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The Modular Engine Concept: a Cost Effective Way to Reduce Pollutant Emissions and Fuel Consuption

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ABSTRACT

A promising technique to enhance fuel efficiency of large capacity S.I. engines is the de-activation of some cylinders at partial load, through the cut-out of fuel metering and a specific control of the airflow. Thanks to the ensuing reduction of throttling losses (the active cylinders operate at a much higher load), fuel consumption can be reduced, without any negative perception from the driver. Such a technique has been already applied successfully on some production engines, at the cost of some additional complication on the valve-train system.

The application analyzed in this study is a little bit different, being aimed to reduce both fuel consumption and emissions, with a minimum impact on engine design. Larger fuel savings may be obtained by coupling the cylinder de-activation with VVT.

However, the most important advantage of the modular engine concept proposed in this paper is in terms of emissions: this study demonstrates that the light-off time of the catalysts may be strongly reduced, and a further improvement is obtained by doubling the effective surface of the catalytic bed.

The study has been carried out on a conventional SI 4.2L V8 engine. The first step of the analysis has been the experimental validation of a 1D-CFD model of the engine, achieved with a very good accuracy at both full and partial load. Then, the engine has been simulated on a grid of 15 operating points, representing the usage in the New European Driving cycle. The following configurations have been analyzed and compared to the base engine: 4 active cylinders, 3 active cylinders; 4 active cylinders and optimization of valve timings; 3 active cylinders and optimization of valve timings.

INTRODUCTION

It is well known that a conventional SI engine, operating at low load and speed, is characterized by a poor fuel conversion efficiency, due to the relevant weight of friction and pumping losses on the indicated work. In a driving cycle, the larger is the engine capacity, the lower is the average Brake Mean Effective Pressure, so that big capacity engines are generally characterized by high fuel consumption.

Some emerging technologies, such as variable valve actuation and gasoline direct injection and down-sizing through supercharging, are very effective, but they generally have a strong impact on engine design and development costs. A more cost-effective technique is de-activation of some cylinders, through the cut-out of fuel metering and a specific control of the airflow. Thanks to the ensuing reduction of throttling losses (the active cylinders operate at a much higher load) and the better combustion patterns, fuel consumption can be reduced, without any negative perception from the driver.

The idea is obviously not new: already in 1981 GM applied the concept to the Cadillac 4-6-8 V8 engine, which was able to run on 4, 6 and 8 cylinders, as loads demanded [1]. The major drawback shown in that early application (engine roughness, particularly when switching the number of operating cylinders) could be cancelled today by the electronic management of the power train.

In 1993, Mitsubishi proposed cylinder deactivation on a 4 cylinder in-line 1.6L engine by means of the so called MIVEC system (Mitsubishi Innovative Valve timing and lift Electronic Control) [2]. Depending on driving conditions, the MIVEC system switches among low-speed, high-speed and MD (Modulated Displacement) modes. In the first two modes, MIVEC works as a traditional VVA system, selecting the most suitable valve profiles. In the MD mode, MIVEC switches off two cylinders, by keeping close the correspondent valves.

Since 1998, Daimler Chrysler offers the V5.0L-V8 and the 6.0L-V12 engines with a valve de-activation system (called ZAS), which is able to switch off one half of cylinders within one engine cycle. On the V8 [3], at low speed and part-load operations, two cylinders in every bank are shut off (cylinder 2 and 3 on the right cylinder bank, 5 and 8 on the left). To stop actuation of the valves (both intake and exhaust ones), the link between the valve and the camshaft is interrupted by means of an electro-hydraulic device. To prevent the deactivated cylinders from cooling down during the cut-out phase, the exhaust valves are always closed immediately after a power stroke. The hot gas therefore remains in the cylinder after combustion, keeping the cylinder walls warm. The exhaust system has been modified to ensure smooth transitional operation of the power units. An exhaust valve in the terminal pipe, downstream of the catalytic converters, closes immediately after the cut out of the four cylinders. This lessens the high-pressure waves occurring in the exhaust-gas system in four-cylinder mode and the resulting tendency of the engine and transmission assembly to oscillate. On the V8, Mercedes claims a 7% reduction of fuel consumption in the New European Driving Cycle (NEDC) and even greater savings in other driving conditions, such as at 90 km/h (15%) and 120 km/h (13%).

General Motors too is moving along this path with the “Displacement On Demand” concept which appears to be very close to the one applied by Mercedes Benz [1, 4]. Fuel economy on the next V6 and V8 is expected to be improved up to 25% in some real world driving conditions.

Also BMW Group is assessing the cylinder deactivation strategy, performed by means of the Valvetronic mechanical fully variable valve train [5].

“Meta Motoren und Energie Technik”, has developed a variable valve actuation device, called Cylinder and Valve Deactivation system (CVD), suitable to activate and deactivate the intake and exhaust valves of reciprocating engines [6]. Meta claims that the system can be applied to any rocker-arms valve train, allowing a reliable cylinder cut-out within one engine cycle. The firing order can be modified, in order to optimise driveability, comfort and acoustic emissions. Experiments carried out on a 4 cylinder in-line engine, operating with 2 cylinders completely deactivated, demonstrated that fuel consumption can be improved up to 20% at very low load, while HC emissions under steady operations can be reduced up to 40%, thanks to the improvement of combustion process in the firing cylinders (higher effective compression ratio, lower residual gas fraction). Conversely, NOx emissions are slightly higher. Further investigations have been carried out on a V8 engine, installed on a luxury vehicle. The majority of the New European Driving Cycle operations can be run in a four cylinder mode, with a subsequent improvement of fuel economy amounting to 10.5%. At constant engine speed fuel efficiency depends on load: savings are 18% at 60 km/h, 15% at 90 km/h and 12.5% at 120 km/h.

The Delphi cylinder deactivation system is available for V8 gasoline, port fuel-injected engines, having both pushrod and overhead cam (OHC) valvetrains [7]. The system automatically switches between eight and four cylinder operation modes as a function of speed and load by keeping the intake and the exhaust valves closed within the deactivated cylinders. The actuator used to disable the valves can be either hydraulic or electric.

Ford Motor Co. has investigated cylinder deactivation sensitivity on vehicle driveability [8]. Tests have been carried out on a V10 6.8L displacement truck engine, showing a fuel consumption improvement between 6 and 14% on the EPA cycle. Nevertheless, during low load and speed conditions, vibrations due to torque fluctuations add constraints to the deactivation strategy, reducing its effectiveness.

Recently, Honda patented a device called Variable Cylinder Management (VCM) system, that achieves cylinders deactivation on V6 engine by keeping the intake and exhaust valves closed in the switched off bank [9]. This deactivation is obtained through a solenoid that unlocks the cam followers from their own rockers.

A few experiments carried out by Ferrari on a V8 3.6L unit showed that the engine, operated at partial load with 4 motored cylinders, runs quite smoothly, without a critical increase of vibrations and noise [10]. The deactivation was obtained by means of a fully variable valve actuation device, allowing the ECU to keep the valves closed in the deactivated cylinders. The smoothness of the engine operations was demonstrated by acceleration measures performed by a tri-axial transducer. Such an experimental evidence supports the theoretical consideration that, in SI engines, the crankshaft balance depends much more on inertia forces (not affected by cut-out) than on in-cylinder pressure. Furthermore, the pressure cycle of a motored cylinder is not so far from the corresponding cycle in a firing cylinder at low load.

AN ALTERNATIVE APPROACH

In this paper, an alternative approach to the modular engine concept is proposed. Reference is made to traditional S.I. engines having a large total capacity and an high number of cylinders (six or more) distributed between two banks. A vehicle powered by this kind of units is able to run smoothly in a driving cycle test, even if one half of cylinders is deactivated, i.e. without fuel injection and spark ignition.

The simplest way to pursue the cut-out is to separate the airflow through each cylinder bank, using one throttle valve per bank. At partial load, a bank of cylinders is deactivated (fuel injection is switched off), while the other one operates at a much higher load. No additional device is required for the valve train. The throttle valve controlling the motored cylinders is kept wide open, while the angle of the other throttle is electronically modulated, in order to meet the load target, without any perception from the driver.

The exhaust system requires some specific arrangements, as visible in figure 1. First, a set of control valves allows the flow from the motored bank to by-pass the catalysts. In this way, the back-pressure at the exhaust valve of the motored bank is lessened, with advantages in terms of pumping work. Furthermore, the catalysts is not cooled by the flow of pure air coming from the cylinders. Second, the exhaust gases out of the active bank are routed to the catalysts of the motored bank, in order to keep them warm and to double the effective catalytic surface for the gas after-treatment.

Third, the exhaust gases from the active bank return to their own silencer, except when there is just one silencer for both banks.

The most important advantage of this control strategy is the dramatic reduction of catalysts light-off time. For a prescribed driving cycle, the heat transfer rate from exhaust gases to the catalysts of the active bank is strongly enhanced by both the increase of mass flow rate and gas temperature, since the load of the active bank is much higher than in the case of normal operations.

The limitation of light-off time will reflect directly on the total exhaust emissions of the driving cycle, and a further reduction of pollutants is expected from the longer path (twice) of the exhaust gas through the catalysts.

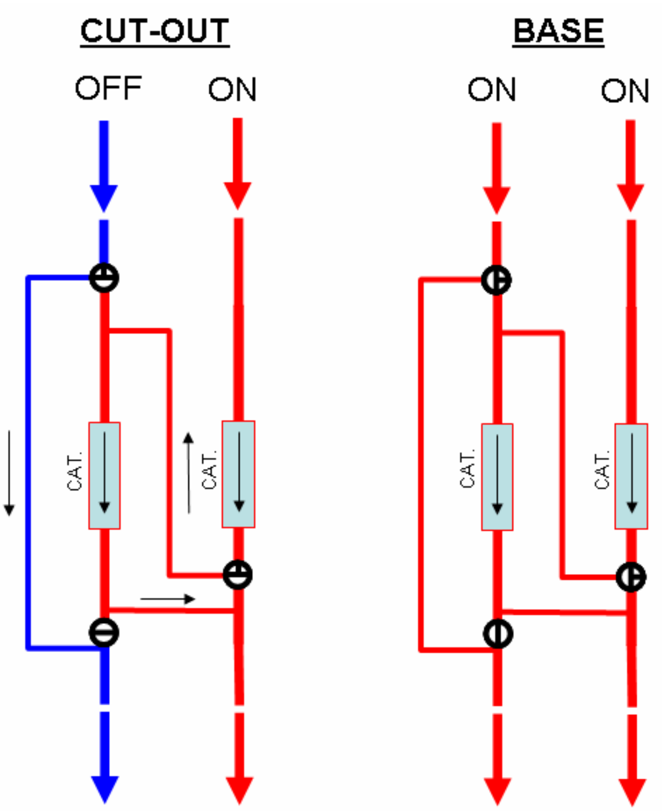

Figure 1: Schematic of the cut-out concept

It should be noticed that large capacity engines, as the one considered in this study, can operate in the cut-out mode all over a standard driving cycle, thus the problem of transition from one mode to the other occurs only on the road, when the driver requires big power. However, the swapping should be quite smooth. In facts, during cut-out operations the motored cylinders are kept warm by the cooling fluid (one circuit must be used for both banks), while the catalysts always operate with hot gases. Furthermore, when the driver push the accelerator pedal down, he should perceive a very quick response from the engine, since there is no fluid-dynamic inertia in the switched-off cylinders, on the contrary, the airflow rate is higher than in the active bank (because of the throttle valve wide open). The response time of the control valves in the exhaust system is not critical in terms of performance, since it affects only exhaust back-pressure.

The longer and more winding path of the exhaust gas, associated with the higher gas flow rate, is expected to generate a higher back-pressure for the engine, in comparison to normal operations. However, the calculations performed in this study demonstrated that this effect is not very relevant on a large capacity engine, being the exhaust system permeability designed to cope with flow rates much higher than those typical of a driving cycle.

The concept described above can be further expanded to include the cut-out of an other cylinder in the active bank, as visible in figure 2. For the sake of brevity this strategy will be referred to as “cut-out+1”. While the intake system remains the same, a new control valve must be introduced between the primary pipe of the switched-off cylinder and the exhaust junction. Such a valve allows the fresh air to by-pass the catalyst.

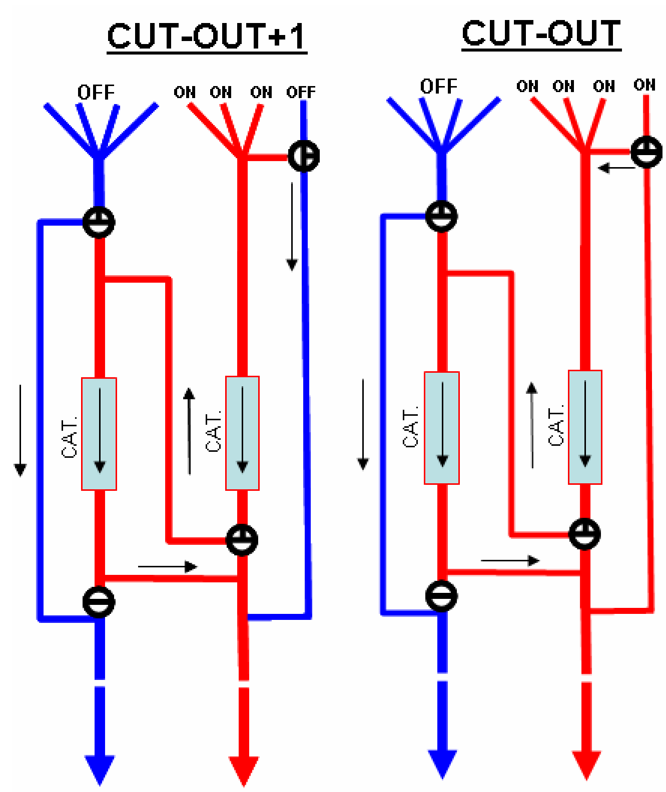

Figure 2: Schematic of the “cut-out+1” concept

Obviously, the “cut-out+1” strategy can be applied only at very low load, since the engine output is quite limited. A further advantage of this strategy is the possibility of using the airflow from the switched-off cylinder in the active bank as a secondary air injection, i.e. an addition of air to the products of a rich combustion, before entering the catalyst. For this purpose, the control valve opening must be modulated.

In the authors' opinion, the most critical issue related to this type of cylinder de-activation is the complex set-up of the control software, requiring a specific experimental campaign for each car application. Obviously, also the constructive complexity will grow (in particular, the material selected to build the control valves must be able to cope with the high temperatures of the exhaust system), but the increase of sophistication remains very limited in comparison to other de-activation strategies.

However, before going to the bench, it is fundamental to carry out a theoretical investigation, in order to assess the potential of the concept, as well as for optimizing the system lay-out.

1D cycle simulation is the most suitable tool for this type of study, being the software generally able to predict the influence of most engine parameters on performance. Even if this type of numerical approach has become a standard in engine design practice, in order to get reliable results it is fundamental to use experimentally validated simulation models, as well as to support 1-D calculations with experimental data, particularly on combustion and mechanical losses.

The authors have assumed a 4.2L V8 engine as a reference for the study, because of the availability of data. However, the results of the study should keep their validity also for different engines: the larger is the total capacity of the engine, the higher is the advantage ensuing to the analyzed cut-out strategy.

ENGINE MODEL

Engine cycle simulations have been carried out by using GT-Power, a 1-D code licensed by Gamma Technologies, Westmont, IL. [11]. The code considers the conservation equations of mass, momentum and energy in any network of pipes, volumes and junctions, modelled in terms of a set of building blocks. Engine cylinders, turbines and compressors can be attached to this network to serve as sources or absorbers of pulsating flows. The code provides a fully integrated treatment of time dependent fluid dynamic and thermodynamic parameters by means of a one-dimensional finite difference scheme, incorporating a general thermodynamic treatment of working fluids.

The engine analyzed in this study will be referred to in the following as V8. V8 is a 8 cylinder naturally aspirated S.I. engine produced by Maserati. It has two banks of cylinders, arranged on a V of 90 degrees, for a total displacement equal to 4244 cc. The combustion chamber is of the pentroof type, 4 valve per cylinder, central spark plug. The engine features a phase shifter on the intake valves, and constant geometry intake manifolds. One intake plenum is lodged between the banks, while the exhaust systems of each bank are separated. Load is controlled by one throttle valve, placed at the intake plenum inlet. Finally, the exhaust system of each bank includes: a pre-catalyst, a main catalyst, a silencer. The basic engine parameters are reviewed in table 1.

A comprehensive experimental campaign has been carried out on this engine at the dynamometer bench, at constant speed and load. The most relevant experimental data used to build the simulation model are listed below.

- Discharge coefficients vs. lift for intake and exhaust valves (measured at a steady flow bench).
- Equivalence ratio at each operating condition.
- Heat release vs. crank angle at each operating condition (evaluated from the in-cylinder pressure trace).
- Friction mean effective pressure at each operating condition (calculated as the difference between IMEP and BMEP)

Engine Lay-Out	*V8-90°*
Engine type	*S.I.*
Air metering	*Natural*
Fuel metering	*M.P.I.*
Bore [mm]	*92*
Stroke [mm]	*79.8*
Connecting rod length [mm]	*141*
Compression ratio	*11.3:1*
Number of valves	*4*
Combustion chamber	*Pentroof*
Top Power [kW]@speed[rpm]	*275@7000*
Max. Torque[N.m]@speed[rpm]	*425@4500*

Table 1 – Engine features

The air cleaner and the silencers have been represented in a lumped fashion, aiming only to reproduce the pressure drop measured at the test bed. Conversely, each catalyst has been carefully modeled, since the calculation of wall temperature in warm-up simulation is a fundamental issue of this study. In particular, the catalyst model includes the influence of the main geometric parameters and materials [11]. Furthermore, all the temperatures along the exhaust system walls, from the engine head outlet to the silencers, are calculated in each simulation, considering the heat transfer from gas to ambient. The only term not accounted for in the wall temperature prediction is the heat released by chemical reactions in the exhaust pipes and in the catalysts. The authors acknowledge the importance of this term, but believe that its influence should not affect very much the difference between two simulations, carried out under the same conditions.

Some care is necessary also to model the intake plenum (visible in figure 3), since the volume is relatively small, thus the interference among cylinders are relevant. This component is modeled as a net of 9 sub-volumes connected by orifices, taking into account the main geometrical features of the actual geometry.

For partial load simulations, the only difference in the model is the throttle valve, modeled as an orifice, whose effective area is calibrated in order to match the experimental airflow rate. Obviously, the numerical inputs, such as equivalence ratio and heat release rate and friction losses are specific of each operating condition.

A first comparison between simulation and experiments at full load is presented in figure 4, where average pressures at the intake plenum and the catalyst inlet, volumetric efficiency, IMEP and BSFC are plotted against engine speed. The agreement between experiments and simulation is very good. The same accuracy is found when comparing in-cylinder pressure traces at four engine speed values, as shown in figure 5.

Figure 3: View of the V8 engine, showing the intake plenum.

Since the purpose of this study is to analyze some control strategies at partial load, the validation of the numerical model has been carried out also at these operating conditions. In figure 6 and 7, reference is made to a BMEP of about 6 bar , while in figures 8 and 9 the applied load is about 2 bar. The comparison between experiments and simulation is carried out on the same parameters considered at full load (manifolds pressure, volumetric efficiency, IMEP, BSFC, in-cylinder pressure trace at four speeds). Figures 6-9 demonstrate the good level of accuracy reached by the model even at low load.

The most critical issue found in the model calibration at partial load is the in-cylinder pressure trace during the compression stroke, at low engine speed: in order to match experiments, the compression ratio entered in the model ought to be slightly reduced from the actual geometric value (the maximum reduction is 20% at 2000 rpm, BMEP=2 bar). This phenomenon may be explained by the fact that high performance SI engines have very light piston rings, whose tightness depends on the in-cylinder gas pressure, which pushes the ring against the liner and the piston groove. At low load and speed, the gas pressure is not enough to prevent some blow-by, so that the effective compression ratio becomes lower than the geometric one.

SIMULATION OF THE EUROPEAN DRIVING CYCLE

In order to assess the potential of the analyzed control strategies, the experimentally validated engine model is used to simulate the operations in the New European Driving Cycle. Two different types of calculations have been performed.

First, a set of steady points representing the cycle elementary operations has been defined. The simulation of these points allows the authors to compare the analyzed control strategies in terms of fuel efficiency.

Second, transient engine simulations are performed, considering the very first Elementary Urban Cycle (i.e. the first 195 s of the cycle). This analysis, including a detailed treatment of the thermal transient, is focused on the calculation of catalysts wall temperature.

Some empiricism is necessary to find a set of steady points representing the cycle elementary operations, as well as to calculate, from these steady results, the global fuel consumption. The authors fully acknowledge the complexity of the actual engine control under transient operations, and the fundamental influence of such a control on fuel consumption. However, since the purpose of this study is just a comparison among strategies, for the sake of simplicity it is assumed that the engine is operated with stoichiometric mixture, even in the “cold” phase of the driving cycle. Furthermore, the influence of engine temperature on mechanical losses is neglected, thus the considered values of FMEP are referred to hot operations.

Processing the huge amount of experimental data obtained at a steady dynamometer bench, accurate maps for FMEP and heat release rate have been generated as a function of engine speed and load. These maps have been used to provide the input data to the simulation model. As far as FMEP is concerned, it has been assumed that mechanical losses do not change if some cylinders are de-activated. This hypothesis is fully consistent with the experimental evidence reported in [10]. It is highlighted that the definition of friction losses considered by the authors does not include pumping losses. For the modeling of combustion, the curves of heat release applied to the active cylinders in the cut-out configurations, are referred to an equivalent BMEP, computed according to equations (2) and (3), as explained in the following.

For the calculation of the total engine load, it is necessary to enter the following data: car total weight (M), front area surface (S), aerodynamic drag coefficient

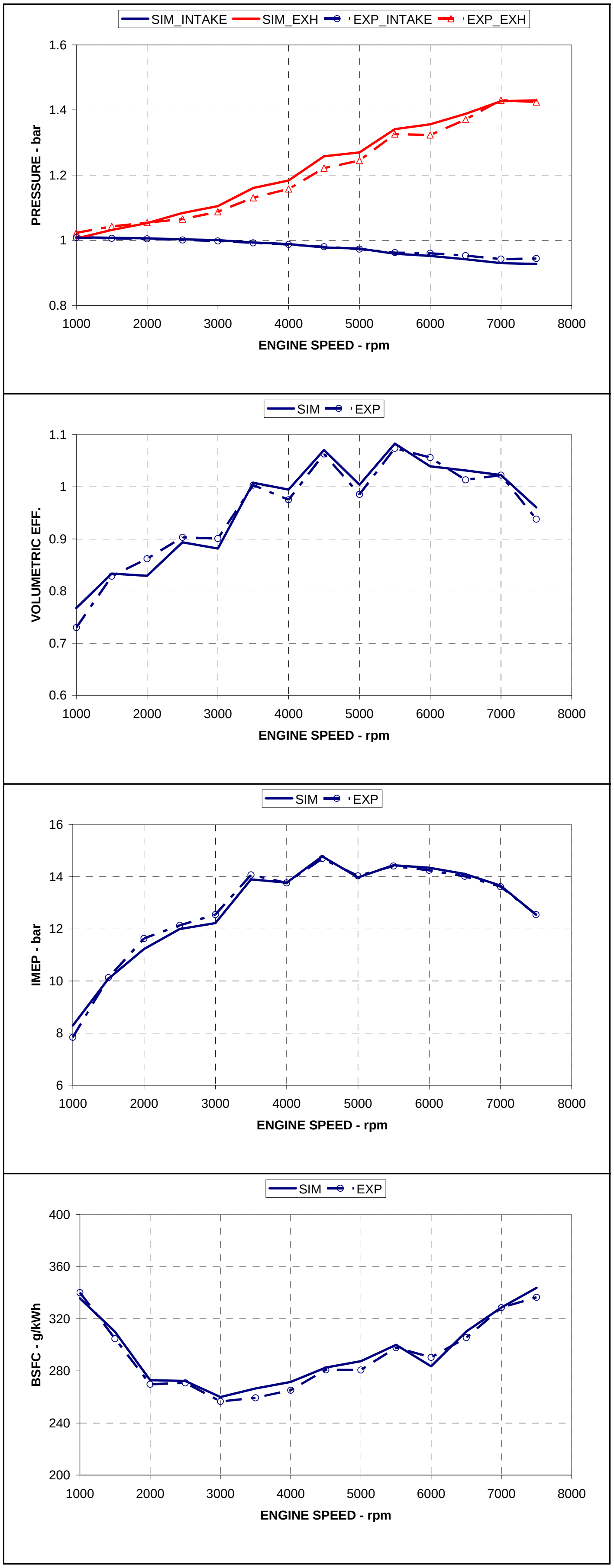

Figure 4: Comparison between simulation and experiments at WOT, average quantities.

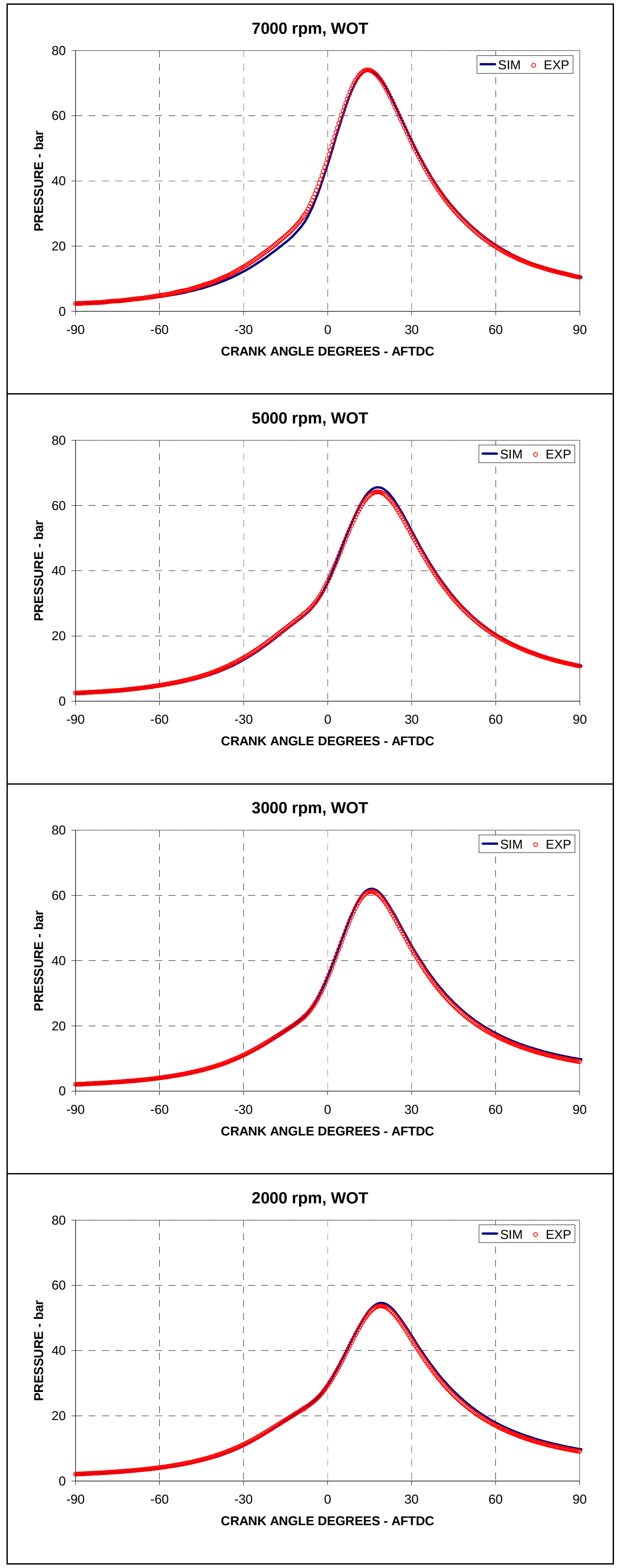

Figure 5: Comparison between simulation and experiments at WOT, in-cylinder pressure traces.

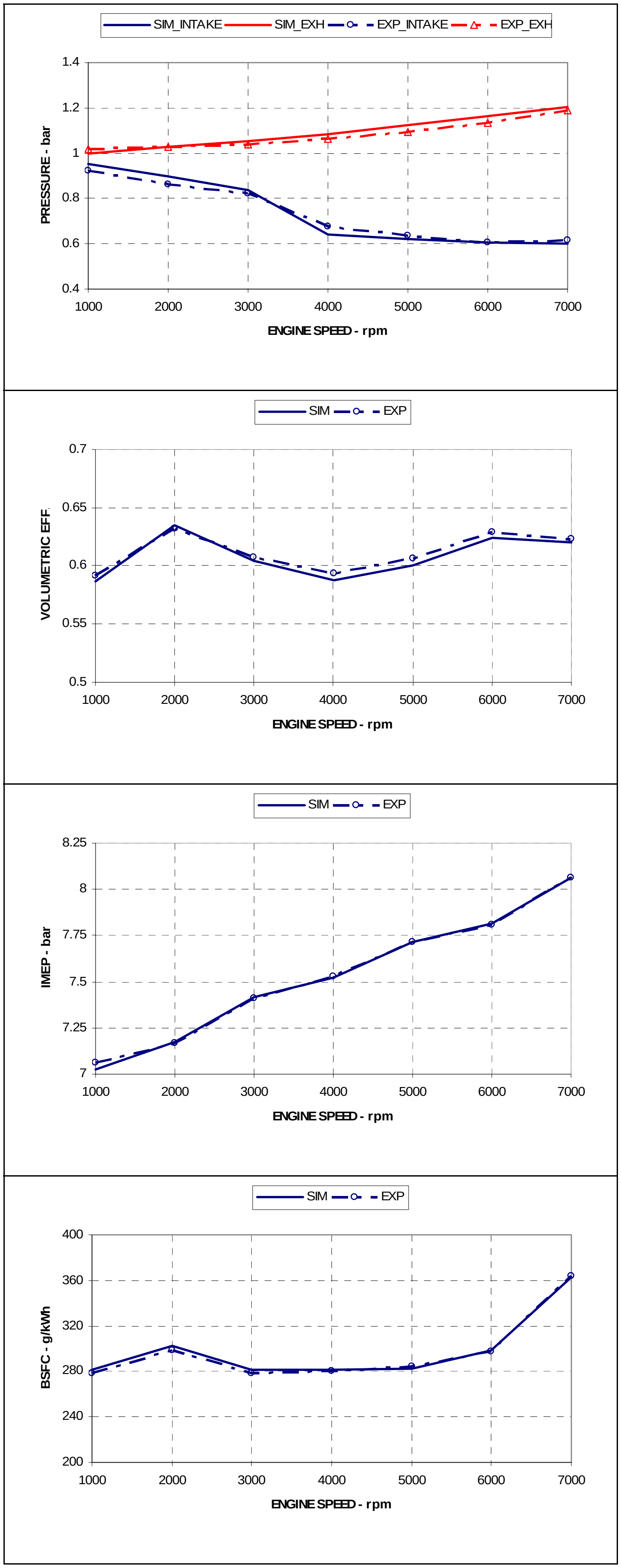

Figure 6: Comparison between simulation and experiments at partial load (BMEP = 6 bar), average quantities.

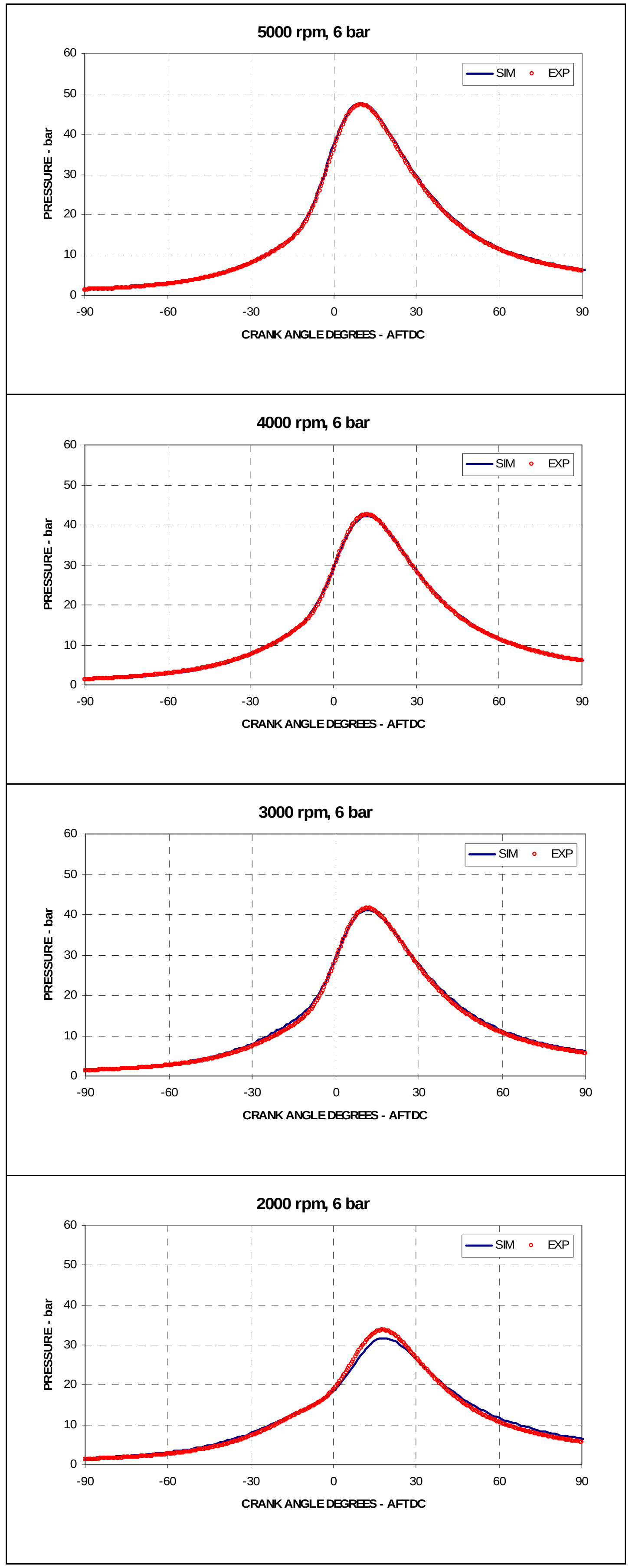

Figure 7: Comparison between simulation and experiments at partial load (BMEP = 6 bar), in-cylinder pressure traces.

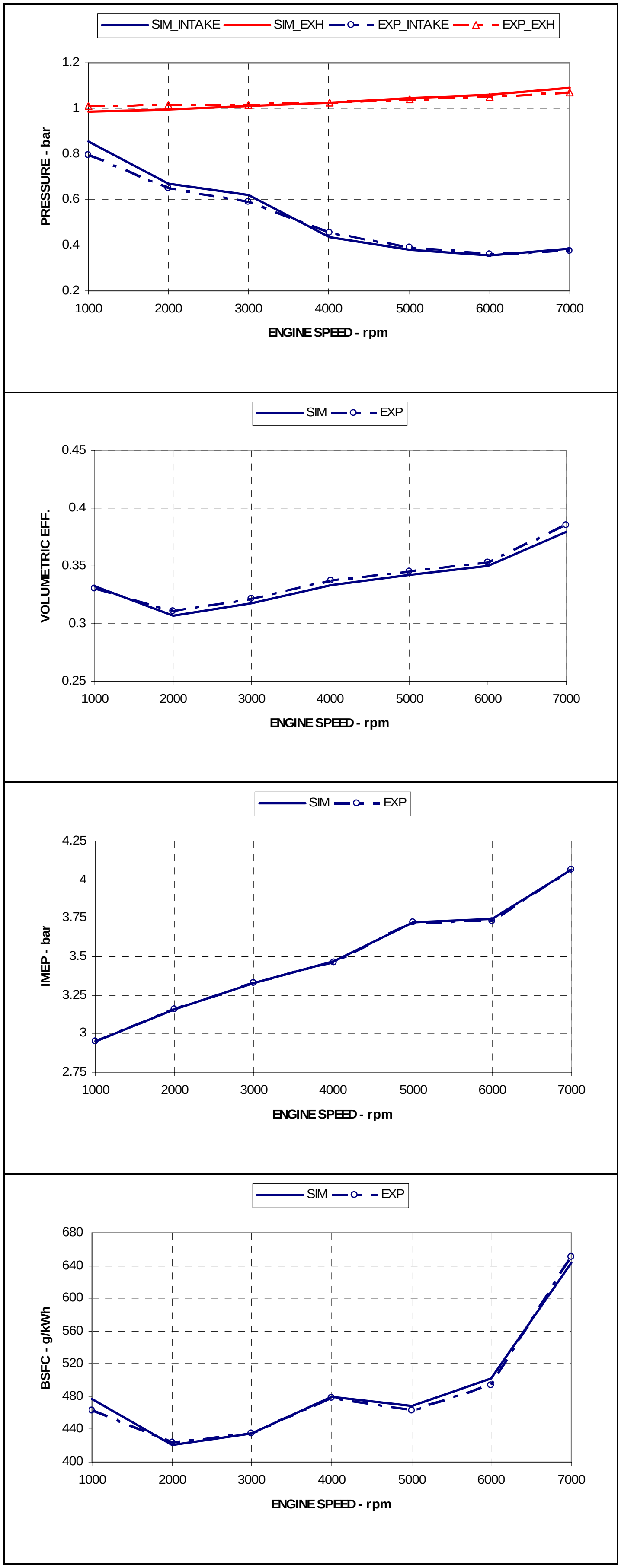

Figure 8: Comparison between simulation and experiments at partial load (BMEP = 2 bar), average quantities.

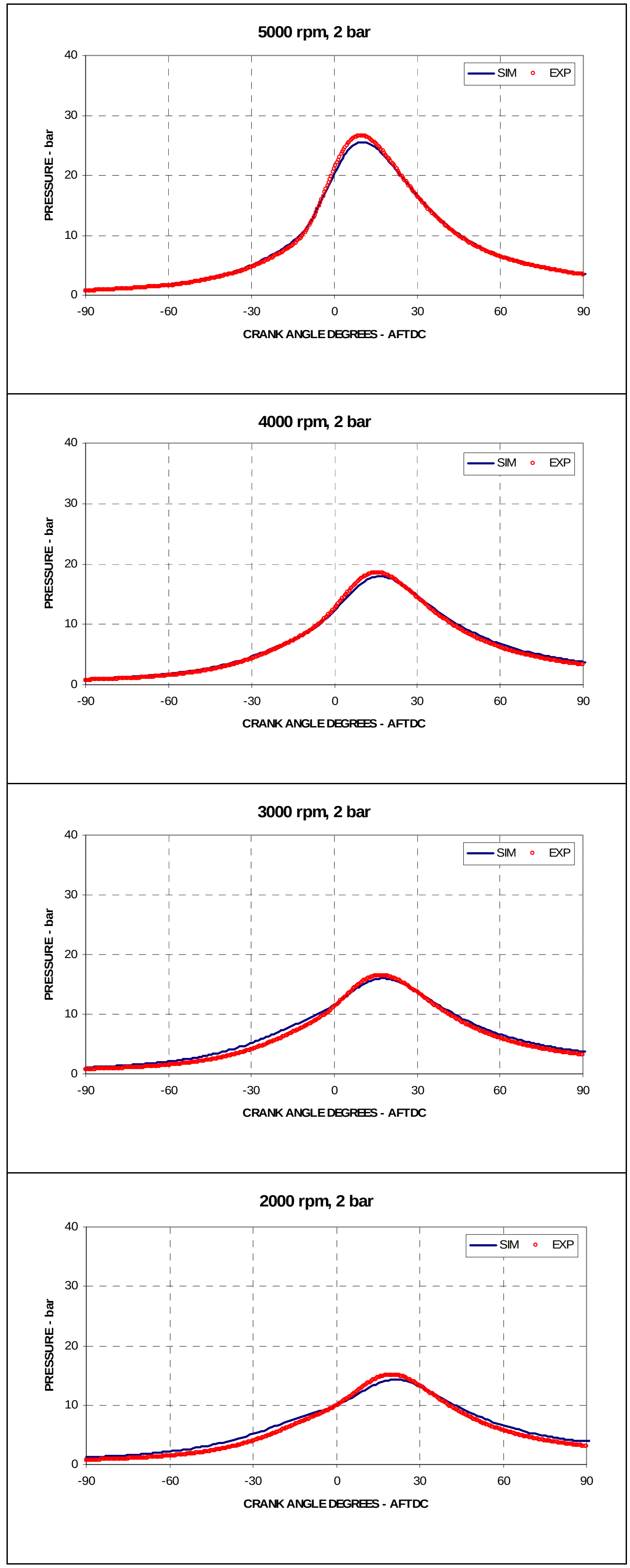

Figure 9: Comparison between simulation and experiments at partial load (BMEP = 2 bar), in-cylinder pressure traces.

(C_D), gear ratios, wheel diameter, tires rolling resistance (F_{roll}), driveline and engine inertia, driveline mechanical efficiency (η_{drv}). The values have been assumed considering a typical sporting car.

The data listed above, matched with a driving cycle chart, enable the calculation of engine speed and brake power at any point of the cycle. When the car is running at constant speed, referred to as 'v', the engine must provide a power given by:

$$P_{eng}(v) = \left[F_{roll} + \frac{1}{2}\rho C_D S v^2\right]\frac{v}{\eta_{drv}} \qquad (1)$$

Each elementary acceleration between two velocities (indicated as v_1 and v_2), is treated as an equivalent steady condition, represented by an average velocity and an average engine power. The latter is:

$$P_{eng}\left(\frac{v_1+v_2}{2}\right) = \left[\begin{array}{l} M\cdot\left(\dfrac{v_2^2-v_1^2}{2\Delta t}\right)+ \\ F_{roll}\cdot\left(\dfrac{v_1+v_2}{2}\right)+ \\ \dfrac{1}{8}\rho C_D S\left(v_1^3+v_1^2 v_2+v_1 v_2^2+v_2^3\right)\end{array}\right]\frac{1}{\eta_{drv}} \qquad (2)$$

The total car weight is corrected in order to account for the inertia of the engine and driveline, being the last one usually negligible in comparison with the car mass.

The analyzed cases are represented by a grid of 15 steady operations (9 points for constant speed conditions, 6 for the accelerations).

	Operations	**Speed**	**Pow.**
#	***Constant Speed***	[rpm]	[kW]
1	IDLE	750	0
2	15 km/h - I gear	1731	1.20
3	32 km/h - II gear	2400	2.86
4	35 km/h - III gear	1944	3.21
5	50 km/h - III gear	2778	5.31
6	50 km/h - IV gear	2206	5.31
7	70 km/h - V gear	2500	9.36
8	100 km/h - VI gear	2727	19.19
9	120 km/h - VI gear	3273	29.06
	Acceleration		
10	0-15 km/h, I gear, 4s	865	4.93
11	15-32 km/h, II gear, 5s	1763	14.31
12	35-50 km/h, III gear, 8s	2361	16.48
13	50-70 km/h, IV gear, 13s	2647	21.38
14	70-100 km/h, V gear, 35s	3036	24.86
15	100-120 km/h, VI gear, 20s	3000	40.72

Table 2: Equivalent operating conditions for the European Driving Cycle.

For the second type of calculations (i.e. the simulation of the first elementary Urban Driving Cycle), the previously used engine models have been modified, in order to account for transient operations.

First, engine speed and throttle position become time-dependent. While the former can be easily determined on the base of the driving cycle chart and the vehicle characteristics (gear ratios, wheel diameter), the latter requires some approximations. It is assumed that, at the beginning of each acceleration, the throttle is switched from the previous position to a new one, corresponding to the average acceleration load. Such a position is kept constant during the whole acceleration. During decelerations or gear shifting, throttle remains at the idle position. For the sake of simplicity, fuel cut-off is not included. According to these hypotheses, the throttle actuation law during the Elementary Urban Cycle is approximated through a sequence of steady conditions, that can be easily calculated apart.

For the prediction of exhaust wall temperatures, it is assumed that, when the engine is started, pipes and manifolds have the same temperature of the surrounding air. While the exhaust system warms up, the temperatures of piston, cylinder liner and engine head are kept constant throughout the simulation. This hypothesis is clearly a simplification. However, the influence on the comparison between two strategies should be almost negligible, while the degree of complexity of a detailed engine heat transfer model is very high. Furthermore, variable combustion chamber temperatures would affect brake performance, requiring a throttle adjustment, in comparison to the equivalent steady operating condition. Even if this could be done in the GT-Power model, the outcome seems not worth the effort.

Results of the transient simulations have been processed in conjunction with experimental emissions maps, in order to get an estimation of the pollutants quantity emitted by each analyzed configuration.

It is assumed that the concentration of each pollutant (CO, HC and NOx) in the exhaust flow corresponds to the value measured at the steady dynamometer bench, at the equivalent speed and load.

When considering the CUTOUT operations, the equivalent load of the active bank (indicated as $BMEP_{eq}$) can be calculated imposing the equivalence of the total effective work, i.e.:

$$BMEP_{eq} = 2\cdot BMEP + FMEP - (2\cdot PMEP_c - PMEP) \qquad (3)$$

$PMEP_c$ is the Pumping Mean Effective Pressure calculated in the CUTOUT configuration.

For the CUTOUT+1 configuration, i.e. with 3 active and 5 switched-off cylinders, the equivalent load becomes:

$$BMEP_{eq} = \frac{8}{3} \cdot BMEP + \frac{5}{3} FMEP - (\frac{8}{3} \cdot PMEP_c - PMEP) \quad (4)$$

It may be observed that the active bank in the CUTOUT configuration operates with a load that is much higher than the average BMEP. As a consequence, the pollutants concentration, in particular NOx, is higher than under normal operations. However, it should be also considered that only one bank is emitting pollutants, while the other is delivering clean air.

Once the equivalent load has been determined, the concentration of each pollutant at the engine outlet is derived by interpolation from a grid of experimental points.

Another relevant hypothesis made in the study is that conversion efficiency of each catalyst depends only on the brick temperature, calculated by the simulation code. The correlation between conversion efficiency and temperature is modeled by a simple parametric function, whose coefficients are calibrated on the base of data in literature [12].

The authors fully acknowledge that the absolute values of pollutants calculated by each simulation are affected by a number of approximations. However, the relative comparison among the strategies should be quite accurate.

INFLUENCE ON FUEL ECONOMY

A first set of simulations has been carried out on the grid of operating points shown in table 2. The following strategies are analyzed.

1. BASE: load is controlled by throttling the whole airflow

2. CUT-OUT: one bank of cylinder is switched off; load is controlled by the throttle plate on the active bank, while in the other bank the throttle is kept wide open; the exhaust system is arranged as in figure 1 or 2; valve timings are set as in the BASE configuration

3. CUT-OUT+1: the differences as regards the previous strategy are: a) one cylinder switched off in the active bank; b) ; exhaust system arranged according to figure 2

4. CUT-OUT-VVT: same as the CUT-OUT strategy, but intake and exhaust valve timings are optimized in order to minimize fuel consumption

5. CUT-OUT+1-VVT: same as CUT-OUT+1, but intake and exhaust valve timings are optimized in order to minimize fuel consumption

As far as the optimization of valve timings is concerned, the constraint of non interference between valves and piston crown has been considered, assuming no modification to the current combustion chamber geometry.

Table 3 shows a comparison between BASE and CUT-OUT in terms of fuel consumption. The percent improvement of fuel economy when passing from the former to the latter is also reported.

Table 4 presents the comparison between CUT-OUT and CUT-OUT+1, in the same terms of table 3.

Tables 5 and 6 review the results of another set of calculations, carried out in order to assess the influence on fuel efficiency of VVT in conjunction with CUT-OUT and CUT-OUT+1. This type of simulation is very time-consuming, since, for each steady operating condition listed in table 2, it is necessary to analyze no less than 20 combinations of intake and exhaust valve timings, even when using a DOE tool, as in this case. Therefore, the total amount of steady simulations required to analyze a single strategy is about: 20x15=300. Furthermore, for each analyzed combination, the throttle must be adjusted in order to meet accurately the BMEP target, thus the number of iterations required to reach convergence at each case is generally high (20 or more, depending on the imposed convergence tolerance)

	FUEL FLOW [kg/h]		
#	**BASE**	**CUTOUT**	**Improvement**
1	0.874	0.740	15.4%
	BSFC [g/kWh]		
2	2175	1947	10.5%
3	1466	1240	15.4%
4	1047	942	10.0%
5	1008	893	11.4%
6	783	713	9.0%
7	602	527	12.4%
8	421	380	9.8%
9	380	352	7.4%
10	471	413	12.3%
11	432	403	6.9%
12	466	425	8.7%
13	393	355	9.7%
14	377	350	7.2%
15	323	301	7.0%

Table 3: Comparison between BASE and CUT-OUT in terms of fuel economy.

Finally, in table 7, all the 5 configurations are compared in terms of fuel consumption in the NEDC, while table 8 shows the percent enhancement obtained as regards the BASE configuration.

On the base of the results presented in tables 3-8, the following considerations can be made.

- As expected, the larger improvement is obtained adopting the cut-out strategy from the base configuration (see table 3): the lower the load, the higher the enhancement.

- The de-activation of a further cylinder, beside the engine bank cut-out, produces little benefits in terms of fuel economy (see table 4). At a few points (3, and 7), fuel efficiency is even worse, while the target load cannot be reached at the operating condition #15. Obviously, for these operating conditions it's convenient to adopt a simpler CUT-OUT strategy

- VVT in conjunction with CUT-OUT (see table 5) helps a little bit to further reduce fuel consumption; benefits are larger at the lower loads (points 2-7)

- VVT is slightly more effective in conjunction with CUT-OUT+1 (see table 6); also in this case, differences are larger at the lower loads (points 2-7).

- As expected, the most effective VVT strategy is, in almost all the analyzed cases, a Late Exhaust Valve Closing combined to an Early Intake Valve Opening, thus a maximization of valve overlapping. Unfortunately, this parameter is limited by the interference between valves and piston crown.

- In the whole driving cycle, the CUT-OUT strategy improves fuel economy of 10.5%, while CUT-OUT+1 arrives to 12.8%; in conjunction with VVT, the former strategy produces a benefit of 11.9%, the latter of 14%

	FUEL FLOW [kg/h]		
#	**CUTOUT**	**CUTOUT+1**	**Improvement**
1	0.740	0.703	5.0%
	BSFC [g/kWh]		
2	1947	1790	8.1%
3	1240	1263	-1.9%
4	942	895	4.9%
5	893	885	0.9%
6	713	698	2.1%
7	527	532	-0.9%
8	380	375	1.3%
9	352	343	2.5%
10	413	384	7%
11	403	379	5.9%
12	425	416	2.1%
13	355	349	1.6%
14	350	340	2.7%
15	301	301	0.0%

Table 4: Comparison between CUT-OUT and CUT-OUT+1 in terms of fuel economy.

	FUEL FLOW [kg/h]		
#	**CUTOUT**	**CUTOUT-VVT**	**Improvement**
1	0.740	0.715	3.3%
	BSFC [g/kWh]		
2	1947	1907	2.1%
3	1240	1223	1.3%
4	942	935	0.7%
5	893	852	4.5%
6	713	697	2.3%
7	527	521	1.2%
8	380	375	1.3%
9	352	349	0.9%
10	413	411	0.5%
11	403	402	0.1%
12	425	412	3%
13	355	351	0.9%
14	350	344	1.6%
15	301	297	1%

Table 5: Comparison between CUT-OUT and CUT-OUT –VVT in terms of fuel economy.

	FUEL FLOW [kg/h]		
#	**CUTOUT+1**	**CUTOUT+1-VVT**	**Improvement**
1	0.703	0.678	3.5%
	BSFC [g/kWh]		
2	1790	1725	3.6%
3	1263	1214	3.9%
4	895	873	2.4%
5	885	854	3.5%
6	698	676	3.2%
7	532	514	3.3%
8	375	370	1.3%
9	343	343	0%
10	384	381	0.9%
11	379	375	1.1%
12	416	407	2.2%
13	349	346	0.8%
14	340	340	0.2%
15	301	301	0%

Table 6: Comparison between CUT-OUT+1 and CUT-OUT+1-VVT in terms of fuel economy.

	Urban consumption [l/100km]	**Extra-urban** consumption [l/100km]	**Combined** consumption [l/100km]
BASE	17.97	11.97	14.18
CUTOUT	15.91	10.82	12.69
CUTOUT+1	15.31	10.66	12.37
CUTOUT-VVT	15.63	10.67	12.5
CUTOUT+1-VVT	15.1	10.5	12.19

Table 7: Influence of the analyzed strategies on the fuel consumption in the New European Driving Cycle

	Urban improvement	**Extra-urban** improvement	**Combined** improvement
CUTOUT	11.5%	9.6%	10.5%
CUTOUT+1	14.8%	11%	12.8%
CUTOUT-VVT	13%	10.9%	11.9%
CUTOUT+1-VVT	16%	12.3%	14%

Table 8: Fuel efficiency improvement obtained in the NEDC by applying the analyzed strategies

INFLUENCE ON THE CATALYST LIGHT-OFF

The CUT-OUT and CUT-OUT+1 strategies, described in the previous sections, are applied to the simulation of the first elementary urban cycle of the NEDC (i.e. for a transient calculation, 195 seconds long). The BASE configuration is taken as a reference.

Figures 10-13 present the calculated temperature histories of the pre-catalyst and catalyst monoliths, in both the active and the switched-off bank.

When considering the active bank (figures 10 and 11), the advantage of the cut-out strategies is more than evident: assuming the light-off temperature at 500 K, with CUTOUT the pre-catalyst starts to operate 25 seconds before BASE, while CUT-OUT+1 gains 5 further seconds. Even more dramatic is the difference for the main catalyst: in the base configuration, it is never active, while with CUTOUT and CUTOUT+1 the light-off occurs after 125 and 80 seconds, respectively.

In the switched-off bank (figures 12 and 13), it is observed that the analyzed strategies allow the monoliths temperature to reach the values of the base configuration well before the end of the first elementary urban cycle. Thus, it is reasonable to deduce that, in the actual usage of the car, there will be no significant draw-back, caused by the lower temperature in the non-active bank.

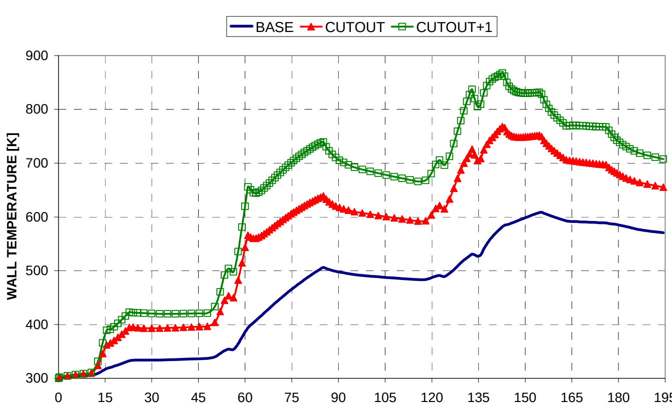

Figure 10: Temperature history of the active bank pre-catalyst monolith calculated during the first elementary urban cycle of NEDC

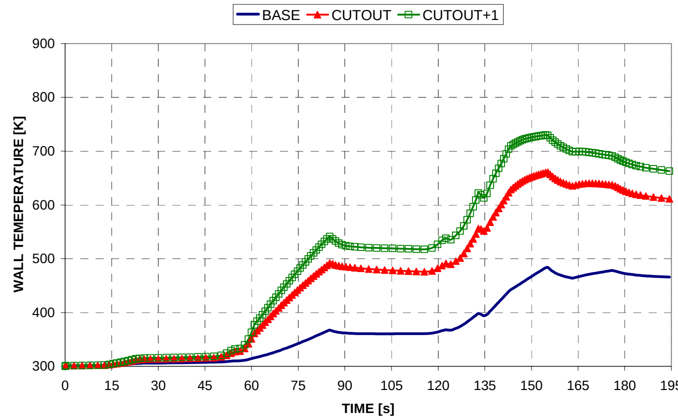

Figure 11: Temperature history of the active bank catalyst monolith calculated during the first elementary urban cycle of NEDC

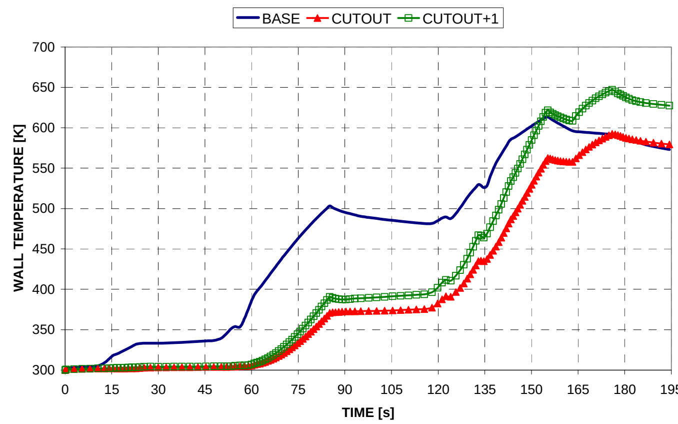

Figure 12: Temperature history of the non-active bank pre-catalyst monolith calculated during the first elementary urban cycle of NEDC

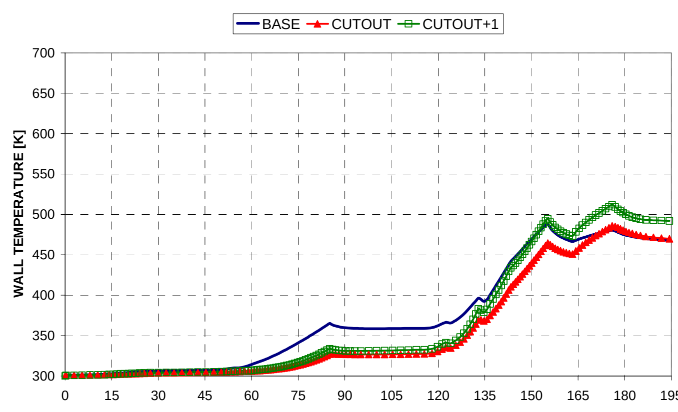

Figure 13: Temperature history of the non-active bank catalyst monolith calculated during the first elementary urban cycle of NEDC

The exhaust system predicted temperatures have been used to calculate the conversion efficiency of each catalyst, throughout the transient simulation. The global conversion efficiency of the whole exhaust system is then defined as:

$$\eta_{conv,TOT} = 1 - \prod_i \left(1 - \eta_{conv,i}\right) \tag{5}$$

$\eta_{conv,i}$ is the conversion efficiency of each catalyst along the gas flow path.

Figure 14 shows a comparison among the strategies in terms of global conversion efficiency. The lead of both CUT-OUT and CUT-OUT+1 from BASE is large. Two issues contribute to this outcome: the higher catalysts temperature and the doubling of the number of catalysts along the exhaust gas path.

Finally, the emissions at the tail-pipe have been assessed on the base of the experimental pollutants rates and the calculated catalysts conversion efficiency. Figure 15 reviews the results, normalized with reference to the amount of pollutant produced by the BASE configuration. It is observed that both CO and HC are strongly reduced by the adoption of a CUT-OUT or a CUT-OUT+1 strategy, while for NOx the gain is less evident. This result is explained by the fact that the higher is the load, the higher the concentration of NOx in the exhaust gas: thus, the active cylinders in CUTOUT or CUTOUT+1 strategy, produce more Nitrogen Oxides than all the cylinders in the BASE configuration.

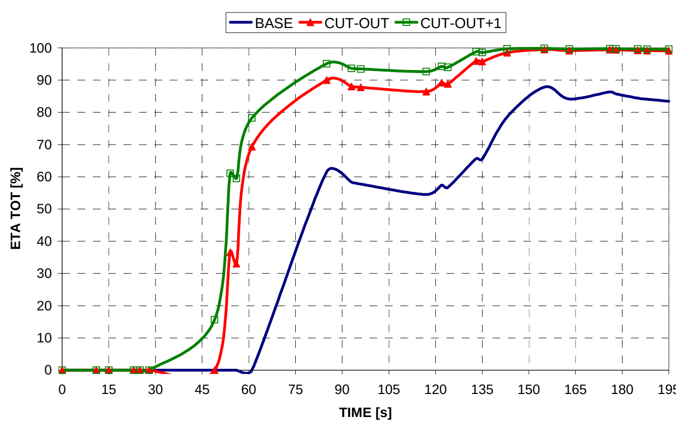

Figure 14: History of global conversion efficiency for the three analyzed strategies calculated during the first elementary urban cycle of NEDC

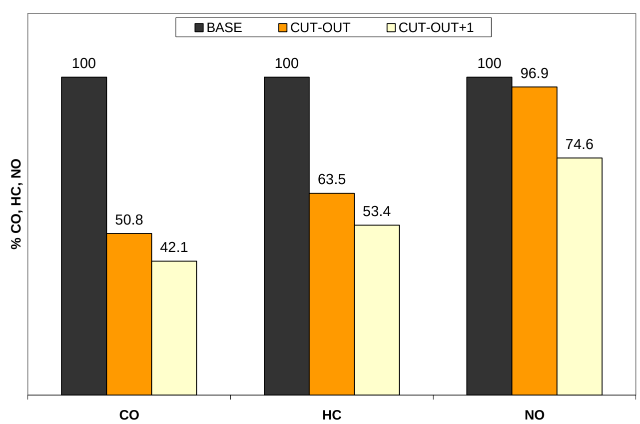

Figure 15: Predicted values of tail pipe pollutant emissions normalized with reference to BASE strategy

CONCLUSIONS

Two cylinders deactivation strategies have been analyzed on a 4.2L, S.I. V8 high performance engine: a strategy called CUT-OUT, with one active and one switched off bank, and a configuration named CUT-OUT+1, with only three active cylinders over eight. In both cases, the exhaust system is arranged so that cold air bypasses the catalysts, while the exhaust gases are routed through all four the catalysts.

After the experimental validation of the engine 1D thermo-fluid-dynamic model at both full and partial load, the influence on fuel economy of the analyzed control strategies has been studied on a set of 15 steady points representing the NEDC driving cycle elementary operations. Results have shown an enhancement of fuel efficiency at every operating condition, and a further margin of improvement by coupling deactivation with VVT. The calculated reduction of fuel consumption in the New European Driving Cycle is up to 16% for the urban cycle and 12% for the extra-urban one, as shown in Table 8.

The influence on the catalysts light off has been also studied, by simulating the first elementary urban cycle of the NEDC. CUT-OUT and CUT-OUT+1 strategies present a much faster light off than BASE; this issue, associated to the doubled effective surface of the catalytic bed, heads to the strong reduction of exhaust tail pipe emissions visible in Figure 15.

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