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PROMOTING SYNERGIES BETWEEN TURBOCHARGERS AND VARIABLE VALVE TIMING ACTUATION TO REDUCE FUEL CONSUMPTION OF GASOLINE ENGINE

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Abstract: *The reduction of fuel consumption is a key point for automotive car makers. One major strategy they implement is to replace big engines with large displacement by turbocharged smaller ones, the so-called downsizing concept. In addition to the turbocharger, Variable Valve Timing Actuation systems (VVTA) are very useful to push forward the low end speed limits and, thus, to reduce the fuel consumption of the vehicle. VVTA promote the scavenging process of the cylinder head that is firstly very helpful for the combustion. In the meantime, this scavenging process increases also the exhaust mass flow rate through the turbine and thus increases the compressor power, the boost pressure and the engine performance. This process is illustrated with a 4 cylinder GDI engine and a 3 cylinder MPI engine during full load steady test and during transients. According these examples, some limits with regards emissions will be raised. We will present secondly tools that can be used to study, in one hand, the scavenging process during engine tests and, in the other hand, the turbine. Some limits are also listed.*

Keywords: engine; scavenging; turbocharger; Variable Valve timing.

INTRODUCTION

The reduction of CO2 emissions is mandatory to limit earth warming. As 13 % of greenhouse gases are coming from transportation (1), governments are setting more and more stringent regulations to automotive manufacturers. Among all the solutions that are envisaged, one of the most affordable technologies is the replacement of engines with big displacement by smaller turbocharged ones. Thus, in Western Europe, in 2015, with the growing share of gasoline turbine, one can estimate that 75 % of engines will be turbocharged. Indeed, the so-called downsizing concept leads to a reduction of the pumping losses of gasoline engine, thermal losses and of friction during usual driving conditions. However, turbocharger technologies suffer from a lack of performance at low engines speed due to the reduced mass flow rate available to the turbine. The low steady torque and, more important, the transient one are limitations to increase the gear ratios; this drivetrain optimization is also an efficient way to decrease fuel consumption (also called down-speeding strategy).
For gasoline engines, one counter-measure to overcome these low end speed issues is to add variable valve timing actuation systems (VVTA) to the engine technical definitions. Indeed, these systems implemented at intake and exhaust allows to control:

- the intake closure valve timing (IVC) that determine, in first hand, the quantity of fresh air that is trapped into the cylinder and, in second hand, the effective compression that is a key parameter for the knock limitation;
- the exhaust opening valve timing (EVO) that is so important for the blow-down process and its PMEP consequences and for the effective expansion ratio;
- by controlling the overlap period (IVO/EVC, Intake valve opening and exhaust valve closure), these systems allows to control the scavenging process. This overlap period must take into account the vicinity of the piston.

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As for two stroke engines, the scavenging process done by fresh air allows to decrease the internal burnt gases (IGR) remaining at the end of the exhaust phase. Thus the knock sensitivity is decreased, more advanced spark timing can be applied and better torque can be obtained. Furthermore, the scavenging process allows an increase of mass flow rate through the turbine.
At high engine speed, the exhaust backpressure is often so high that the scavenging process is replaced by a backflow of exhaust gases into the intake ports. To avoid the resulting reduction of fresh air and the increase of burnt residual gases, the overlap period have to be reduced.
At low engine speed, as usual turbocharger technology for gasoline engines are fixed geometry ones, boosting pressure is close to the one before turbine. The scavenging can be promoted if the exhaust pressure pulses are correctly phased with regards to the overlap period. If possible, the increased mass flow rate due to the scavenging has big impacts on the available turbine power and, thus, the deduced boosting ratio.
In the first part, based on test bench data, we will describe the scavenging process and its consequences of a 4 cylinder GDI engine and a 3 cylinder MPI engine. In the second part of this paper, we will describe methodologies that can be applied to understand and predict test bench results of four-stokes gasoline turbocharged engines. We will explain their fundamentals and the limitations of those approaches.

EXPERIMENTAL ILLUSTRATIONS OF THE SYNERGIES

Within this part, we will illustrate first the effect of VVT on the turbocharger behavior on gasoline engines. As scavenging process is highly dependent to the instantaneous pressure ratio at intake and exhaust valves, we will describe the situation of 4 cylinder and 3 cylinder engines. These illustrations are based on the Renault experience of TCE 115 and TCE 90 development.

Table 1. Main characteristics of studied engines.

TCE 115	TCE 90
1.2 L GDI / 4 cyl engine / 16 valves Bore x Stroke : 72.2 /73.1 Compression Ratio : 9.5 :1 2 VVT	0.9 L MPI / 3 cyl engine / 12 valves Bore x Stroke : 72.2 x 73.1 Compression Ratio : 9.5 :1 1 intake VVT

SCAVENGING AND 4 CYLINDER ENGINES

As shown in Figure 3, with fixed valve setting, the scavenging period (figure out by the green rectangle) must be rather short due to idle requirements. As there is a coincidence between intake phase duration and the period between all exhaust blow-down, a maximum exhaust pressure occurs during the overlap. Both at low engine speed and at high engine speed, this shape of exhaust pressure promotes exhaust back flow and poor volumetric efficiency. A short exhaust valve event can then help to get a lower pressure during overlap but, obviously, has some drawbacks of PMEP at high engine speed.

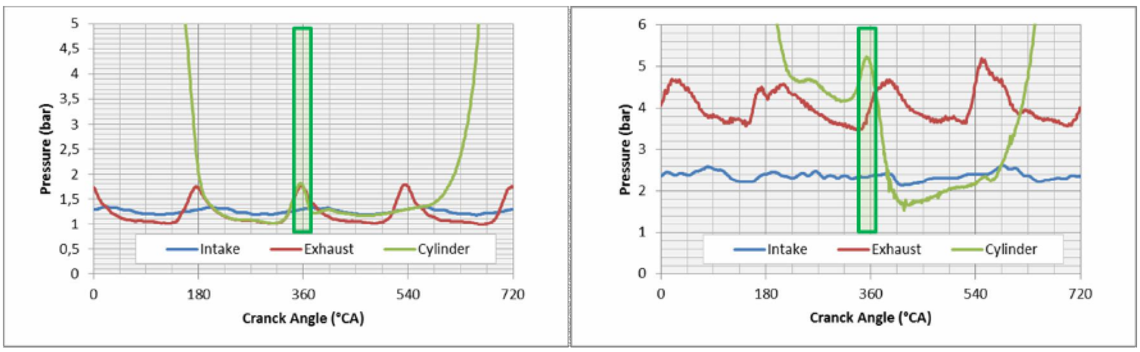

Figure 3. Pressures with no VVT.
(left : 1500 rpm – right : 5500 rpm)

When VVTA systems are implemented, at low engine speed, an anticipated IVO allows setting the overlap period when intake pressure is above the instantaneous exhaust one (Figure 4). This advanced IVO has also the advantage to have the IVC close to the BDC and thus to avoid an intake backflow at the end of the intake phase. By retarding the EVO that is then close to the TDC, the overlap period is increased. As the exhaust pressure rise occur later, scavenging is still promoted. Maximum intake mass flow rate that has been calculated with the basic method describe later is reached during the overlap period.

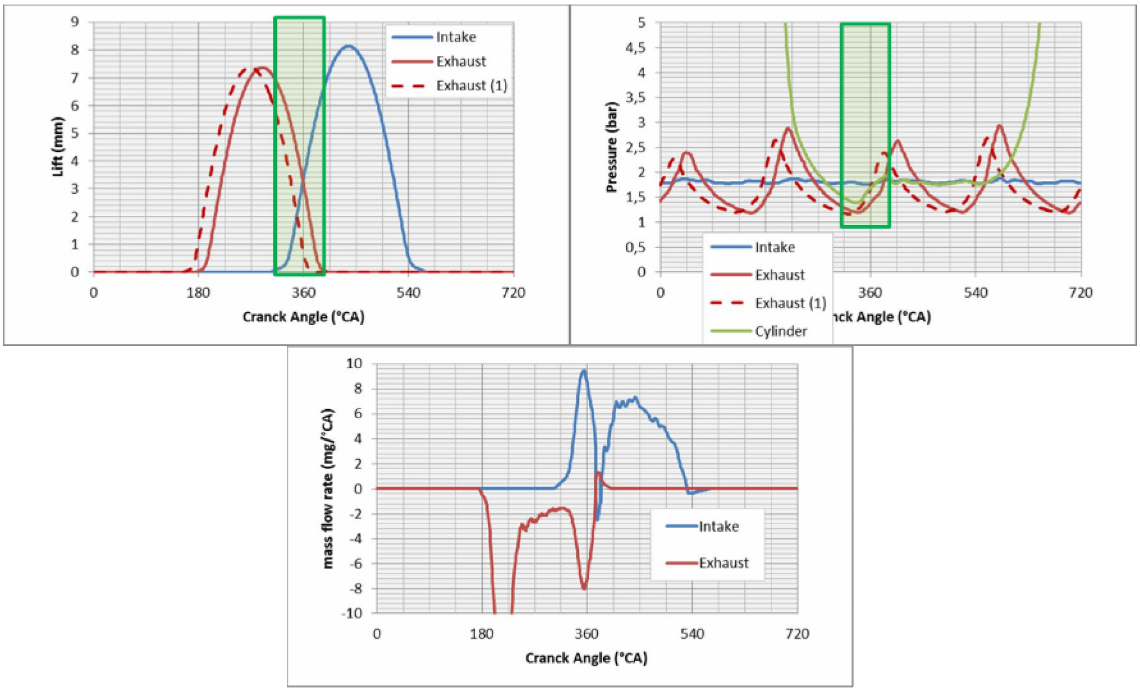

Figure 4. Valve lift, pressures and mass flow rates with VVTA systems at 1500 rpm on 4 cylinder engines.
Dashed line: exhaust initial position – plain line: final setting

Figure 5 show the transformation of the transient behavior of the turbocharger and, thus, the engine with VVTA systems. The synergy between VVTA and turbocharger leads to an engine that has roughly 50 % more apparent displacement at 1500 rpm. At lower engine speed (1000 rpm), the improvement is still consequent: 20 %.

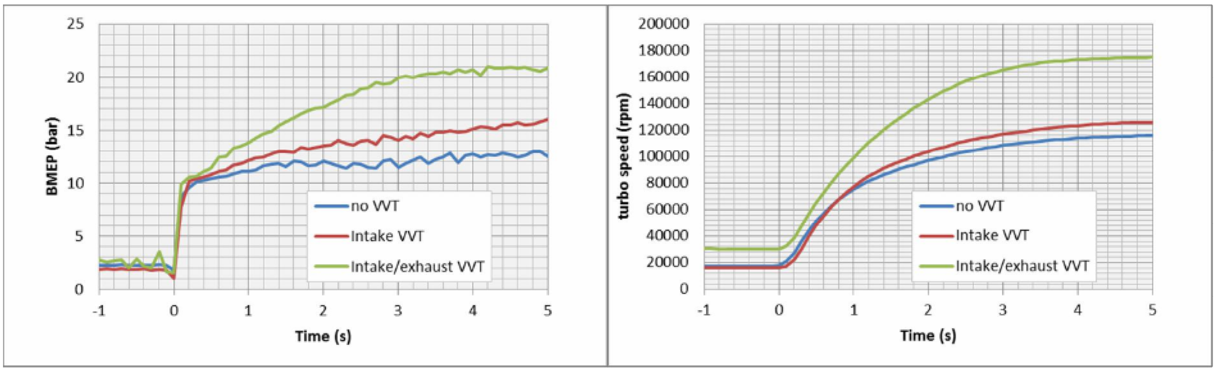

Figure 5. Effect of VVTA systems on tip-in at 1500 rpm on a 4 cylinder engine.

SCAVENGING AND 3 CYLINDER ENGINES

At low engine speed, Figure 6 shows that the exhaust pressure rise due to the adjacent blow down occurs naturally after the overlap period, the scavenging is thus naturaly very high and can be promoted by an advanced IVO. At high engine speed, as the averaged pressure before turbine is around 3.2 bar (concequence of turbo matching choice and back pressure), the exhaust pressure is higher than the intake one. Scavenging must be avoided. However one can notice that the instanteneous exhaust pressure is at its lowest level during the overlap; that will have a positive effect on the IGR content and then the combustion.

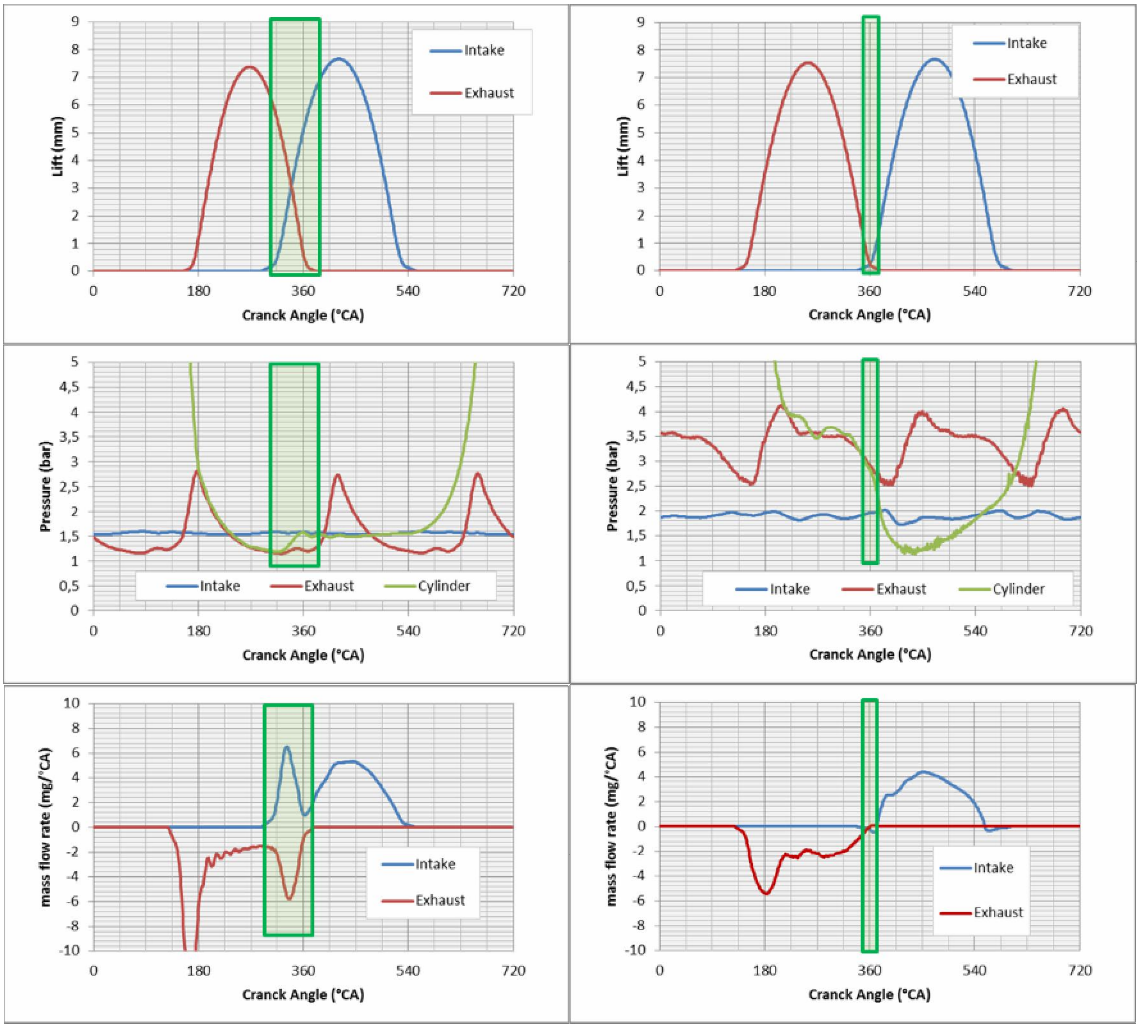

Figure 6. Effect of the intake VVTA on a 3 cylinder engine.
(Left: 1500 rpm – Right: 5500 rpm)

By just applying the intake VVTA system, the transient behavior is also transfigured (Figure 7). An equivalent of 20% increase of apparent displacement is obtained.

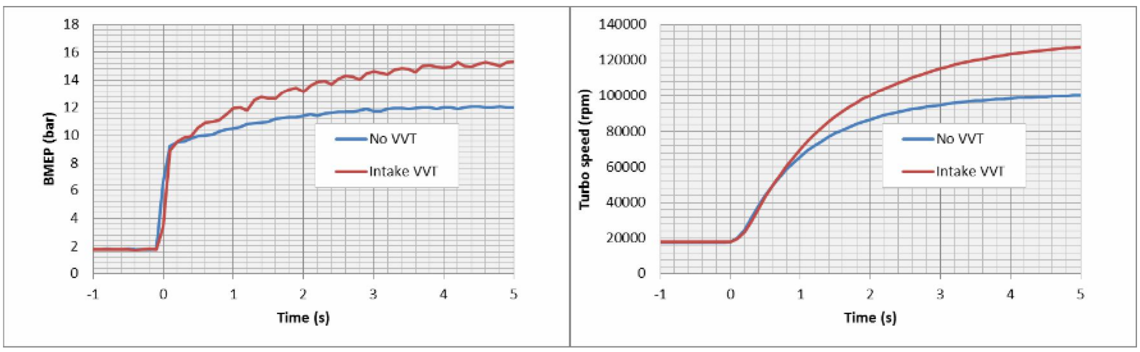

Figure 7. Effect of VVTA systems on tip-in at 1500 rpm on a 4 cylinder engine.

Figure 8 illustrates the limited interest of adding an exhaust VVTA system on a MPI engine. Indeed, as the injection is done within the intake ports when valves are closed, the fresh air that goes directly to the exhaust incorporates fuel vapor. Even with the intake VVT, the increase of the plenum volumetric efficiency through the scavenging leads to a 15 % of the BSFC. On MPI engines, there is a compromise between the high torque level and the increased BSFC.

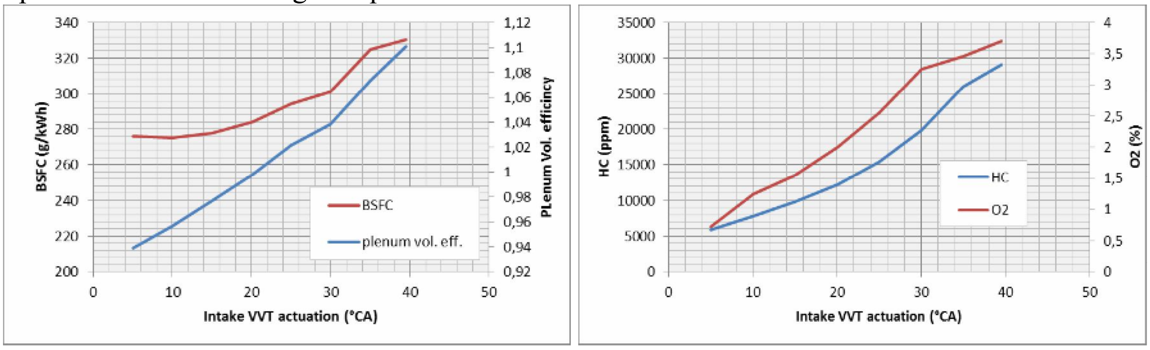

Figure 8. Effect of VVT actuation on BSFC and HC/O2
at 2000 rpm/16 bar.

The implementation of the second VVTA at the exhaust side is only interesting for GDI engines

METHODOLOGY TO STUDY THE SYNERGIES BETWEEN SCAVENGING AND TURBOCHARGERS

We have described how effective can be a scavenging process involving VVTA systems and turbochargers. In order to well optimize their synergy, we will describe 0D tools based on test bench measurements and that also the base of simulation software.

STUDY OF THE SCAVENGING PROCESS

Basis

The scavenging process is generally studied through a three pressure setup (intake, exhaust and in-cylinder instantaneous pressure sensors). As described in Figure 9, a model will determine the different mass flow rates.

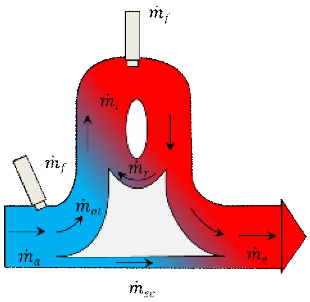

Figure 9. scheme of general scavenging process

Where:

- $\dot{m}_a$ is the intake mass flow rate;
- $\dot{m}_f$ is the fuel mass flow rate that can occur before the pumping loop for MPI engines or in the cylinder for GDI engines;
- $\dot{m}_e$ is the exhaust mass flow rate;
- $\dot{m}_{sc}$ is the scavenged flow rate;
- $\dot{m}_{at}$ is the air trapped mass flow rate;
- $\dot{m}_t$ is the air trapped mass flow rate;
- $\dot{m}_r$ is the residual gas mass flow rate.

Those mass flow rates are linked by the following equations:

$$\begin{aligned} \dot{m}_a + \left(\dot{m}_f \ if \ MPI\right) &= \dot{m}_{at} + \dot{m}_{sc} \\ \dot{m}_t &= \dot{m}_{at} + \dot{m}_r \\ \dot{m}_e &= \dot{m}_a + \dot{m}_f \end{aligned} \tag{1}$$

The intake and exhaust mass flow rates, $\dot{m}_a$ and $\dot{m}_e$, can be calculated according the Barré Saint Venant equation for compressible flows through an orifice:

$$\dot{m} = \frac{C_d \cdot A_r \cdot P_0}{\sqrt{r \cdot T_0}} \cdot \left(\frac{p_t}{p_0}\right)^{1/\gamma} \cdot \sqrt{\frac{2 \cdot \gamma}{\gamma - 1} \cdot \left[1 - \left(\frac{p_t}{p_0}\right)^{(\gamma-1)/\gamma}\right]} \tag{2}$$

Where:

- $C_d \cdot A_r = S_e$ is the experimentally determined effective area of the valve for the given lift;
- P_0 and T_0 are respectively the upstream stagnation pressure and temperature;
- p_t is the downstream static pressure assumed to be the one at the restriction
- γ and r are the ratio of specific heats and gas constant of the upstream gases.

When flow is choked, i.e $p_t/P_0 \leq \left[2/(\gamma - 1)\right]^{\gamma/(\gamma-1)}$, the mass flow rate is blocked at a value:

$$\dot{m} = \frac{C_d \cdot A_r \cdot P_0}{\sqrt{r \cdot T_0}} \cdot \sqrt{\gamma} \cdot \left(\frac{2}{\gamma + 1}\right)^{(\gamma+1)/2\cdot(\gamma-1)} \tag{3}$$

To determine the upstream gas composition, the mixing process must be modeled as in Figure 10.

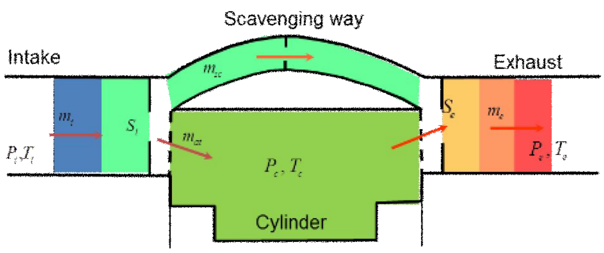

Figure 10. Mixing process.

Within intake and exhaust ports, volumes are defined. As for the cylinder volume, their inner compositions in burnt gases and fresh air are derivate according the mass flow rates and the perfect mixing assumption.

The scavenging way is determined through an assumption of a fixed fraction of the valves that communicates directly.

Difficulties encountered with scavenging models

If the methodology seems simple, many issues can be encountered:

1. This methodology is based on equation (2) and thus the measurement of intake and exhaust pressure: instantaneous sensors have to be implemented as close as possible to the valves to avoid delay between their implementation and the valves. Cares have also to be taken in order to measure the averaged static pressure in the section: as far it is possible, flow recirculation zones have to be avoided for the implementation;
2. Dynamic pressure is rarely taken into account for the upstream stagnation pressure used in equation (2). That can lead to some deviation during the exhaust blow down or scavenging process

3. Those pressure sensors should be absolute ones. However, most of the time, and especially for the cylinder ones which is often a piezoelectric type, they are relative and their static level must be corrected. For intake and exhaust ports, static pressure measurements can be implemented at the same position and same cares as the instantaneous ones. For the cylinder one, the correction can be set at the end of the intake phase or during exhaust phase when the valve pressure drop is low;
4. The experiments to evaluate the port/valve permeability $S_e = C_d \cdot A_r$ are based on flow bench setup. The pressure drop in such installations is often set at a rather low level (~ 50 mbar) corresponding on the average condition valves see during intake or exhaust phases. However, during scavenging phase with turbocharger, the pressure drop at valves can increase up to 300 mbar. Some authors (2) (3) show that discharge coefficients can be highly affected by this increase of the pressure ratio;
5. The experiments to evaluate the port/valve permeability $S_e = C_d \cdot A_r$ are often done without the presence of the piston. During the scavenging process, the piston and valves are very close. Effective area is affected by their interference especially when valve pockets are designed on the top of piston. Unsteady CFD calculations can help to set properly the changes of effective area;
6. The assumption of perfect mixing is also doubtful. Flows during scavenging have really 3D shapes. Thus it is quite difficult to know what fraction of valves correspond to the scavenging way. Perhaps that's why widely used commercial software, TPA/GT Power from Gamma Technology (4) or GCA/Boost from AVL (5), does not take into account this way;
7. The scavenging process is highly affected by the "on-line" valve timing. It can differ from the theoretical one due to mounting uncertainties, different thermal expansions of valves and cylinder head that can change the valve lash, compressible effect on hydraulic lash compensation systems, wear of the valve train that affects the ramps;
8. Heat transfer within the cylinder affects also a lot mass flow rates at low engine speed. As cycle time is long, the hot surfaces of cylinder head, liner, valves, piston and ports increase the in-cylinder temperature and thus decrease the trapped mass at the end of the intake phase. The prediction of these temperature are important: they can be modelled with simplified approaches (6) but need a lot of data.

The 5 last issues are also valid for simulation software and limit highly the validity of the predictions.

STUDY OF THE TURBOCHARGER PERFORMANCE AT LOW END SPEED

Study

In parallel of gas exchange process, it is meaningful to study the turbocharger behavior. Obviously, placing the operating point in the compressor map is interesting but the main interest is the turbine behavior.

Placing the averaged operating points on the turbine map can help to see how far the saturation zone at low engine speed is. Indeed, as the exhaust mass flow rate is low, the expansion ratio is not increasing rapidly.

The calculation of apparent efficiencies of compressor, η_c, or turbine, η_{tm} and global turbocharger efficiency, η_g, is also useful to approach pulsation effects:

$$\eta_c = \frac{T_{2is} - T_1}{T_2 - T_1}$$
$$\eta_{tm} = \left(1 + \frac{\phi}{R_{\frac{A}{F}st}}\right) \cdot \frac{c_{p_e}}{c_{p_i}} \cdot \frac{T_2 - T_1}{T_3 - T_{4is}} \tag{4}$$
$$\eta_g = \eta_c \cdot \eta_{tm}$$

where T_1, T_2 and T_3 are the averaged measured temperatures respectively before the compressor, after the compressor and before the turbine. T_{2is} and T_{4is} are the theoretical isentropic temperature.

$$T_{2is} = T_1 \cdot \left(\frac{P_2}{P_1} \right)^{(\gamma_i - 1)/\gamma_i}$$
$$T_{4is} = T_1 \cdot \left(\frac{P_3}{P_4} \right)^{-(\gamma_e - 1)/\gamma_e} \tag{5}$$

By comparison with averaged efficiencies obtained through maps, unsteady factors can be evaluated. Especially at the turbine side, these factors show the influence of pulsations and mechanical losses changes.

Difficulties in turbocharger performance evaluation

The main difficulties are coming from the extreme sensitivity of the turbine performance at low engine speed:

1. Atmospheric pressure, intake manifold temperature regulation affect the global mass flow rate. A 10 mbar increase or a deviation of 3°C of the intake manifold temperature can have more than 2% effect on the boost pressure;
2. An oil temperature change affects the oil viscosity and thus the apparent turbine efficiency that globalized thermodynamic efficiency and mechanical efficiency. For instance, a 10 °C deviation around 90 °C lead to a change of 25 % of the viscosity and thus the bearing losses. As mechanical losses power represent roughly 50% of the turbine power (7);
3. To assess transient behavior of turbochargers, the initial thermal condition of the wall temperature before tip-in has to be keep carefully under control (8);
4. The prediction of turbine power cannot be directly derivate from usual steady gas stand measurement. Usually, the given efficiency is a mix between aerodynamic effects, mechanical friction and thermal losses inside and outside the turbocharger (9). Models have to be implemented to separate those effects (10);
5. New setups, as closed loop turbine measurement ones (11), are also required to avoid the prediction uncertainties generated by the extrapolation of the efficiency versus blade speed ratio (BSR) curve. Indeed, exhaust pulsations at low engine speed with fixed geometry turbocharger cover a range of BSR larger than the usual ones;
6. When twin scroll turbochargers are used, the flow a better prediction of the performance requires 3 maps (12);
7. As twin scroll turbochargers cannot be always designed cost effectively for small 4 cylinder engines (below 1.6 L engines), effect of separation walls (13) can help to reduce the exhaust pressure during the overlap period. Their modeling with 1D code can be very tricky.

CONCLUSIONS

The presented results of 2 engines show the interest to get turbocharged engines with VVTA systems. At low engine speed, 3 cylinder engines as well as 4 cylinder ones have their turbocharger behavior transformed by the scavenging process and lead to an increase of BMEP for 20 % to 50 %. That's why the most efficient gasoline engines are nowadays synonyms of GDI turbocharged engine with VVTA systems.

However, as shown with MPI engines, this scavenging process has to be limited to avoid excessive fuel consumption during too frequent acceleration maneuvers. On GDI engines, injection is done when exhaust valves are close; only fresh air travels through the scavenging way. A limitation could appear in the coming years due to the particle number that will be regulated. As those particles are highly linked to the in-cylinder air/fuel ratio, it requires improvements in the real-time estimation of trapped air mass flow.

To control and predict this scavenging process with VVTA and turbochargers, we have presented tools. For the analysis of the test bench results, the use of a model that exploits the measurement of instantaneous pressure transducers allows to estimate the flows during this crucial period. However, this method addresses plenty of issues and in particular those linked to the coherence of the intake, exhaust and cylinder pressure transducers and to the discharge coefficient definition when valves are close to the piston or due to the high pressure ratio through valves. In addition, the turbine behavior has to be studied also carefully at low engine speeds: many sources of deviations can trouble the analysis and the results.

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