage with a fixed geometry turbine while in the other systems, the mechanical supercharger and the e-Booster are placed in the LP stage. Finally in each configuration a bypass valve is arranged around the second charger to avoid parasitic losses in single-stage operations (sequential mode).

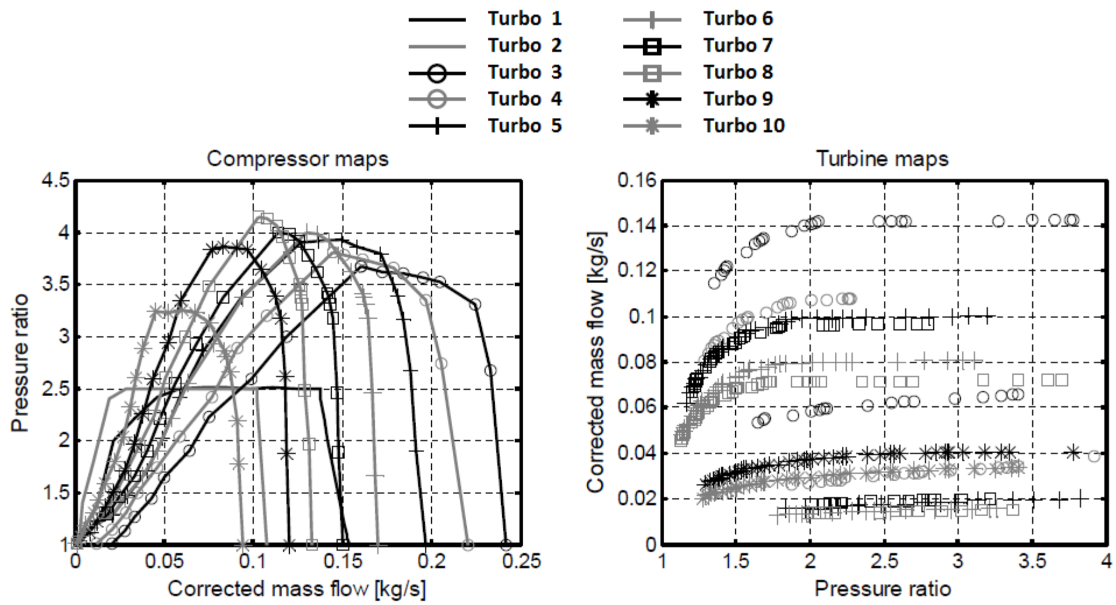

Figure 2: HTT turbocharger family and small Eatons superchargers map

3.1 Turbochargers

The information relating to the turbochargers comes from characteristics maps measured in turbocharger test benches (*?*). These data correspond to specific compressor and turbine designs which can be optimized for each application to achieve particular objectives. In the automotive market, a wide range of turbocharger designs are present and no map generalization can be made to perform global parametric stud- ies. As shown in figure 2 where compressors and superchargers operating ranges have been plotted, the maps from an entire turbocharger family can give information about the actual technological limits. But both surge line and over speed limits (respectively right limit and left limit of the compressor maps) are too dependent of the installation and measuring methods (*8*) (*9*) (*16*) to be assumed as strict limiting factor in the calculations.

Efficiencies are also strongly dependent of wheels designs and important variations can be observed between different turbochargers with similar operating ranges. That is why in this

		Main Turbocharger		Second Stage	
		Compressor	Turbine	Compressor	Turbine
Single-Stage	State-of-art	70	65	-	-
	Hypothesis	80	75	-	-
Two-Stage Turbocharger	State-of-art	70	65	67	65
	Hypothesis	80	75	77	75
Two-Stage Supercharger	State-of-art	70	65	65	-
	Hypothesis	80	75	75	-
Two-Stage eBooster	State-of-art	70	65	67	-
	Hypothesis	80	75	77	-

Figure 3: Charger efficiencies used in the steady state calculations

study, particular characteristic maps have not been used in the steady-state calculations and an energetic approach has been preferred. This energetic approach avoids design influences assuming infinitely large turbocharger operating ranges and making some hypotheses on the efficiencies. The charger efficiencies used in steady state calculations are showed in figure 3

3.2 Gas Path Elements

Pressure losses in the intake and exhaust lines elements have important impacts on engine and boosting architecture performance. Their characteristics are mainly dependent of mass flow rate and component design. The selection of the engine elements is specific to each application and responds to a delicate balance between pressure drops, packaging constraint, efficiency to fulfill the component function, cost, etc. . . So, an energetic approach has also been considered for the engine components to generalize their pressure losses characteristics to the different engine displacements and rated power levels (maximum mass flow)

Pressure losses measured under full load conditions in the air filter, aftercooler, and muffler and after treatment system of Engine C (mentioned in methodology)are shown in table 4. As similar drops have also been measured on the engines A and B, especially for the aftercooler

Engine speed	Air filter	Aftercooler	Muffler	DOC+DPF reference	DOC+DPF large capacity
1000 rpm	11	10	10	38	19
1250 rpm	13	16	18	74	37
3000 rpm	73	72	117	468	234
3500 rpm	98	101	160	644	322

Figure 4: Pressure losses (mbar) in gas path elements under full load conditions

and muffler, the same data have been considered independently of the mass flow rate. This hypothesis amounts to scaling the pressure losses characteristics for each application in order to maintain the same component influences in the simulations. A picture of this hypothesis is given in figure 5 where it can be seen how the reference pressure losses characteristic is adapted to the considered maximum gas mass flow.

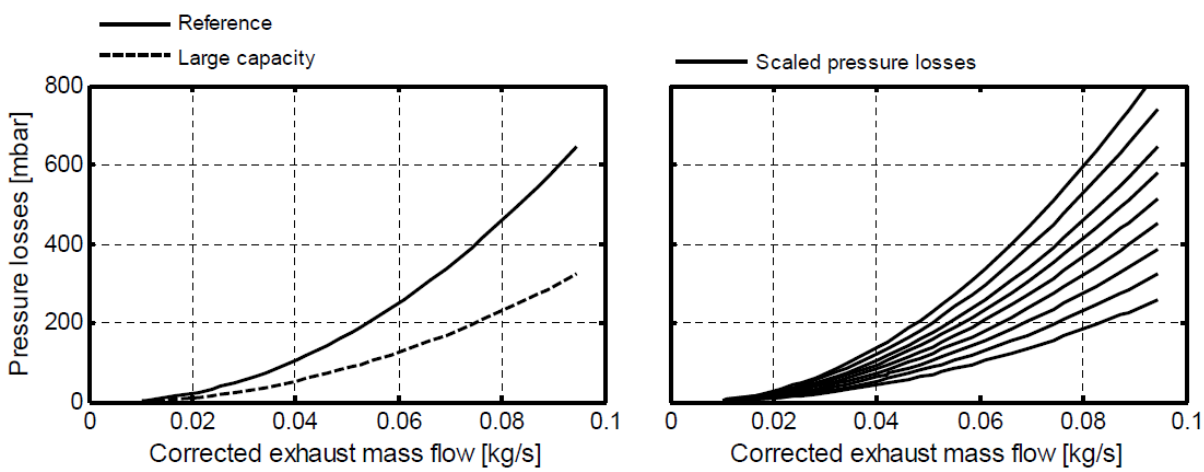

Figure 5: scaled pressure losse characteristics in aftertreatment system

The after treatment system is the engine component that involves the higher pressure drops. To analyze the performance sensitivity of its design, a large capacity system producing only half losses has also been considered. For the charge air coolers, the same pressure losses characteristics have been employed for the intercooler and aftercooler and NTU models have been replaced by ideal cooling efficiencies (external cooling fluid temperatures of 35°C).

3.3 Injection Setting

To limit the number of parameters, the injection process has been reduced to a unique main injection without any pre- or post-injections. The injection timings have been optimized to maximize the IMEP (minimum specific fuel consumption) or to respect the maximum allowable cylinder pressure. At 1000 rpm and 1250 rpm, the relative fuel-to-air ratio has been fixed to 0.9. This value represents a typical maximum fuel-to-air ratio allowed by smoke limiters. While at 3000 rpm and 3500 rpm, a fuel-to-air ratio of 0.7 has been retained to limit exhaust manifold temperatures. This lower fuel-to-air ratio obviously imposes a higher demand on the boosting system. That is why its value has been progressively increased up to 0.9 when turbine inlet pressure or compressor outlet temperature becomes a limiting factor.

3.4 EGR System

Low Pressure and High Pressure EGR systems have been analyzed under three different EGR rates: 0% (without EGR), 15% (Euro VII objectives) and 30% (strong EGR constraint). In the coolers, ideal efficiencies have been employed with external cooling fluid temperatures of 90°C. Their pressure losses have been fixed in the calculations at 3 mbar at 1250 rpm, 18 mbar at 3000 rpm and 25 mbar at 3500 rpm. EGR performance has not been considered at 1000 rpm as no emissions test cycle requires EGR under full load at that speed. For the other gas path components, two hypotheses have been assumed on their pressure losses characteristics. On the one hand, the same pressure drops have been used between the three different EGR rates scaling the elements characteristics for each running operation. In that way, as LP EGR involves higher gas mass flows in the intake/exhaust lines, bigger charge air coolers and after treatment system effective sections are considered for LP EGR operations. On the other hand, the same pressure losses characteristics have been employed under LP and HP EGR rates scaling the characteristics for the LP EGR mode. In that case, the same elements are considered between both modes and pressure drops in charge air coolers as after treatment system result lower in the HP EGR mode. In the following section, this second hypothesis is labeled HP EGR low dP.

4 Steady State Results

As the main objective of these simulations was to characterize the boosting system and the thermomechanical limits affecting the maximum reachable brake power. Hence The operating conditions have been defined as a function of brake power objectives increasing brake power until reaching one of the thermomechanical limits. With the energetic approach, the different engine components are directly matched to the considered brake power level so that the obtained results correspond to the optimized configurations.

To compare the different architectures and to analyze the influences of the considered design factors, the Brake Specific Fuel Consumption (BSFC) has been retained. Generally under full load conditions, BSFC is not so important because the current passenger cars emission test cycles dont include these running conditions. But this parameter becomes relevant for future engine development as the new emission test cycles integrate more and more highly loaded operations. Furthermore here, BSFC has been selected to quantify in each study the overall system efficiency taking into account not only the engine or the boosting architecture performance but also all the systems interactions. The BSFC allows therefore to evaluate the impact of each parameter from a global point of view such as the brake thermal efficiency.

For the thermomechanical limits, two levels of maximum compressor outlet temperature have been defined, one at 190°C and one at 210°C. The first level corresponds to the old part in turbocharger and intake line development, while the second represents the maximum allowable working temperature for cast aluminum alloy compressor wheels. This second level does

Engine speed	State-of-art	Hypothesis
1000 rpm	130 bar	150 bar
1250 rpm	150 bar	180 bar
3000 rpm	170 bar	190 bar
3500 rpm	170 bar	190 bar

Figure 6: Maximum incylinder pressure used in the simulations

not involve major modifications in compressor wheel design but requires advanced plastic materials for the intake piping. Although turbine inlet pressures have also been limited at 4.5 bar, maximum compressor outlet temperatures have always been a more restrictive factor in the calculations. Here, exhaust temperatures have not been constrained in order to define new maximum temperature requirements.

4.1 Effects of Maximum Allowed In-cylinder Pressure

For maximum in-cylinder pressures, two levels have been analyzed: one corresponding to the past in engine development and one considering future thermomechanical limits evolutions (*19*) (*13*). These limits, which depend on engine speed, are defined to ensure that oscillating gas force loads do not exceed the material fatigue strength in bearing and cylinder head top desk areas. The considered values are shown in table 6 while the performance results are plotted infigure 7

As it can be observed in figure 7, the BSFC presents a trend that firstly decreases and then increases as a function of brake power level. This trend is explained by both combustion velocity and injection timings. In fact, increasing the brake power level increases the charge density in the combustion chamber accelerating the RoHR and improving the combustion efficiency. However, when the maximum in-cylinder pressure is reached, injection timings are retarded and combustion efficiency decreases.

A higher maximum in-cylinder pressure moves therefore the point of minimum BSFC to higher BMEP and reduces the BSFC at high BMEP. At low speeds with moderate BMEP ob-

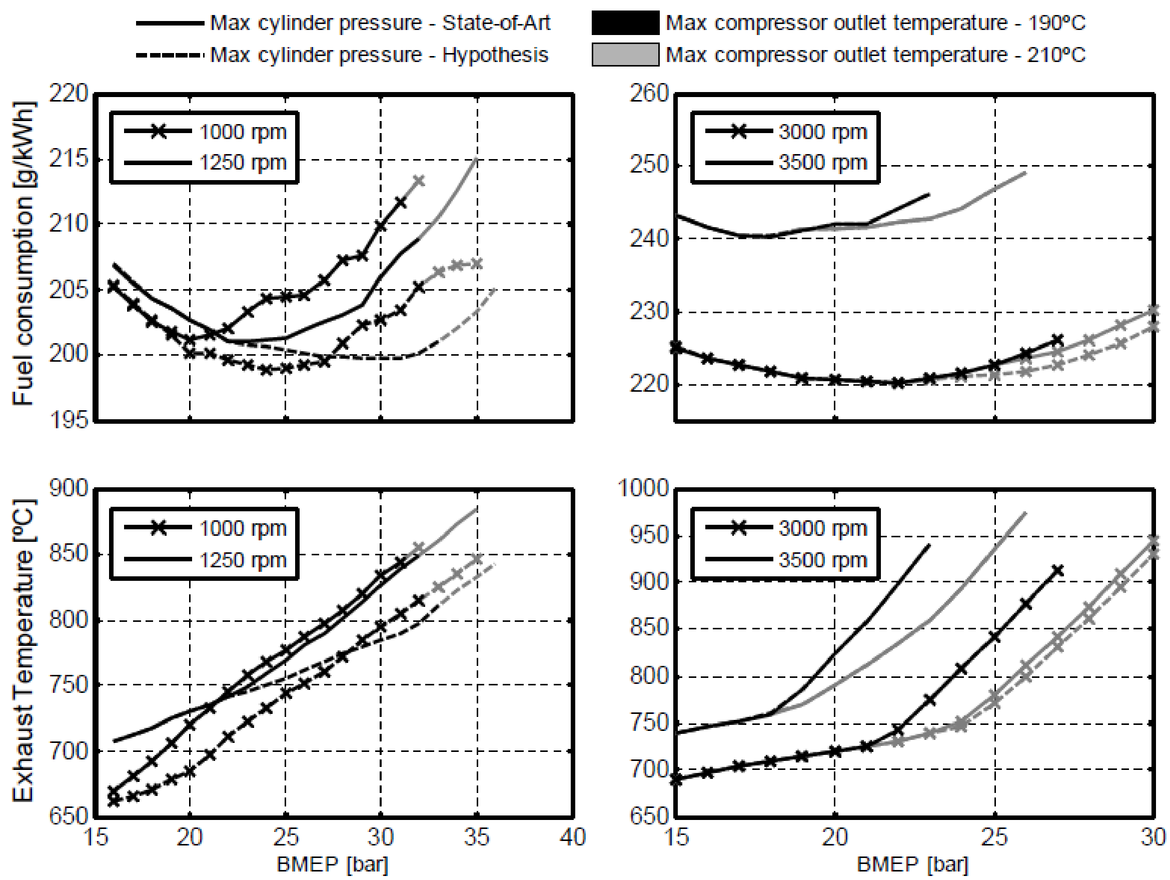

Figure 7: Impact of maximum in-cylinder pressure and maximum compressor outlet temperature on engine performance as a function of brake power levels

jectives (around 20 bar), there are no benefits to increase the actual state-of-art limits. But for strong BMEP objectives (around 30 bar), fuel savings up to 7 g/kWh can be obtained. Exhaust temperatures rise more or less linearly with the brake power level. Increasing the maximum in-cylinder pressure allows also the reduction of the temperature constraints at high BMEP limiting the need to retard injection timings.

At 3000 rpm and 3500 rpm, the variation of fuel-to-air ratio is an additional factor affecting the BSFC. The change of trend noticed on the exhaust temperature shows how the fuel-to-air ratio is gradually increased to respect the maximum compressor outlet temperatures. A relatively low fuel-to-air ratio requires a higher compression work but reduces the thermal constraint in the exhaust. It also increases the charge density and oxygen concentration in the combustion chamber. As already explained a higher charge density can improve or deteriorate the BSFC, while a higher oxygen concentration always increases the combustion velocity and the corresponding combustion efficiency. The impact of lower fuel-to-air ratio on BSFC is therefore a balance between boosting systems losses and combustion benefits which mainly depends on the in-cylinder pressure limit. This balance is generally positive until the injection timings need to be delayed. At 3500 rpm, the in-cylinder pressures do not reach the state-of-art pressure limits. So, the fuel consumption increases from the moment when fuel-to-air ratio rises. The same effect is observed at 3000 rpm with the 190?C limit at the compressor outlet. With the 210?C limit, the fuel consumption increases before modifying the fuel-to-air ratio as the higher charge density requires some injection timings delays. Nonetheless, these injection timings delays are relatively small and generate only resultant fuel penalties of 2 g/kWh. From these considerations, at 3000 rpm and 3500 rpm the differences in BSFC are therefore mainly explained by fuel-to-air ratio variations and the small benefits observed at 3000 rpm do not justify an increase of the current state-of-art in-cylinder pressure limits at rated speeds. In terms of maximum BMEP, the maximum allowable compressor outlet temperature always limits cylinder charge densities before exceeding the maxi- mum in-cylinder pressure at the end of the compression stroke. Extending the thermal limit from 190?C to 210?C allows to increase the maximum BMEP of around 3 bar at rated speeds and between 1 bar and 3 bar at low speeds. At low speeds, similar benefits are also obtained increasing the maximum in-cylinder pressures due to higher combustion efficiencies (more centered injection timings). Maximum in-cylinder pressures appear therefore as indirect limiting factors. These results are obviously dependent of the cylinder compression ratio. If a higher value is retained, the impacts observed on the BSFC will be more marked but the main trends will remain and the curves will be only shifted to lower BMEP. Finally comparing running operations performed at 3000 rpm and 3500 rpm, the effectiveness of the downspeeding technique to reduce fuel consumption can be noticed with differences up to 20 g/kWh between both considered rated speeds. Exhaust temperature constraints stay as for them relatively constant.

4.2 Effects of Exhaust Back Pressure

The influences of engine components pressure losses characteristics on engine and boosting system performance are shown in figure 8. Having higher pressure drops, a sensitivity study has been performed on the aftertreatment system considering a reference and a large capacity design as previously de- scribed. With both designs, it can be observed that elements pressure characteristics have minor impacts at 1000 rpm and 1250 rpm because gas mass flow as pressure drops are relatively small at these speeds. However at high engine speeds, their impacts have important consequences on the BSFC. In fact here it can be noticed how pressure losses differences of 234 mbar and 322 mbar between both designs at 3000 rpm and 3500 rpm offset the BSFC of around 5 g/kWh and 10 g/kWh respectively. In addition, the large capacity design in- creases the maximum reachable BMEP of 1 bar decreasing the exhaust thermal constraints of around 30°C at both rated speeds. The optimization of elements pressure characteristics is therefore fundamental to improve in the medium to high speed range the fuel consumption of downsized-downspeeded engines.

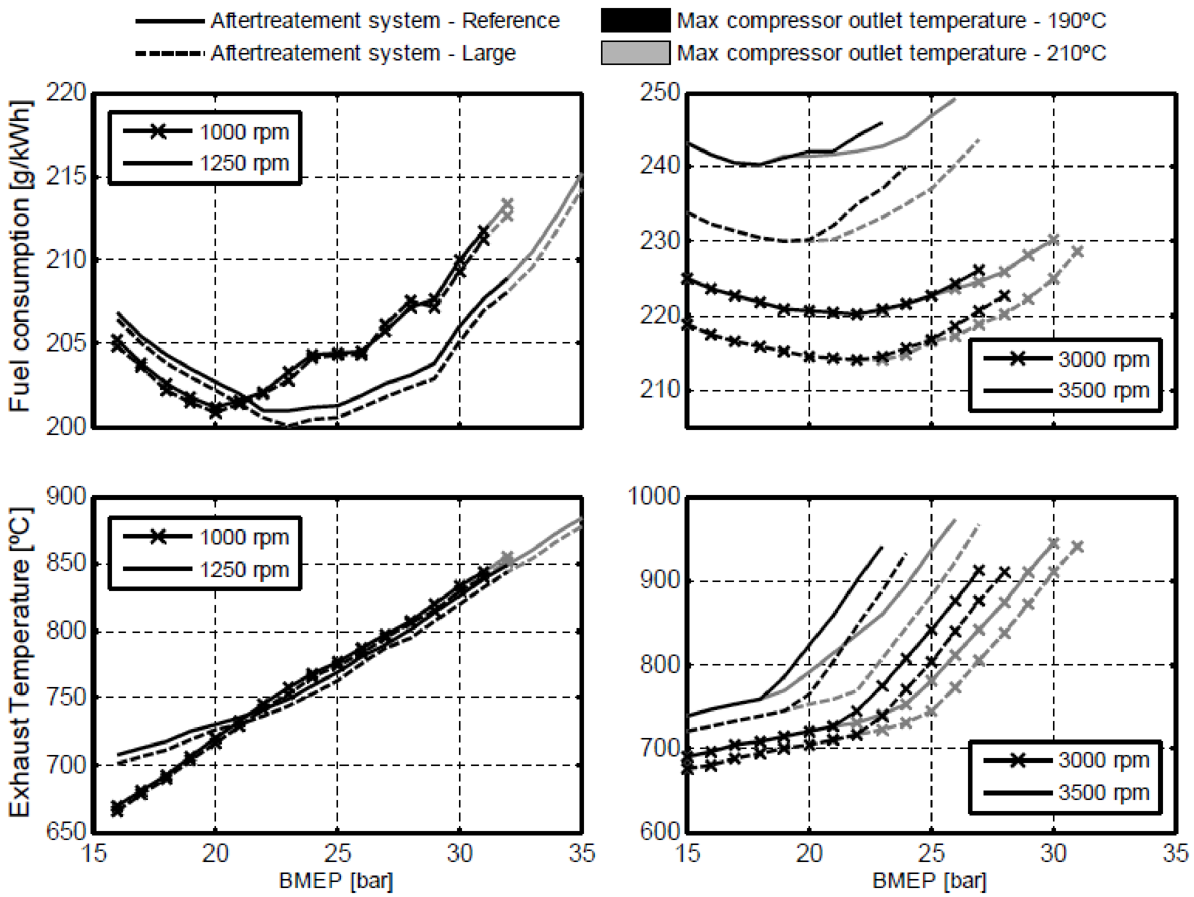

Figure 8: Impact of pressure drops across the aftertreatment system and maximum compressor outlet temperature on engine performance as a function of brake power levels.

4.3 Effect of Turbocharger Efficiency

For the influences of turbocharger efficiencies on engine and boosting system performance, different hypotheses have been assumed to fix state-of-art levels before considering variations of 10 points on both compressor and tur- bine efficiencies. As it can be observed in figure 9, these important efficiency variations have limited consequences on the BSFC at low speeds reaching fuel savings of only 2-3 g/kWh at 1000 rpm and 1250 rpm. However at rated speeds, their impacts are much more significant achieving BSFC reductions of around 5 g/kWh and 10 g/kWh at 3000 rpm and 3500 rpm respectively. These reductions are similar to those obtained with the large capacity after treatment system. That means, optimizing the elements pressure characteristics can bring the same BSFC benefits as increasing by 10 points the turbocharger efficiencies. In terms of maximum BMEP, compressor outlet temperatures are highly dependent of turbocharger efficiencies and variations of 10 points allow to increase the maximum BMEP of around 3-4 bar in the whole engine speed range.

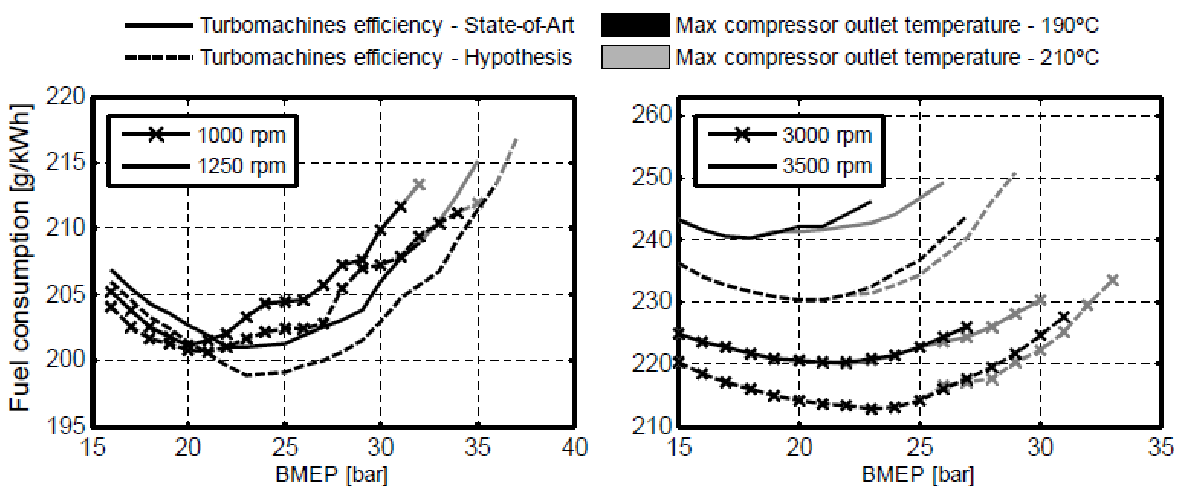

Figure 9: Impact of turbocharger efficiencies and maximum compressor outlet temperature on engine performance as a function of brake power levels.

These results also demonstrate that the conclusions obtained with this energetic approach can be generalized to similar downsized-downspeeded engines. In fact, efficiency hypotheses have been established with a turbocharger size corresponding to a 2.3l engine. But it has been shown maximum efficiency variations do not exceed 3 points for the compressor and 5 points for the tur- bine when smaller turbochargers and smaller engine displacements are considered (1.2l-1.6l engines). These efficiency variations are relatively limited when compared to the variations performed in the sensitivity study. As efficiency variations mainly offset the performance results keeping identical trends, the same conclusions can be easily extrapolated to other turbocharger efficiencies and to other engine displacements.

4.4 Synthesis of Thermal Constraint

In order to analyze how the thermal constraints limit the engine performance, the maximum reachable BMEP obtained in the previous sensitivity studies have been plotted in figure 10 with several levels of maximum exhaust temperature. As the simulations are not limited by turbocharger operating ranges, it can be noticed that maximum BMEP are higher at low speeds than at rated speeds. This is mainly explained by lower gas path pressure losses and lower friction plus auxiliaries mechanical losses suffered at reduced speeds. Between both considered rated speeds, the higher losses suffered at 3500 rpm offset the brake power benefits implied by a higher speed and both downspeeding levels achieve similar maximum engine powers. Regarding the different component optimization scenarios, turbocharger efficiencies and maximum in-cylinder pressures involve the major BMEP variations at low speeds. While at high speeds, the major BMEP variations are produced by turbocharger efficiencies and element pressure characteristics.

These results have been obtained limiting directly the maximum outlet compressor temperature in the calculations. Taking into consideration the exhaust thermal constraints, it can be seen the maximum exhaust temperature is much more restrictive than the maximum outlet compressor temperature. In fact, the allowable exhaust temperature must be higher than 850°C at low speeds and higher than 950°C at rated speeds so that the maximum outlet compressor temperature becomes the limiting factor. A high exhaust temperature limit is therefore a fundamental requirement to increase the performance of downsized-downspeeded engines. Due to torque limitations in vehicle trans- mission, maximum BMEP objectives are generally constant

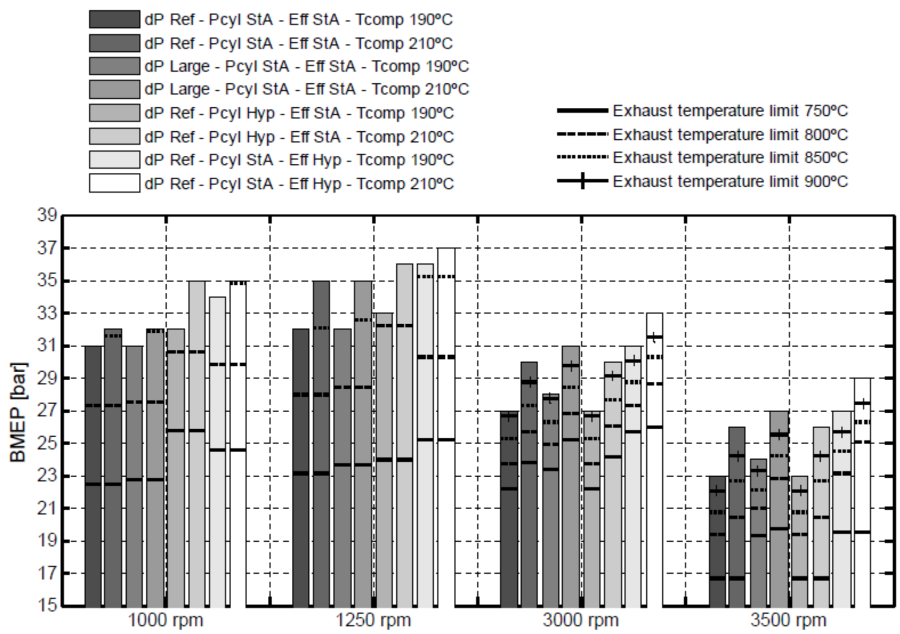

Figure 10: Influences of thermal constraints on maximum reachable BMEP for different component optimization scenarios.

between 1250 rpm and rated speed. Analyzing the results at iso-BMEP objectives, it can be noticed the exhaust temperatures are higher at rated speeds than at 1250 rpm despite the lower fuel-to-air ratio. The rated power represents thus the most critical running operation and exhaust temperature limitations must be rated at that point. Nowadays, exhaust temperature limitations vary between 750°C and 850°C according to the load duty cycle of each application. But exhaust manifolds and turbochargers able to withstand temperatures higher than 1050°C have already been developed for passenger car gasoline engines (*20*). Considering the exhaust constraints shown in figure 10, materials and turbocharger technologies used on gasoline engines are thus necessary to develop highly-rated Diesel downsized-downspeeded engines.

4.5 Effect of EGR Level

EGR requirements imposed by new emission test cycles have important consequences on the engine and boosting system performance. To analyze these consequences, a first sensitivity study has been performed on the EGR rate provided by the LP EGR circuit. The previous parameters (engine components pressure characteristics, turbocharger efficiencies and maximum in- cylinder pressure) have been maintained at their conservative or reference values. Low Pressure EGR has an impact on the combustion process, the turbocharger work and the gas path pressure drops. Here, with the hypotheses assumed on the pressure losses characteristics, the engine components are directly matched to the different LP EGR rates and gas mass flows. So the components pressures losses do not have any influence in this first EGR sensitivity study. Besides with the pressure drops retained for the air filter and muffler, the use of the second LP EGR valve placed at the muffler inlet has not been required in the calculations. For the combustion process, EGR increases the density in the combustion chamber but reduces significantly the oxygen concentration and the resultant combustion velocity. Combustion efficiency and fuel consumption are thus deteriorated with EGR. However, a slower combustion velocity decreases the in-cylinder pressure and requires lower injection delays to respect the in-cylinder pressure limitations. In that case, the more centered combustion obtained with EGR can improve the fuel consumption. This effect depends obviously on the hypotheses assumed for the injection settings and can be avoided using multi-injection strategies or defining other objectives for the injection timings optimization process. For the turbocharger, LP EGR increases the compressor gas mass flow and the required turbocharger work to provide a given boost. LP EGR increases also the gas mass flow passing through the turbine but this higher flow does not offset the higher compression work and turbine expansion ratio increases. Introducing EGR in the cylinders lowers gas temperature during the combustion process and reduces the available energy at the turbine inlet which further in- creases the turbine expansion ratio. LP EGR deteriorates therefore the fuel consumption due to higher engine pressure losses.

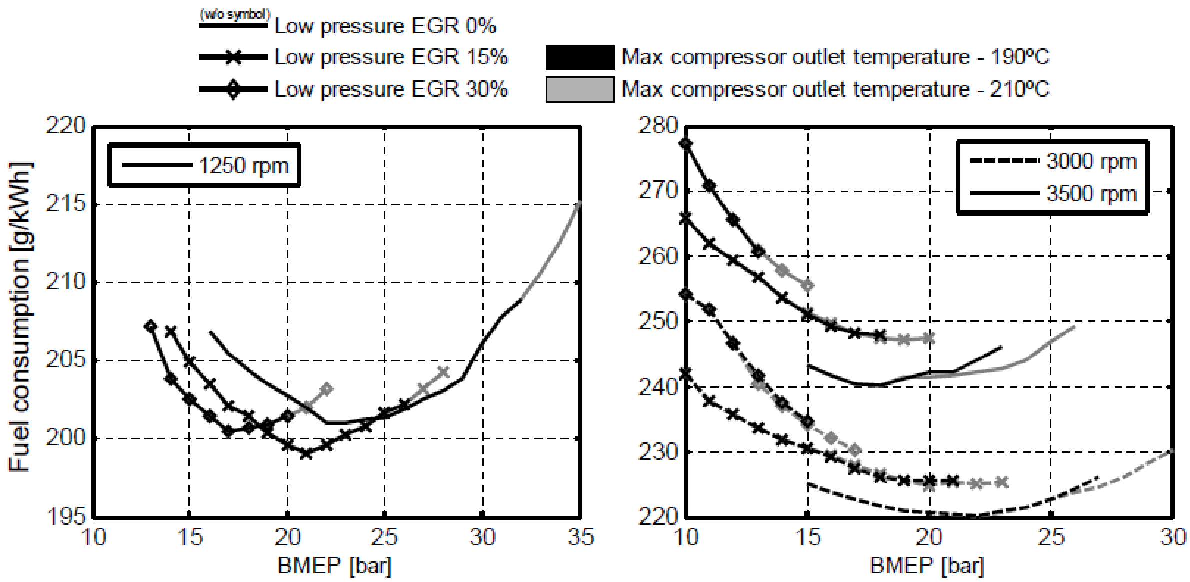

Figure 11: Impact of LP EGR rates and maximum compressor outlet temperature on engine performance as a function of brake power levels.

In figure 11, the balance of these different impacts can be observed for various LP EGR rates (0%, 15% and 30%). At 1250 rpm, the higher cylinder charge densities move the BSFC curves and the point of minimum fuel consumption to lower BMEP. With a LP EGR rate of 15%, the lower injection delays allow fuel benefits that largely compensate for the losses involved by higher turbine expansion ratios and BSFC are improved. With 30%, the combustion benefits just offset the backpressure losses and BSFC are relatively closed to ones obtained without EGR. In terms of maximum BMEP, even employing an ideal EGR cooler which corresponds to the most optimistic situation, the maximum compressor outlet temperature strongly limits the engine performance with decreases of 7 bar and 13 bar under LP EGR rates of 15% and 30% respectively. In two-stage architectures, these performance falls can be minimized dividing the compression work between the HP and LP stages and using an intermediate intercooler. But at low speeds, the main turbocharger has generally no ability to produce significant compression works forcing the boosting architecture operating only with the second charger. In these conditions, an intermediate intercooler does not present any potential to maintain or increase the engine performance. At 3000 rpm and 3500 rpm, increasing by 15% the LP EGR rates generates fuel consumption penalties from 5 g/kWh to 10 g/kWh. In fact, the in-cylinder pressure limitations have lower influences on the injection timings and the injection strategy does not bring any fuel benefits when working with EGR. The gas mass flows are also relatively important and the backpressure losses generated by higher turbine expansion ratios become significant. For the maximum BMEP, performance reductions from 5 bar to 7 bar can be no- ticed between the different EGR rates. These performance reductions cannot be minimized by an intermediate intercooler because the second charger is generally too small to provide boost at theses speeds.

4.6 Effect of EGR Architecture

With these results, a second sensitivity study has been carried out to analyze the impacts of the EGR circuit (High Pressure and Low Pressure) to provide different EGR rates (15% and 30%). The main differences between both EGR circuits lie in turbocharger works and intake temperatures. Under LP EGR, turbocharger works are more important due to higher gas mass flows passing through the intake/exhaust lines and intake temperatures are lower thanks to the aftercooler cooling process. Considering ideal aftercooler and EGR coolers, the intake temperature variations reach 8°C and 16°C under 15% and 30% EGR respectively. These temperature variations deteriorate the engine breathing process. Higher boosts are therefore necessary under HP EGR to admit the desired gas mass flows into the cylinders. As previously described for the pressure losses characteristics, two hypotheses have been assumed; one considering the same pressure drops between both systems (HP EGR) and one considering the same elements effective sections (HP EGR Low dP). The results of this analysis are shown in figure 12. Having the same trends, the 3500 rpm rated speed operations have not been represented here for the sake of clarity.

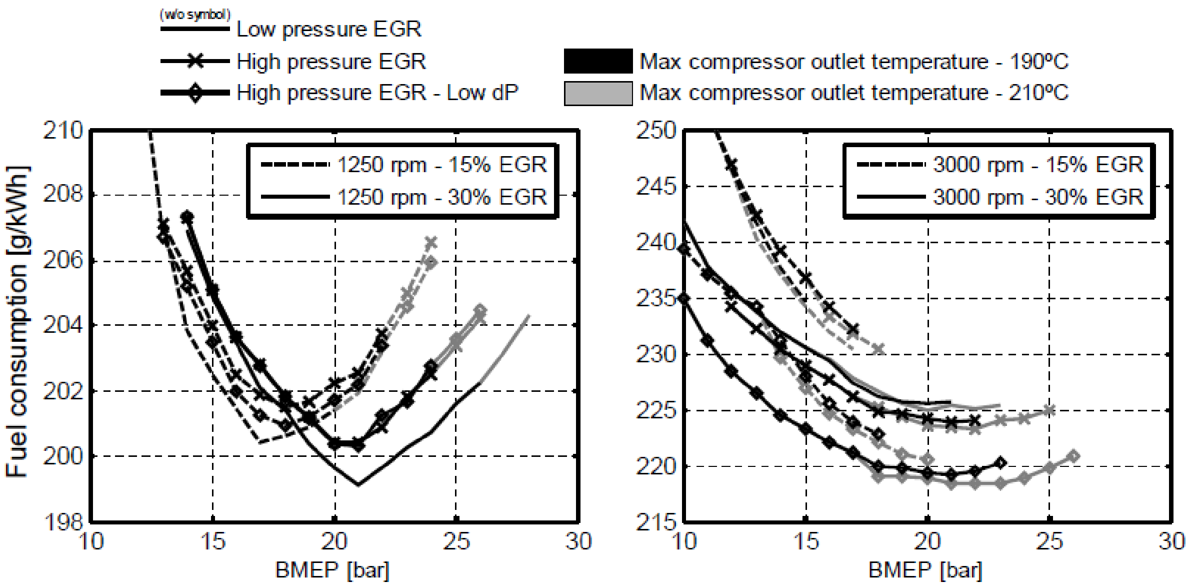

Figure 12: Impact of EGR rates, EGR systems and maximum compressor outlet temperature on engine performance as a function of brake power levels.

At 1250 rpm, the different hypotheses assumed on turbocharger efficiencies and element pressure losses forced to use the intake throttle to provide the 15% HP EGR rate. The pressure losses required in the intake line to increase the engine backpressures range from 50 mbar at 15 bar BMEP to 300 mbar at 25 bar BMEP. These losses imply higher compression ratios which increase fuel consumption and reduce maximum reachable BMEP by 2 bar. BSFC are thus higher with the HP EGR circuit. At 30% EGR, the intake throttle is no more required due to higher turbocharger works involved. But volumetric efficiency differences still imply higher boost demands for the HP EGR. As the benefits of lower turbocharger gas mass flows do not offset these higher boost demands, the HP EGR circuit stays less efficient. Nonetheless, with its lower compressor inlet temperatures, it allows to reach at this EGR rate 2 bar higher maximum BMEP. Regarding the HP EGR Low dP configuration, no significant differences are noticed at 1250 rpm between both HP EGR systems because the elements pressure losses are relatively small at that speed. At 3000 rpm, with identical turbocharger efficiencies and pressure losses, similar fuel consumptions are obtained between both LP and HP circuits. The impacts of different volumetric efficiencies are more or less offset by the influences involved by the different turbocharger gas mass flows. Slight benefits can thus be observed for the LP system at 15% EGR while at 30% EGR these benefits are reported for the HP system. However, when the same engine components are used in both circuits, fuel savings of up to 7 g/kWh can be noticed with the HP EGR low dP system. That means the elements pressure drops are the most influential factors when both circuits are compared and pressure losses characteristics are critical for the LP EGR system. Unless large capacity components are employed, the HP circuit presents therefore significant benefits at rated speeds. In terms of maximum BMEP, variations from 1 bar to 3 bar give additional advantages to the HP systems. Hypotheses of identical turbocharger efficiencies between both EGR systems are obviously unexpected in practice because the different gas mass flows move the running operations to different places in the compressor and turbine maps. At low speeds, turbocharger efficiencies are greater with LP EGR because the higher gas mass flows center the operating conditions in the characteristics maps, while at high speeds this effect is produced with HP EGR. These efficiency variations which strongly depend on the turbocharger maps can therefore positively or negatively influence the results previously found. Nevertheless, these variations are relatively small and generally go in the same directions as the trends observed. Their impacts have thus limited consequences on the obtained conclusions.

4.7 Synthesis of Thermal Constraints with EGR

To synthesize how the EGR rates and thermal constraints limit the engine performance, the maximum reachable BMEP obtained with the different EGR configurations have been plotted in figure 13. With EGR requirements at full load, it can be seen that the maximum allowable compressor outlet temperature is now more restrictive than the maximum allowable exhaust temperature. In fact, engine performances are limited by compressor outlet temperatures before exhaust constraints exceed 800°C. As an intermediate intercooler presents limited potential to reduce the compressor thermal constraints, advanced materials for both compressor wheel and intake piping are thus necessary for the further development of highly-rated downsized-downspeeded engines running at full load with EGR. Titanium compressor impellers able to withstand higher temperatures and higher cyclical loads are already present in the market for special applications [103], but their costs are still challenging to see their rapid spread in low to medium class vehicles.

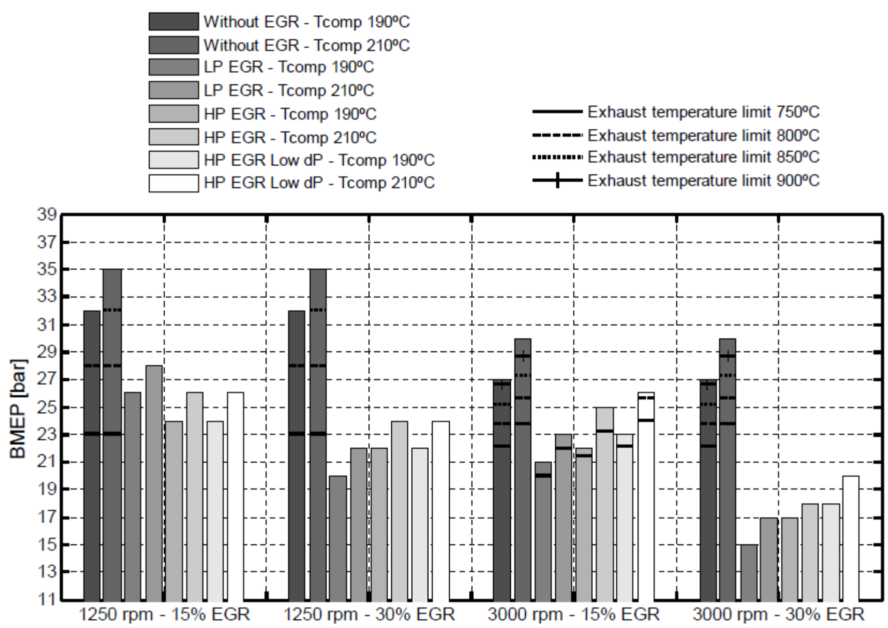

Figure 13: Synthesis of maximum reachable BMEP under LP and HP EGR rates. temperatures

5 Two Stage Operation

In this subsection, the energetic approach has been extended to the two stage operations. Simulations have been performed at full load at 1000 rpm and 1250 rpm which represent the most critical two-stage running conditions for the considered boosting architectures. As already mentioned, the ability of the main turbocharger to produce boost at these speeds is generally very limited and mainly depends on the turbocharger matching. That is why the results have been divided in two representations. On the one hand, the desired boost is entirely provided by the second charger and the engine performances are analyzed as a function of brake power levels. On the other hand, as calculations are not limited by turbocharger operating ranges, the required boost is provided by a combination of both chargers and the engine performances are analyzed as a function of compression ratio distribution for a given brake power level. 0% compression ratio distribution corresponds to a boost demand entirely produced by the main turbocharger while 100% represents one completely supplied by the second charger.

Comparing the boosting architectures, a representation is obtained with the different second charger technologies, because the supercharger uses net mechanical power from the crankshaft, the turbocharger recovers waste energy from the exhaust gases and the eBooster consumes electricity supplied by an external source. For the eBooster, the electric consumption is not taken into account in the calculations (free driving energy). It is assumed recovery systems such as regenerative brakes (*4*) (*18*) can produce enough electricity to respond to the eBooster demands through energy storages (i.e. supercapacitors). Therefore three electric power levels have been considerd for simulations which are 2 kW, 4 kW and 8 kW. To analyze the engine and boosting architecture performance under two stage operations, a first sensitivity study has been performed on the charger efficiencies with the values presented in Figure 3. The calculations have been carried out without EGR, without intermediate intercooler and using the hypotheses of maximum in-cylinder pressures corresponding to future engine developments (see figure 6). These hypotheses have been selected to reduce the influences of in-cylinder pressures limitations and to increase the maximum brake power level range for systems comparisons. Since pressure losses characteristics have limited impacts at these speeds, the reference engine com-

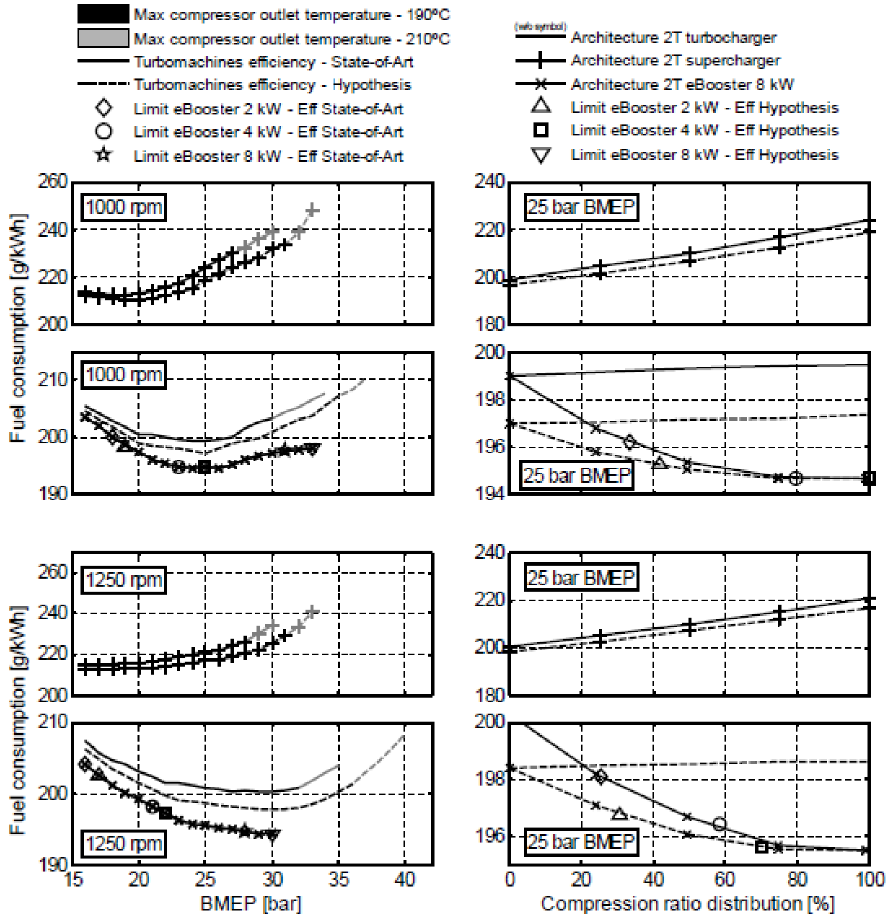

Figure 14: Engine and boosting architecture performance under two-stage operations.

ponents described in figure 4 have been retained. The results of this sensitivity study are plotted in figure 14. As expected, the supercharger presents the highest fuel consumptions. When compared to the turbocharger, the supercharger fuel penalties reach 15 g/kWh at 20 bar BMEP and more than 35 g/kWh at 35 bar BMEP. Be- tween the turbocharger and eBooster, the differences are relatively small with values around 5 g/kWh. As the eBooster driving energy has no impact on fuel consumption, these small differences show the efficiency of the turbocharger to fulfill the desired boost demands through waste energy recovery from the exhaust gas. Regarding the efficiency variations, the same conclusions as those obtained in the previous subsection can be noticed for the turbocharger (fuel savings of around 2-3 g/kWh and maximum BMEP increase of around 3-4 bar). For the supercharger, an efficiency variation of 10% does not reduce in a significant way the required mechanical power. In fact, BSFC are only decreased from 2 g/kWh to 5 g/kWh according to the brake power level. That means efforts in supercharger design optimization do not show important potential to diminish fuel penalties generated by mechanical chargers. The efficiency variation also increases the maximum BMEP by 2-3 bar but, as part of the brake power is employed to drive the supercharger, the maximum BMEP stays around 4-5 bar lower than those reached with the turbocharger. For the eBooster, the maximum reachable BMEP strongly depends on the electric power limitations. For example at 1250 rpm with conservative efficiencies, maximum powers of 2 kW, 4 kW and 8 kW restrain the engine performance to 16 bar, 21 bar and 28 bar BMEP respectively. Without these limits, the engine performance could be increased until reaching the maximum allowable compressor outlet temperatures and the corresponding maximum BMEP would be slightly greater than the turbocharger ones. Increasing the eBooster efficiency by 10%, It will reduce the electric power needs allowing for a given electric power level to increase the maximum BMEP by 1-2 bar. The electric power results are shown here for the 2.3l engine. Although BSFC results can be generalized to similar downsized-downspeeded engines, the electric power results rely on gas mass flows and are specific to a given swept volume. They cannot therefore be assumed for other engine displacements. For that reason, the specific power limitations obtained on the 1.2l and 1.6l engine will be presented at the end of this subsection with the synthesis of the maximum performance results. Thanks to the energetic approach, the impact of the compression ratio distribution between both stages can be analyzed without turbocharger op- erating range limitations. Considering a representative brake power level (25 bar BMEP), it can be seen how the fuel consumption is progressively reduced in the 2T supercharger and 2T turbocharger configurations as the proportion of boost provided by the main turbocharger increases. In the 2T eBooster configuration, this trend is reversed as the electric power is supplied by an external source. At this brake power level, modifying the compression ratio distribution from 100% to 0% brings for the 2T supercharger configuration fuel benefits of up to 20 g/kWh. This is mainly explained by the reduction of brake power needs. For the 2T turbocharger configuration, these fuel benefits are much smaller reaching only 2 g/kWh due to the limited efficiencies differences between both turbochargers. These small fuel savings give thus certain flexibility to the boosting architecture to optimize other objectives such as en- gine control, mode transition, EGR abilities at part loads [345], etc. . . without significantly deteriorating the fuel consumption. For the 2T eBooster configuration, using the main turbocharger can increase the BSFC up to 3-4 g/kWh. However in this architecture the selection of the optimum compression ratio distribution depends not only on the main turbocharger boost abilities but also on the electric power limitations which can make unachievable a 100% distribution. For example here with conservative efficiencies, the 2 kW and 4 kW maximum electric powers limit the compression ratio distribution at 25% and 58% respectively.

5.1 Effect of Interstage Cooling

With the same approach, the fuel benefits obtained using an intermediate intercooler have also been analyzed for two brake power levels (20 bar and 30 bar BMEP). With this cooler, the maximum reachable BMEP have not been considered due to the extremely high values that could theoretically be achieved. After a first compression in the LP stage, an intermediate intercooler allows to reduce the HP compression work increasing the gas density at the HP charger inlet. Nonetheless, adding an intermediate intercooler increases the pressure losses in the intake line. Fuel savings are thus a balance between both effects. The intermediate intercooler operates only at low speeds during two-stage operations. Its design is generally smaller than that of the after- cooler. However here to analyze an optimistic situation, the same pressure losses characteristics have been retain in both coolers.

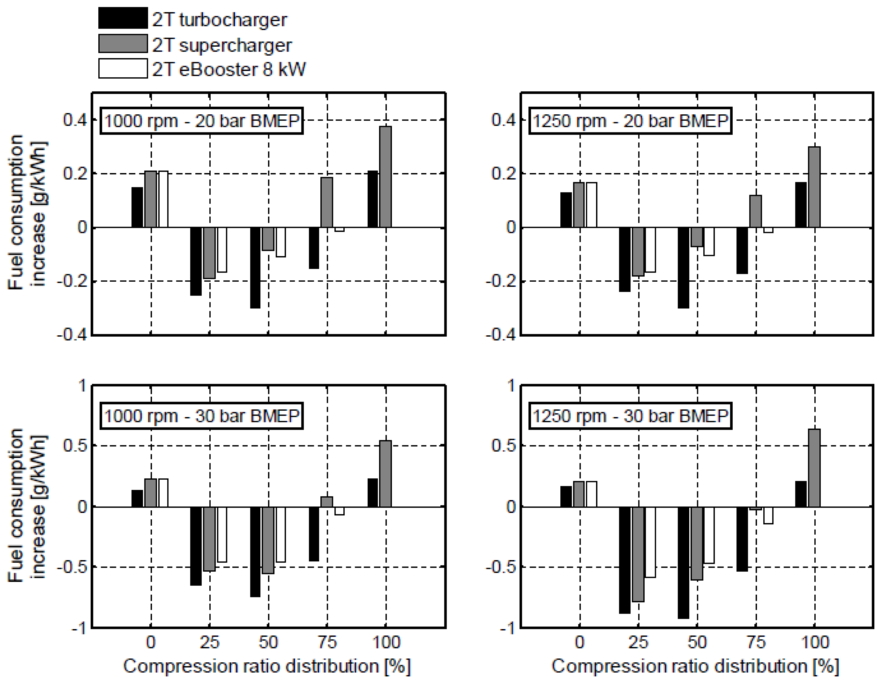

Figure 15: Impact of inter-stage cooling on engine fuel consumption as a function of compression ratio distribution.

The results of this study are shown in figure 15 as a function of compression ratio distribution. At 0% and 100% compression ratio distribution, there is obviously no fuel benefit from compression work reductions and the results reflect fuel penalties generated by higher pressure losses. The differences observed at 0% between the different architectures mainly lie in the intercooler relative position. In fact in the 2T turbocharger configuration, the intercooler is fitted downstream the main turbocharger while in the 2T supercharger and 2T eBooster configurations it is placed upstream. At 100%, the differences are higher with the supercharger as the pressure losses must be offset using mechanical power, while they are null with the eBooster as its electric consumption is not considered. For the 2T configuration, at 25% the HP compression work is relatively small. So, a reduction of this work has limited consequences on the fuel consumption. At 75%, the HP charger work is much more important but the temperature rise in the LP charger is relatively small. So, an intermediate cooling process has also little effect and the maximum benefits are obtained around 50%. For the other configurations, the same effects are noticed but the maximum benefits are rather observed around 75% due to the different costs that represent offsetting the pressure losses with the second charger (any impact with the eBooster while important fuel penalties with the supercharger). At the end, the fuel benefits are generally very small with maximum values of around 0.2 g/kWh at 20 bar BMEP and 0.9 g/kWh at 30 bar BMEP. So, even though the main turbocharger has the ability to produce boost at low speeds, these small fuel savings do not justify the cost and packaging constraints that involve the implementation of an intermediate intercooler.

5.2 Effect of EGR Level in 2-Turbo Operation

To complete the results obtained under two-stage operations, a second sensitivity study has been performed on EGR rates provided by the LP EGR circuit (0%, 15% and 30%). Here the HP circuit has not been considered because, on the one hand, the supercharger and eBooster do not have any ability to produce the required engine backpressures, and on the other hand the main conclusions regarding the differences between HP and LP systems working with a turbocharging architecture have already been given in the last subsection.

The calculations have been carried out with conservative efficiencies and without intermediate intercooler. The results are plotted in figure 16 using the representations previously defined.

With the hypotheses assumed on the elements pressure losses characteristics, LP EGR has an impact on the combustion process and chargers work. For the e-Booster, the compressor work is produced with electricity coming from an external source. The fuel benefits of around 5 g/kWh that can be observed between the different EGR rates correspond therefore to the combustion efficiency improvements generated by the injection timings strategy. For the turbocharger, the higher compression works increase the turbine expansion ratios and the resultant

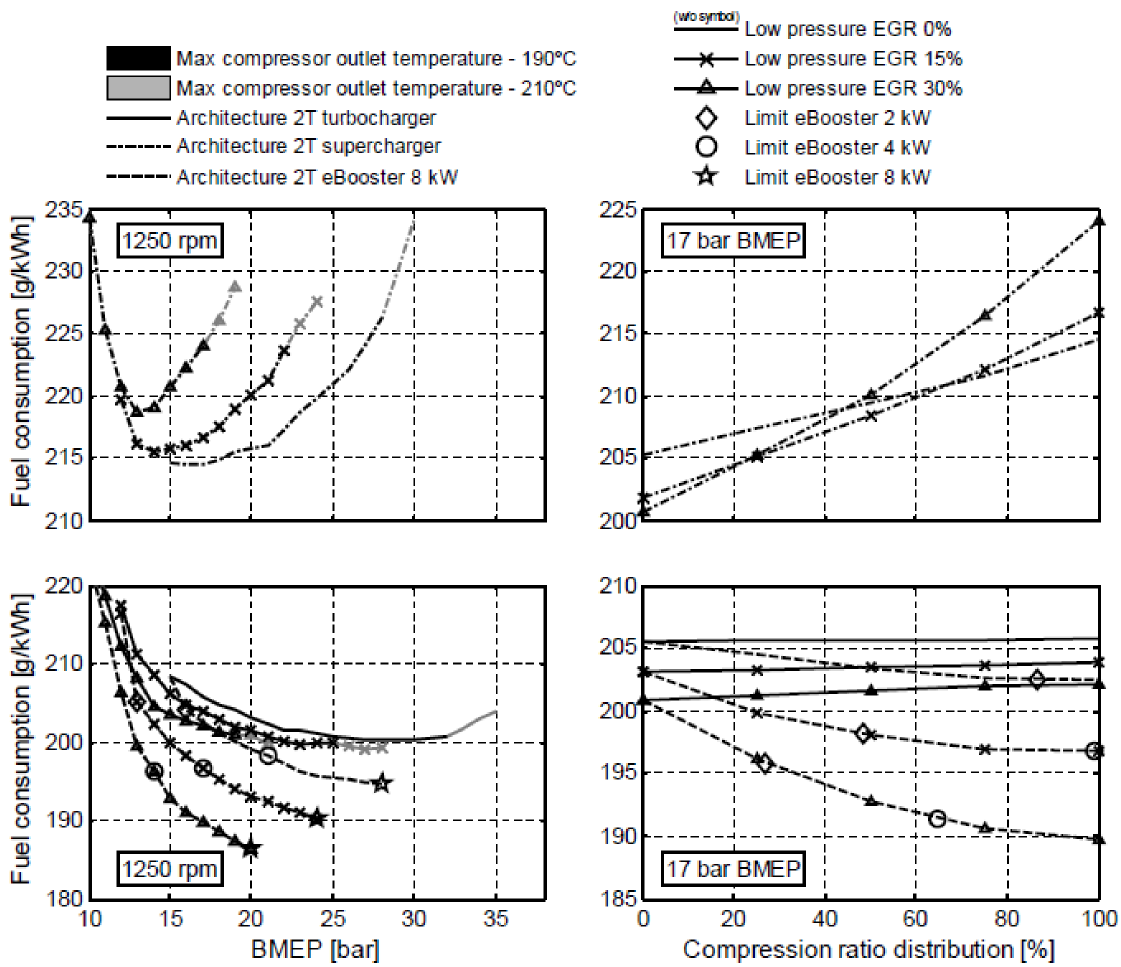

Figure 16: Impact of LP EGR rates, electrical power limitations and maximum compressor outlet temperature on engine and boosting architecture performance under two-stage operations.

engine backpressure losses. Comparing the e-Booster and turbocharger results, the fuel penalties involved by these losses can thus be estimated to around 8 g/kWh and 13 g/kWh at 15% and 30% LP EGR respectively. However here, the combustion improvements offset these loses and BSFCs are maintained almost constant between the different EGR rates. For the supercharger, the fuel penalties involved by higher brake power demands are too important to be offset by the combustion improvements and fuel consumptions are deteriorated under LP EGR. In terms of maximum engine performance, increasing by 15% the LP EGR rate reduces the maximum

BMEP by 5-7 bar in the case of the turbocharger and supercharger due to the maximum compressor outlet temperatures, while this reduction is around 2-4 bar with the e-Booster due to limited electric power levels. Regarding the compression ratio distribution influences, it can be noticed the same trends as those previously described for the two-stage operations running without EGR.

5.3 Synthesis of Thermal Constraints in 2 Turbo Operations

Finally, to synthesize how the thermal constraints, the EGR rates and the electric power levels limit the engine performance, the maximum reach- able BMEP obtained under two-stage operations have been plotted in figure 17. As it can be observed, when the electric power is not restrained, the 2T e-Booster architecture allows to reach 1-2 bar higher maximum BMEP than the 2T turbocharger configuration due to free exhaust gas mass flows. Whereas, the 2T supercharger architecture reaches 2-5 bar lower maximum BMEP due to brake power consumption. For the thermal constraints, if the exhaust temperature limitations are lower than 850°C, the maximum exhaust temperature stays the limiting factor in the 2T turbocharger configuration running without EGR. Otherwise, with higher exhaust temperature limitations or under EGR, the maximum compressor outlet temperature becomes more restrictive. In the 2T e-Booster and 2T supercharger configurations, the exhaust temperature limitations are not so critical because the engine backpressures are significantly lower. In these architectures, the maximum compressor outlet temperature is therefore always the limiting factor. Modifying the thermal resistance of the intake piping system from 190°C to 210°C presents thus important benefits in most cases to improve by 2-3 bar the maximum reachable BMEP.

Regarding the electric constraints, the electric power level requirements are proportional to the gas mass flows which mainly depend on the engine displacement. Here, it can be noticed how the 2 kW, 4 kW and 8 kW electric power limitations restrain the maximum reachable

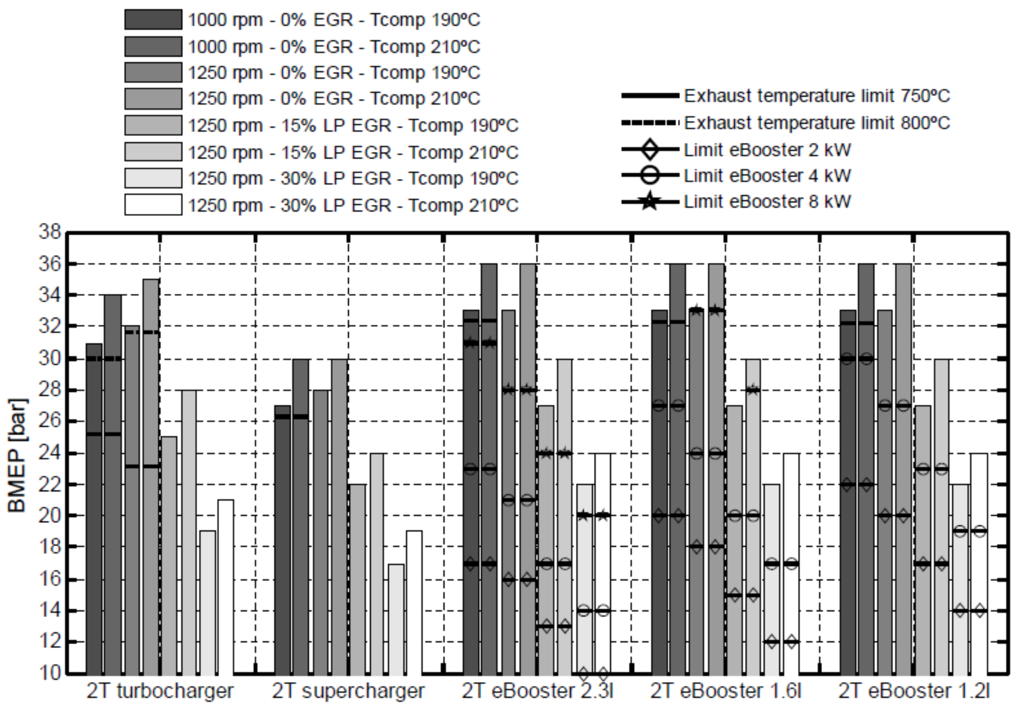

Figure 17: Synthesis of maximum reachable BMEP in two-stage operation.

BMEP for the different engine displacements. To achieve the maximum compressor outlet temperatures, the electric power levels must approximately exceed 10 kW, 8 kW and 6 kW for the 2.3l, 1.6l and 1.2l engines respectively. The maximum electric power level defined by the e-Booster motor or by the electric vehicle network is therefore in most cases the limiting factor to reach high low-end torques with the 2 Turbo e-Booster configuration.

6 Conclusion

According to the parametric study to characterize the limits and performance of the most promising boosting architecture on the base engines. Simulations have been performed with the 0D engine model and a specific methodology has been defined to obtain general conclusions valid for most downsized-downspeeded engines. This methodology is based on the similarity theory to reproduce analogous behavior between the different downsized engines and, for the steady-state calculations, it is also based on an energetic approach to avoid influences of specific components designs (hypotheses on intake/exhaust line element characteristics, turbocharger maps, etc. . .). Several sensibilities studies have been conducted to determine the main factors that govern the architecture performance and to quantify their impacts on fuel consumption and maximum rated power. These factors regroup the parameters such as turbocharger efficiencies, engine elements pressure losses characteristics, thermomechanical limitations (maximum in-cylinder pressure, exhaust manifold temperature, compressor outlet temperature, etc. . .), EGR rates and EGR system technology (HP and LP circuits). In two-stage operations, additional analyses have also been performed to compare the performance of the considered architectures characterizing the different systems interactions and evaluating possible interstage cooling benefits. From these results, the required charger operating ranges have been confronted to conservative characteristics maps. Through a representative data base that allows the actual technological limits to be judged, new requirements have been defined for future turbocharger developments and new characteristic maps have been extrapolated to perform transient calculations.

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