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Two-Stage Turbocharging for the Downsizing of SI V-Engines

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Abstract

One of the most critical challenges for the specific power increase of turbocharged SI engines is the low end torque, limited by two aspects. First, the big size of the compressor necessary to deliver the maximum airflow does not allow high boost pressures at low speed, due to the surge line proximity. Second, the flame front velocity may become slower than the end gas auto-ignition rate, thus increasing the risk of knocking.

This study is based on a current SI GDI V8 turbocharged engine, modeled by means of CFD tools, both 1d and 3d. The goal of the activity is to lower by 20% the displacement, without reducing brake torque, all over the engine speed range.

It was decided to adopt a smaller bore, keeping stroke constant. Obviously, the combustion chamber, the valves and the intake-exhaust ports have been re-designed, as well as the whole intake and exhaust system. Instead of the two turbochargers, one for each bank of cylinders, a triple-turbocharger layout has been considered.

The development of the engine has been carried out by means of 1D engine cycle simulations, using predictive knock models, calibrated with the support of both experiments and CFD-3d simulations. A few operating conditions for the final configuration have been also analyzed by means of a 3-d CFD tool.

The paper presents the results of this activity, and describes in details the guidelines followed for the development of the engine.

1. Introduction

Turbocharging, variable valve timing and direct injection are the most widespread technologies for enabling engine downsizing in current sporting cars. Some production engines already exhibit BMEPs higher than 23 bar, and this threshold is going to be pushed further. The most important challenges to this tendency are the increased risk of knock and the limited air supply at low engine speed. A possible way to address these issues is to reduce the cylinder displacement by adopting smaller bores and increase boost pressure through an unconventional supercharging system. Smaller bores require a new design of the combustion chamber, including valve ports, injector and spark plug(s).

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As bore decreases, many design constraints become tighter, being generally more difficult to scale head and piston dimensions [1], and becoming more difficult to cool the engine [12]. However, a smaller bore yields the following advantages: a) reduced flame path, about same flame speed (the mean piston speed does not change), thus less risk of knock at the same boost; b) smaller heat transfer area, thus higher thermal efficiencies; c) smaller and lighter valves, then possibility to adopt more "aggressive" cam profiles; d) smaller and lighter pistons, then less vibrations; e) smaller dimensions and weight of the powertrain.

The combustion system design of the downsized engine is a wide and complex subject, and it is fully reviewed in a parallel paper [2]. Therefore, the current report is focused on the problem of air supply at low engine speed, and no detailed combustion analysis will be presented here. The hypothesis adopted in this study, and fully confirmed by the parallel combustion analyses, is that the burned fuel fraction profiles tend to shrink in the optimized downsized engine [3,4]. The parallel paper also shows that, adopting a proper calibration of the ignition and injection parameters, knock does not occur.

Concerning air supply at low speed, conventional turbocharging systems (typically, one turbocharger per cylinders bank) present stalling issues. In fact, the maximum compressor flow rate must be about the same of the full bore engine, in order to deliver the same airflow at maximum power. However, at low speed, boost pressure should be higher, for compensating the displacement reduction, and the operating points on the compressor maps would fall beyond the surge line.

The triple turbocharger system, see figure 1, may be a solution to the above mentioned problem. The triple turbocharger is conceptually similar to a 2-stage system [5,6], the only difference being that the high pressure stage is made up of two parallel machines, instead of one. The low pressure stage consists of a quite big turbocharger, delivering a flow rate about two times higher than that of the single bank turbocharger, in a conventional system. Conversely, the high pressure turbochargers are much smaller, since they are completely by-passed at an engine speed higher than 3500 rpm. Furthermore, below 3500 rpm, the low pressure compressor delivers a charge already compressed to the high pressure stage, whose maximum volumetric flow rate is therefore reduced, in comparison to a compressor with ambient induction.

In the triple layout, with a proper choice of each machine, the turbochargers of both stages may operate at high efficiency conditions, all over the engine speed range. The surge line is no longer a problem, as the two small turbochargers in the high pressure stage have almost no boost limit, even at very low engine speeds (<1500 rpm). Another fundamental advantage is the reduced back-pressure imposed by the big turbocharger at the low pressure stage, yielding a strong decrease of pumping losses at medium-high engine speeds.

The triple turbocharger is obviously more complex than a twin turbocharger, from two points of view: electronic control and packaging. Instead of a pair of waste-gates, the triple layout needs an additional waste-gate on the low pressure turbine, a further intercooler between the two stages, a pair of by-pass valves for the high pressure compressors, and another pair of by-passes for the pre-catalysts. The last ones are necessary in order to prevent the catalyst overheating when the high pressure stage is completely by-passed, and they yield some benefit on the pumping losses.

The necessity of the additional intercooler brings both pros and cons. On the one hand, a further component yields packaging issues and additional weight, on the other hand the charge temperature at high engine speed may be lowered, since the two intercoolers are serially connected without the interposition of a compressor (by-passed). A colder charge entering the cylinders is obviously a big advantage for avoiding knock, especially at high boost pressures.

The adoption of very small turbochargers turns into an advantage, in terms of response to the accelerator tip-in at low engine speeds, despite the larger volumes of both intake and exhaust systems.

Finally, it is observed that the two banks of cylinders may be totally independent up to the low pressure turbine, as in the twin turbocharger system. This feature makes possible to adopt a cylinder de-activation strategy at partial load, as described in[7].
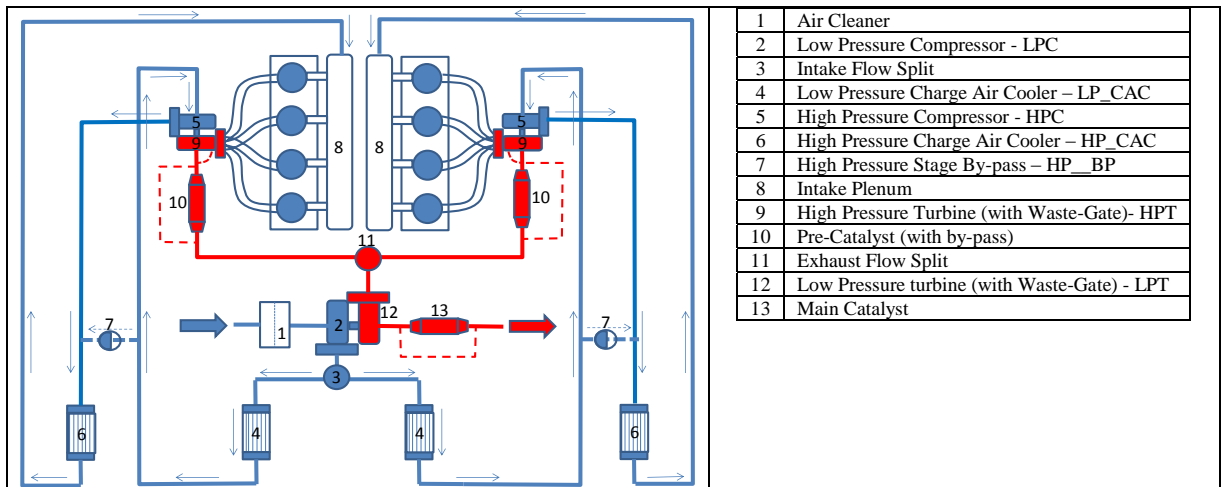
1	Air Cleaner
2	Low Pressure Compressor - LPC
3	Intake Flow Split
4	Low Pressure Charge Air Cooler – LP_CAC
5	High Pressure Compressor - HPC
6	High Pressure Charge Air Cooler – HP_CAC
7	High Pressure Stage By-pass – HP__BP
8	Intake Plenum
9	High Pressure Turbine (with Waste-Gate)- HPT
10	Pre-Catalyst (with by-pass)
11	Exhaust Flow Split
12	Low Pressure turbine (with Waste-Gate) - LPT
13	Main Catalyst

Figure 1: Lay-out of the triple turbocharger system

2. Engine modeling

Simulations are performed by using GT-Power, the reference model being that of a current twin turbocharger engine, V8, 3.8 L, with intake and exhaust VVT. The model, previously calibrated against experiments, has been modified reducing by 10% the cylinder bore (-20% of total displacement), and replacing the original gas-dynamic system with the triple turbocharger.

The following hypotheses are made:

- the full bore and the downsized engine have equivalent flow losses from ambient to the compressor inlet, and from the last turbine outlet to ambient;
- the intake plenums (one for each bank) and the throttle bodies (one for each bank) are identical
- no change in the modeling of flow losses and heat transfer;
- the correlation between fmep and mean piston speed and in-cylinder peak pressure is the same between the full bore and the downsized engine;
- the geometry of intake and exhaust ports, as well as of the combustion chamber, is the same between the full bore and the downsized engine (whose main dimensions are scaled of 10%);
- because of the above mentioned similarity, the valve discharge coefficients as a function of non-dimensional lift are the same;
- the engine speed range is the same (1000-7000 rpm);
- the same constraints are applied to the valve profiles design;
- the 0-90% angle in the burn rate profiles of the downsized engine is scaled of 10%;
- the intercoolers efficiency is 80%;
- combustion efficiency is 1.

The downsized engine has been optimized through GT-Power simulations, considering the following parameters:

- intake and exhaust valve profile and timing;
- intake runners main cross section and length;
- exhaust manifolds geometry (lay-out, ducts cross section area and length);
- size (defined as the maximum reduced mass flow rate) of HPC, HPT, LPC, LPT;
- MAP;
- Lambda;
- spark timing

The goals of the numerical optimization are:

- maximum brake torque of 650 Nm from 2000 to 4500 rpm (bmep=27.2 bar)
- maximum power higher than 400 kW at a speed lower or equal to 7000 rpm

The constraints are:

- same or lower level of knock, in comparison to the full bore engine;
- same or lower max. overlapping angle, in comparison to the full bore engine;
- max. in-cylinder pressure: 100 bar;
- max. gas average temperature at the turbine inlet: 1000 °C;
- max. gas average temperature at the catalyst inlet: 900 °C;
- no variable turbine geometry

In order to assess the knock tendency of the engine, the amount of unburned charge at the knock onset has been considered, as done in [8, 11]. This parameter is calculated by GT-Power on the basis of the Douaud and Eyzat formula for induction time [9]. It should be noted that the knock model does not affect burn rate, which is entered as an input.

Despite the simplified modeling, this approach is able to generate realistic combustion results. Further CFD-3D simulations have been carried out at the most critical operating conditions (2000, 4500 and 7000 rpm, wide open throttle), using the boundary and initial conditions calculated by the 1D software. These results are reported in a parallel paper [2], and they definitely confirm the feasibility of the proposed downsizing.

3. Full load performance (steady operations)

The main features of the optimized downsized engine are reviewed in table 1.

It is observed that exhaust valves stay open for a longer time than intake valves (275 vs 230 °CA), and the relative exhaust maximum lift is higher (0.356 vs. 0.315). These features are unusual but not completely surprising in a turbocharged engine, since exhaust permeability is generally more important than that at the intake side, where the airflow rate can be simply controlled by adjusting boost pressure. Conversely, a reduced mean effective area of the exhaust valves yields higher pumping losses, especially when the valves are small, as in a downsized engine [10].

Another interesting aspect is the total lack of tuning for the intake system (very short runners). The reason for the dynamic effects elimination is that strong pressure waves tend to increase the air temperature, raising the risk of knock.

Concerning the choice of the turbochargers, the following considerations are made:

- the optimum size of the high pressure (HP) compressor makes the operating points fall in the map region of maximum efficiency;
- the size of each turbine should closely match the coupled compressor;
- the ideal low pressure (LP) compressor is able to provide alone the engine boost target over 3500-4000 rpm, at full load; a bigger compressor shifts the switch from HP to LP turbochargers at higher speed values, the contrary occurs with a smaller machine; a bigger LP turbocharger needs a pair of bigger HP turbochargers, which, in turn, yield less engine performance at low speed; conversely, a smaller LP turbocharger is going to waste more energy through the waste-gate at high engine speed, raising pumping losses.
- Figures 2-3 review the triple turbocharger operations at full load and the subsequent engine performance. The comparison with the full bore engine are mostly made in qualitative terms, since data are confidential.
- COMPRESSORS MAPS The position of the operating points on the maps is a clear evidence of the good matching between the engine and the turbochargers: all the points fall in high efficiency regions, far from the surge and choking lines.
- COMPRESSOR EFFICIENCY   The efficiency of LPC is very high, 69-75% except at 1500 rpm (60%). Very good also the efficiency of HPC (about 70%), except at 3500 rpm, where it drops at 55%. At higher speed HPC is bypassed.

Number of cylinders	-	8
Total displacement	cc	3010
Bore to stroke ratio	-	< 1
Compression Ratio	-	9:1
Fuel Injection	-	Direct
# valves per cylinder	-	4
Intake valves opening duration	°CA	230
Intake valves maximum lift / seat diameter	-	0.315
Exhaust valves opening duration	°CA	275
Exhaust valves maximum lift / seat diameter	-	0.356
Max. valve lift at TDC	mm	3.8
Intake Runner diameter / bore	-	0.47
Intake Runner length (plenum-head)	mm	50
Exhaust pipe diameter / bore	-	0.47
Exhaust pipe length (head-turbine)	mm	160
Exhaust pipes lay-out	-	2 in 1
Intake Plenum Volume (1 bank)	cc	1740
LPC: Max. reduced mass flow rate	Kg/s.K$^{0.5}$/bar	9.0
LPT: Max. reduced mass flow rate	Kg/s.K$^{0.5}$/bar	5.0
HPC: Max. reduced mass flow rate	Kg/s.K$^{0.5}$/bar	2.2
HPT: Max. reduced mass flow rate	Kg/s.K$^{0.5}$/bar	1.0

Table 1: Main features of the optimized downsized engine

TURBINE EFFICIENCY   As for the compressor, also the LPT efficiency is good (60-65%). The HPT efficiency is a little worse (44-54%).

COMPRESSORS PRESSURES   The control strategy is clearly visible in this graph: starting from 1500 rpm, as engine speed increases, the contribution of LPC increases, while HPC shows the opposite tendency. Over 4000 rpm, HPC is by-passed, so that its inlet and outlet pressures are coincident. The little pressure drop between LPC and HPC over 4000 rpm is due to the flow losses along the piping and the intercoolers.

TURBINES PRESSURES   The pressure ratio across LPT increases steadily along with engine speed. Over 4000 rpm, a small pressure ratio across the HP turbine is necessary to keep the turbocharger running with the compressor by-pass open.

COMPRESSORS AND ENGINE TEMPERATURES   After HPC, and before entering the cylinders, the charge if further cooled in the second heat exchanger. Thanks also to the high efficiency of the intercoolers (80%), the maximum temperature increment from ambient is just 15 °C.

TURBINES TEMPERATURES   HPT temperatures are critical, being very close to the limit of 1000 °C. If this threshold is to be lowered, the exhaust manifolds cooling becomes mandatory.

ENGINE PRESSURES   Engine inlet pressure is comparable to the one adopted on the full bore engine, while the exhaust back-pressure is lower. The last result can be easily explained considering that, at medium-high speed, the small HP turbocharger is by-passed, and the remaining LP turbine is much bigger and more efficient than the couple of turbines in the full bore conventional engine. Concerning the same level of boosting, the downsized engine succeeds in compensating the smaller displacement thanks to the following issues: a) higher volumetric efficiency at any given boost, due to more "aggressive" lift profiles (since the valves are smaller and lighter) and to a lower back-pressure; b) lower heat losses, due to the smaller heat transfer areas; c) higher mechanical efficiency, due to the lower ratio of friction to indicated MEP; d) lower pumping losses (due to a lower engine back-pressure)

WASTE-GATES DIAMETER   The flow by-passing HPT is not discharged in the ambient, but it feeds LPT: as

a result, the boost control in the triple turbocharger system is less dissipative than in a twin system.

VOLUMETRIC, CHARGING AND TRAPPING EFFICIENCY   As engine speed decreases, for optimizing engine performance it is convenient to reduce the intake valve closure retard (less back-flow through the valve during the compression stroke), as well as the exhaust valve opening advance (more expansion work). As a result, the valve overlapping period increases, and trapping efficiency drops. Since fuel is injected after EVC, fuel consumption is not affected. The higher airflow rate at a given pressure ratio produces also some benefit in terms of compressor operations: the operating points on the map are shifted rightward, far from the surge line.

PMEP, IN-CYLINDER PRESSURE, BMEP   For the given torque target, in-cylinder peak pressure increases (in the reference engine the last parameter does not exceed 80 bar). The triple turbocharger, however, enables a strong reduction of pumping losses, so that at 7000 rpm there is a benefit of more than 2 bar in terms of PMEP.

TRAPPED AIR-FUEL RATIO AND KNOCKING TENDENCY   Air-fuel ratio and start of combustion have been set in the downsized engine in order to achieve the same (or lower) knock tendency of the full bore engine. The most critical conditions for the downsized engine occur at low engine speed, where the air-fuel mixture (supposed to be homogeneous) has to be set quite rich in order to lower the end-gas temperature. A stratification of the charge would be definitively more efficient for controlling knock, but this strategy cannot be analysed by means of a simplified 1-d approach.

BRAKE PERFORMANCE   Brake torque and power curves meet the performance targets, all over the speed range. Over 4000 rpm, brake specific fuel consumption is very good, in comparison to the full bore engine, showing values below 300 g/kWh. The merit of this fuel efficiency goes to the low pumping losses, and to the low knocking tendency. At low-medium speed, fuel efficiency is spoiled by the need of rich mixtures.
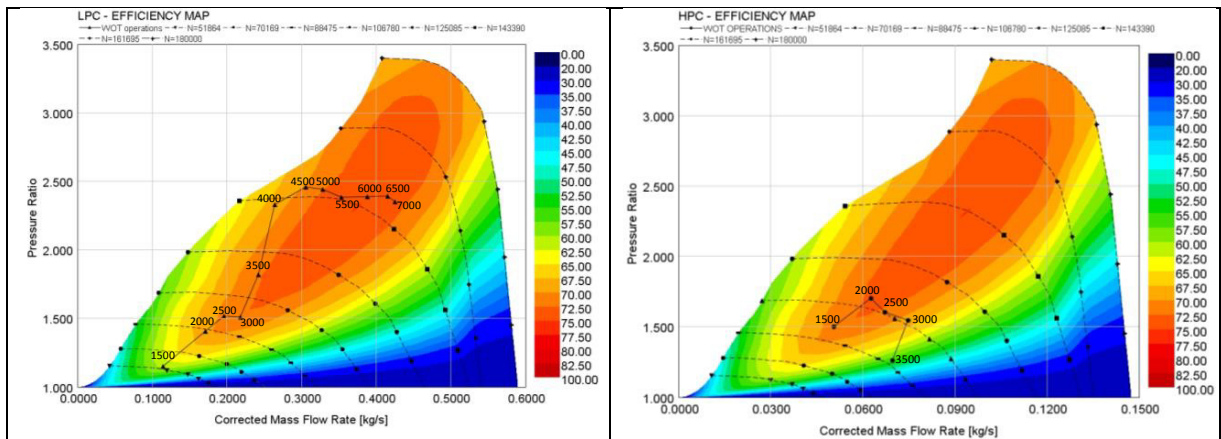

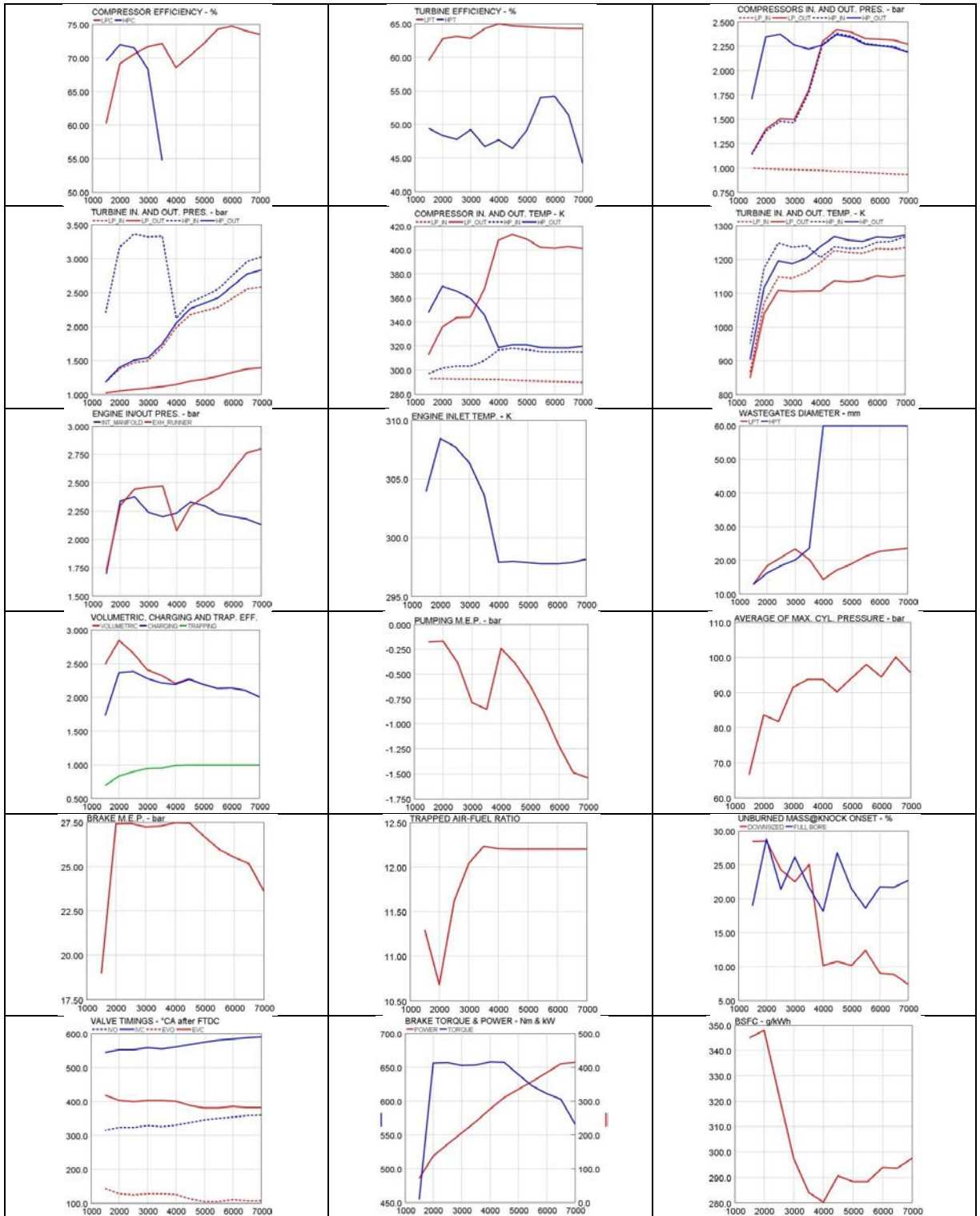Figure 2: LP and HP compressor map, with operating points

Figure 3: Engine performance at full load as a function of engine speed

4. Conclusion

The paper reviews the 20% downsizing of a current SI GDI V8 turbocharged engine, obtained by reducing bore and adopting a triple turbocharger lay-out. The study is performed with GT-Power, while the CFD-3D analyses are presented in a parallel paper [2].

The main advantages found for the triple turbocharger, in comparison to the standard twin configuration, are: higher boost pressure at low engine speed; lower pumping losses at high speed, full load, better transient response.

The main drawbacks are: higher complexity of the control system; larger overall dimensions; cost.

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