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# Torque-Oriented Control of an Homogeneous Charge Compression Ignition Vehicle

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*Abstract*— **This paper presents a torque-oriented structure designed for a Diesel homogeneous combustion compression ignition (HCCI) engine control. The proposed controllers are designed to get the maximum benefit provided by the HCCI combustion system. The engine control issues comes from the sensitivity of the HCCI combustion system and from the complexity of the intake system. The proposed control structure combines several controllers presented in previous papers. Each controller is adapted in order to takes into account of the constraints inferred by this specific engine architecture and combustion system. The air path controller copes with a non-linear coupled multivariable system. The fuel path controller is designed to ensure a good coordination with the airpath system dynamic. These two aspects with the corresponding control strategies are presented in this paper. The HCCI engine is embedded into a demonstration car and the torque structure is adapted accordingly. The controller performances are evaluated during New European Driving Cycle (NEDC) at the rolling test bed.**

## I. INTRODUCTION

The compression ignition engine is an efficient internal combustion engine for ground vehicle applications and its application for small and medium passenger cars has recently been increased. However, the new regulations for pollutant emissions reduction and a stronger need for fuel saving have lead to the application of cleaner combustion systems. For the compression ignition engine the application of the homogeneous combustion is one of the best means to reduce the emission levels and to meet future emission standards. The HCCI combustion has been introduced in the late seventies. The first attempts have been tried on two-stroke gasoline engine [1] and four stroke engines few years later [2]. The main difference between conventional Diesel combustion and the homogeneous combustion concerns the in-cylinder mixture. While the Diesel combustion starts in premixed condition and is followed by a diffusive regime, the HCCI combustion is mainly a premixed combustion using high exhaust gas recirculation (EGR) rates. Since the combustion takes places in a homogeneous mixture the soot production is lower. The nitric oxides (NOx) production is linked with flame temperature, thus, due to high EGR rate used in HCCI the flame temperature tends to decrease and the NOx emission accordingly [3]. Very low NOx emission levels can be reached without the need for expensive NOx after-treatment devices. The HCCI combustion combines the advantage of the high efficiency of the compression ignition engine and the very low emissions of the spark ignition engine equipped with a tree way catalyst.

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Practical application of HCCI on car engines faces fundamental combustion control challenges. The HCCI process is very sensitive to disturbances. This sensitivity makes the HCCI engine control requirements higher than the ones of a conventional Diesel engine. The key issue with HCCI engine is to maintain the appropriate thermodynamic conditions (temperature, pressure and composition). The control system must be very accurate for intake manifold condition control and it must be able to adapt the injection settings to compensate the possible airpath tracking errors. The conventional open-loop control system dedicated to the actual production Diesel engine are not suitable with these new constraints. In addition, standard controller require a time-consuming calibration procedure. The model-based control design has proven to be the best method for Diesel engine control [4]. Including the physical model that rules the underlying phenomena in control law synthesis allows reduction in the controller calibration task and a better engine to engine adaptation. For the HCCI engine, this approach is mandatory to get the maximum benefit of this combustion system.

This paper presents the torque-based controller design for an HCCI engine embedded into a demonstration vehicle. The experimental facilities including the engine and the rapid prototyping system are presented in the section II. The next section is divided in two parts and it starts with the air path controller design (section. III-A). In this part, the boost pressure and intake composition control are presented. This section is followed by the fuel path control design which gives an overview of the full structure with a description of the main modules (section. III-B). The tests and validations are performed on the demonstration vehicle during the European driving cycle (section IV). Conclusions are given in section V.

## II. EXPERIMENTAL SETUP

Figure 1 displays the experimental engine architecture and actuators. The engine is a 2.2 liter, four cylinders direct injection Diesel engine. The engine has a dual mode combustion system able to run in conventional Diesel mode at low load and in HCCI mode from low to part loads. A specific concept of combustion chamber with a compression ratio of 14:1 has been developed at IFP to enhance the in-cylinder air-fuel mixing [5]. The fuelpath system includes a Diesel common-rail injection system with solenoid injectors. The airpath

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system contains two different parts : the high pressure EGR system and the air system. The EGR system is controlled by a valve located downstream of the EGR cooler. This latter can be bypassed during cold start operation to warm the engine faster. A variable geometry turbocharger (VGT) is fitted to the air system. The VGT gives a better boost pressure control as conventional turbocharger and is well suited to a intake architecture including a high pressure EGR loop. An intake throttle, located downstream the air cooler, can be used in transient to increase the EGR flow rate by changing the pressure difference between inlet and outlet receivers. The exhaust after-treatment system consists in a Diesel Oxidation Catalyst (DOC) and a Diesel Particulate Filter (DPF). The DOC treats the HC and CO emissions and the DPF allows a reduction of particulate matter. The engine has been embedded in a VelSatis Renault vehicle.

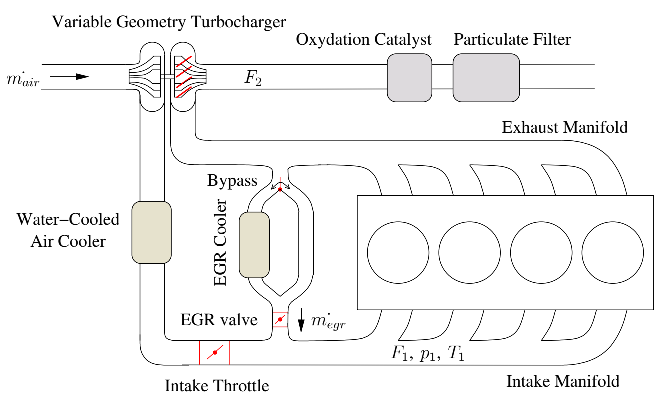


Fig. 1. Dual mode Diesel-HCCI engine architecture.

The torque control structure is developed with code generation tool chain provided by MathWorks (Simulink, Real Time Workshop, xPC Target). The rapid prototyping system if developed by IFP and is composed of : an embedded controller connected to a Target computer.

## III. Torque-Oriented Control Structure

The general purpose of the engine management strategies is to control the engine in order to deliver the torque requested by the driver as fast as possible while maintaining low pollutant emissions. In the case of an HCCI engine, the control issues are more complex than the ones of a conventional compression ignition engine. This is mainly due to the sensitivity of the combustion system according to the in-cylinder conditions. Thus, the air path and fuel path controllers must accurately feed the cylinders with the good amounts of air, fuel and burnt gases. This section explains the air path and fuel path related issues and the controller's principles.

### A. Air Path Control

The composition and the amount of gas aspirated by the cylinder are the key variables for the HCCI combustion. These variables are controlled with the EGR valve and the VGT. The intake throttle is considered as an additional actuator and it is only used to decrease the air flow in some specific cases. The air intake system can be considered as a multivariable system with two actuators (EGR valve and VGT) and two inputs (fresh air and EGR mass flows). From the air path control point of view, two major issues have to be considered :

- *A strongly coupled dynamic :* The EGR valve acts as a discharge for the turbocharger. The energy taken by the turbine on the exhaust gas depends on the mass flow, which is directly linked with the EGR flow. The EGR flow depends on the pressure difference between the exhaust and intake manifolds, which in turn depends on the turbocharger operating conditions.
- *The estimation of the intake manifold gas composition :* For cost and reliability reasons, the commercial cars are usually not equipped with intake manifold composition probe.

Several control approaches have been proposed to deal with the coupled dynamics issue but most of the proposed strategies rely on simple solutions such as the deactivation of the turbocharger control when high EGR rate are needed or linear controllers with gain scheduling. These solutions are efficient only with a fine tuning of the controller gains which requires a large amount of engine tests. Multivariable control techniques have also been investigated to control the same intake system. Most of the published approaches propose to control the intake and exhaust manifolds pressures [6], [7]. Other studies deal with the control of the air flow and the boost pressure [8].

The Diesel engine intake system control structure needs a large calibration effort as far as the intake system complexity increases. In this paper, we addressed this issue by taking into account of the main physical models of the system. The main idea is to limit the number of tuning parameters. The air path architecture is depicted on figure 2. The first layer corresponds to the engine mapping resulting from the engine calibration procedure done at the steady test bench. These values are mapped according to the IMEP setpoint and the engine speed $N$. These setpoint cannot be applied directly to the input of the control system, thus a filtering stage compute two feasible trajectories. The filter time constants are carefully tuned to respect the system dynamics on one side and the actuators dynamics on the other side. Then, two separated systems are defined :

- *The flow control related to the fast actuators.* In our case, we apply a motion planning strategy to where the classical feedback variables for combustion control, the intake pressure and the burnt gas rate (BGR) are transformed into air flow and EGR flow trajectories. The air and EGR flow setpoints are computed from the motion planning strategy which corresponds to an explicit inversion of the intake manifold model. It is assumed that the EGR valve control is fast enough to guarantee the EGR flow tracking. The intake throttle is also a fast actuator but it can only act as a restriction for the air flow. This assumption is accepted since the two actuators are close to the intake manifold. The inversion of the intake manifold provides realistic trajectories.

- *The pressure control performed by the VGT.* The air flow trajectory given by the motion planning cannot be followed directly by the VGT. This latter has a slower settling time due to its inertia. Moreover, intake pipes volume, air cooler volume and turbocharger dynamics are not included on the motion planning of the air flow. Here the control use a model-based approach including the first principle and turbocharger maps to generate a physical feedforward control.

The IFP intake system is treated as two separated control loops : the gas flow trajectories are tracked with the fast actuators (EGR valve and intake throttle). The pressure control is achieve by the turbocharger which has a slower dynamics. These two systems are cascaded since the intake pressure is an input for the flow trajectories control.

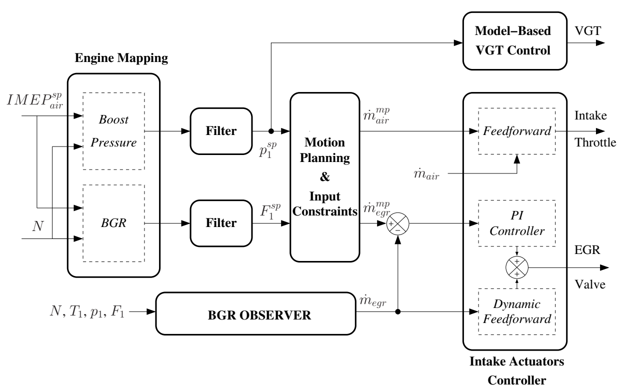


Fig. 2. Simplified control scheme for the airpath control.

*1) Motion Planning Strategy for EGR Valve and Intake Throttle Control :* The reference dynamic of the intake manifold is obtained from the total mass balance, ideal gas law and the composition balance with the standard assumptions for a mean value engine modelling level. The intake manifold temperature is not modified by the EGR gas temperature since the EGR cooler is sized to maintain the manifold temperature lower than 50 degrees. As a result, if we neglect the temperature variation in the intake manifold, the system writes :

$$\begin{cases} \dot{p}_1 &= \dfrac{RT_1}{V_1}\left(\dot{m}_{air} + \dot{m}_{egr} - \dot{m}_{in}\right) \\ \dot{F}_1 &= \dfrac{RT_1}{p_1 V_1}\left(\dot{m}_{egr}\left(F_2 - F_1\right) - \dot{m}_{air}F_1\right) \end{cases} \tag{1}$$

where $p_1$ and $F_1$ are the intake pressure and BGR respectively. The BGR corresponds to the fraction of burnt gas in the intake manifold, it is defined as :

$$F_1 \triangleq 1 - \frac{m_{1,air}}{m_1} \tag{2}$$

$m_1$ is the total mass of gas in the intake manifold and $m_{1,air}$ represents the mass of air into the intake manifold. In equation (1), $F_2$ represents the exhaust equivalence ratio and the aspirated gas flow $m_{in}$ is given by :

$$\dot{m}_{in} = \frac{p_1}{RT_1}\eta_{vol}(p_1, N)\frac{NV_{cyl}}{120} \tag{3}$$

If we note $\alpha = (RT_1)/V_1$ and $\beta = (1/RT_1)V_{cyl}(N/120)$, the system of equations 1 can be rewritten as :

$$\begin{cases} \dot{p}_1 = \alpha\left(\dot{m}_{air} + \dot{m}_{egr} - \beta\eta_{vol}(N, p_1)p_1\right) \\ \dot{F}_1 = \dfrac{\alpha}{p_1}\left(-(\dot{m}_{air} + \dot{m}_{egr})F_1 + F_2\dot{m}_{egr}\right) \end{cases} \tag{4}$$

The computation of the motion planning is an implicit inversion of the system of equations (8) with a filtering of the pressure and BGR setpoints according to the feasible trajectories. This inversion and the tuning of the constraints are demonstrated in [9]. The feasible trajectories depend on the actuators settling time and on the air and EGR systems settling time. As a result, the desired air and EGR flow trajectories are derived from the measurements, the state variables and their first-derivatives :

$$\begin{cases} \dot{m}_{air}^{mp} = f_1(T_1, N, F_2, p_1, \dot{p_1}, F_1, \dot{F_1}) \\ \dot{m}_{egr}^{mp} = f_2(T_1, N, F_2, p_1, \dot{p_1}, F_1, \dot{F_1}) \end{cases} \tag{5}$$

In this expression, some variables are measured : the intake manifold pressure ($p_1$), the intake manifold temperature ($T_1$), the exhaust equivalence ratio ($F_2$) and the engine speed ($N$). The intake manifold burnt gas rate ($F_1$) is estimated using an observer. The last part of the controller is to achieve these two separated trajectories. The EGR flow trajectory, $\dot{m}_{egr}^{mp}$, is tracked almost instantaneously with the EGR valve. The normalized EGR valve position, $v_{egr}$, is computed with a feedforward term combined with an integral term to compensate the model errors :

$$v_{egr} = \frac{\dot{m}_{egr}^{mp}}{\widehat{\theta}_{egr}} - k_{i,egr}\int\left(\dot{m}_{egr}^{mp} - \dot{\widehat{m}}_{egr}\right)dt \tag{6}$$

The normalized EGR flow estimation, $\widehat{\theta}_{egr}$, is explained in the next section. The estimated EGR flow is defined as : $\dot{\widehat{m}}_{egr} \triangleq v_{egr}\widehat{\theta}_{egr}$. The intake throttle position is also based on a feedforward control law such that the normalized intake throttle position, $v_{th}$, writes :

$$v_{th} = \frac{\dot{m}_{air}^{mp}}{\dot{m}_{air}} \tag{7}$$

The air flow ($\dot{m}_{air}$) is a measured variable. The intake throttle acts as a restriction and it can only be used to decrease the air flow in case of overshoot. In order to avoid complete closing, the minimum intake throttle position is saturated.

*2) EGR Flow Estimation :* The EGR mass flow depends on the upstream temperature and the pressure drop over the valve. Here, we do not use any temperature and pressure measurement. The EGR flow is estimated knowing the air flow (air flow sensor), the aspirated gas flow (computed from the equation (3)) and with the intake pressure as the main measurement to build a closed-loop observer. The proposed observer uses the dynamical equations derived from the reference system (1) augmented by a third state for the normalized EGR flow $\theta_{egr}$. Let $x = [p_1 \; F_1 \; \theta_{egr}]$ be the observer state and $y = p_1$ be the measurement. Thus, the proposed observer is as follows :

$$\begin{cases} \dot{\widehat{p}}_1 = \alpha \left( \dot{m}_{air} + v_{egr}\dot{\widehat{\theta}}_{egr} - \beta\eta_{vol}(N,p_1)\widehat{p}_1 \right) \\ \qquad -\alpha \left( l_1 - \eta_{vol}(N,p_1) \right) \; \beta \; (\widehat{p}_1 - p_1) \\ \dot{\widehat{F}}_1 = \dfrac{\alpha}{p_1} \left( -(\dot{m}_{air} + v_{egr}\widehat{\theta}_{egr})\widehat{F}_1 + F_2 v_{egr}\widehat{\theta}_{egr} \right) \\ \dot{\widehat{\theta}}_{egr} = -l_3 \; \alpha \; v_{egr} \; (\widehat{p}_1 - p_1) \end{cases} \tag{8}$$

The constant values of the tracking terms, $(l_1, l_3)$, are computed in order to ensure the observer convergence. This was previously proved in [10].

*3) Turbocharger Control :* The VGT position, $v_{vgt}$, varies the opening guide vanes and modifies the exhaust gas flow on the turbine. The modification of the the VGT position, has the effect to directly control the turbocharger speed and the air flow. The following map gives the relation between the turbocharger speed $n_{tc}$, the air flow and the pressure ratio across the compressor $\pi_c$ :

$$\pi_c = h_1 \left( \dot{m}_{air} \frac{\sqrt{T_{uc}} p_{ref}}{\sqrt{T_{ref}} p_{uc}}, n_{tc} \frac{\sqrt{T_{ref}}}{\sqrt{p_{ref}}} \right) \tag{9}$$

The compressor pressure ratio is given by steady-state maps which depends on the corrected compressor speed and mass flow. This map characterizes a given compressor, it is provided by the manufacturer. The control scheme proposed (Fig. 3) consists in controlling the turbocharger speed in order to reach the desired compressor pressure ratio between the ambient pressure and the intake manifold pressure $p_1$ taking into account for the pressure loss across the air cooler and the measured air flow. The choice of this variable as a control variable proves to be interesting and it can be noticed that the map of equation (9) is invertible and that the turbocharger speed can be written as a function which depends on compressor pressure ratio and mass air flow. The intake pressure setpoint is transformed into a compressor pressure ratio setpoint which gives the turbocharger speed setpoint using inverted compressor map. The objective is to control the turbocharger speed in order to follow its setpoint $n_{tc}^{sp}$ as closely as possible. The pressure ratio setpoint needed to reach the desired turbocharger speed has been demonstrated in [11]. The control law depends on the turbocharger dynamics with additive proportional, derivative and integral terms :

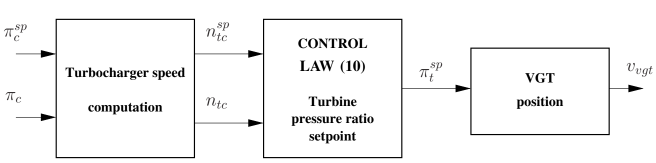


Fig. 3. Model based intake manifold pressure control.

$$1 - \left( \frac{1}{\pi_t^{sp}} \right)^{\frac{1-\gamma}{\gamma}} = \frac{1}{\alpha_{tc}} \Big( \frac{d(n_{tc}^{sp})^2}{dt} - k_p(n_{tc}^2 - n_{tc}^{sp2}) \\ + \beta_{tc} + k_i \int (n_{tc}^2 - n_{tc}^{sp2}) dt \Big) \tag{10}$$

The parameters $\alpha_{tc}$ and $\beta_{tc}$ can be calculated using measured variables and turbocharger maps. Note that, the main control action is contained into the feedforward term $\beta_{tc}$ which depends on the turbocharger physical model. Here, the controller gains $k_i$ and $k_p$ does not need to be calibrated according to the turbocharger operating conditions. Finally, the VGT position is computed from the desired turbine pressure ratio. For this purpose, the equation describing the flow through the turbine has to be inverted :

$$\dot{m}_{air} \frac{\sqrt{T_{ut}} p_{ref}}{\sqrt{T_{ref}} p_{ut}} = h_2 \left( \pi_t, n_{tc} \frac{\sqrt{T_{ref}}}{\sqrt{p_{ut}}}, v_{vgt} \right) \tag{11}$$

### B. Fuel Path Control

The main fuelpath control problem comes from the adaptation of the air system and fuel system dynamics : the fuel injection dynamics (fuel mass flow and injection timing) is faster than the settling time of the intake system feedback variables ($p_1$ and $F_1$). The engine fuel path controller must takes into account of these two separated dynamics avoiding related perturbations. This issue has been addressed for the conventional Diesel engine but is amplified with the HCCI engine due to higher BGR levels applied and due to the greater BGR variations. The proposed control system adapts the injection timing and fuel mass according to : the air flow errors, the BGR errors and the temperature drifts occurring during transient engine operations.

The fuel path control architecture is depicted on figure 4. It can be divided into four separated modules. The engine mapping provides the fuel mass, fuel pressure and start of injection according to the engine operating point (defined by the load and the engine speed). This mapping has been established in stationary conditions, thus, transient corrections are needed :

- The raw load setpoint, $IMEP_{air}^{sp}$, is limited to prevent smoke emissions during large transients. The computation of the corrected load setpoint, $IMEP_{fuel}^{sp}$, is performed by the smoke limiter that corresponds to an air-fuel ratio limiter.
- Another important stage of correction is the one applied to the start of injection (SoI). Without any dynamic adaptation of the SoI raw mapping, a BGR overshoot leads to misfires and a lack of BGR increases the combustion noise and the NOx emissions. More generally,

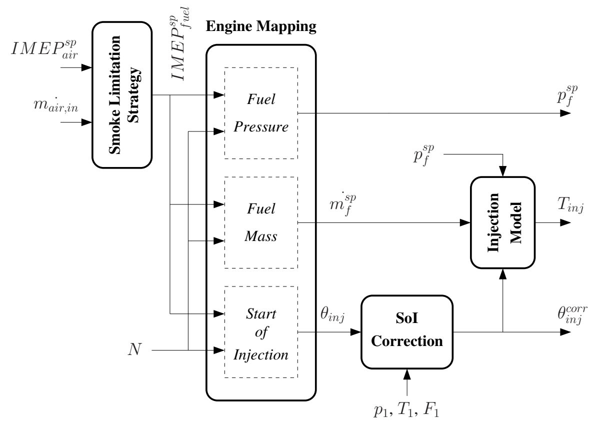


Fig. 4. Simplified control scheme for the fuelpath control.

the combustion is not stabilized during transients. Here, The SoI is used as an actuator to control the start of combustion.

Finally, the injection model computes the injection duration according to the fuel mass setpoint and the measured fuel pressure. The fuel pressure is controlled by two valves acting as a charge and discharge of the fuel into the common rail system. This controller is not presented in the paper.

*1) Smoke Limitation Strategy :* The production of particulate matters in Diesel engines is strongly linked with the equivalence ratio of the air-fuel mixture, $\phi$, trapped into the cylinder [3]. The particulate matter are produced for rich mixture when the equivalence ratio is close or up to its stoichiometric value $\Phi_{st}$. The smoke emission is caused by an increase of the equivalence ratio during transient operating conditions due to the the air settling time. The principle of the smoke limitation strategy is based on the limitation of the load setpoint in transient using the observed value of the aspirated air mass $\dot{m}_{air,in}$ as a main input. Without any corrections, the injected fuel mass is given by the injection mapping as a function of the load setpoint and engine speed $N$ :

$$\dot{m}_f = f(IMEP_{air}^{sp}, N) \tag{12}$$

Knowing the air mass available in transient, the maximum allowable fuel is computed from :

$$\dot{\hat{m}}_f = \dot{m}_{air,in}\ \phi_{max}\ \Phi_{st} \tag{13}$$

where $\phi_{max}$ is the maximum equivalence ratio determined during the engine calibration procedure. The maximum equivalence ratio can be stored in a table a a function of the engine speed. The aspirated air mass is given by : $\dot{m}_{air,in} = (1 - F_1)\dot{m}_{in}$. Knowing the maximum equivalence ratio, the corrected load setpoint, $IMEP_{fuel}^{sp}$, is computed by inversion of the injection mapping :

$$IMEP_{fuel}^{sp} = f^{-1}(\dot{\hat{m}}_f, N) \tag{14}$$

The smoke limitation strategy is a feed-forward correction of the raw IMEP setpoint. But this compensation can be viewed as a filter with varying time constant close to the air system time constant. The load setpoint is limited only during transient condition. The only parameter to calibrate is the maximum equivalence ratio. Its value is chosen in order to avoid too rich mixture taking into account of the driveability of the vehicle (torque production settling time).

*2) Start of Injection Control :* The start of injection control helps to maintain a stable HCCI combustion using a feedforward control law that compensates the cylinder initial conditions offsets (pressure, temperature and composition in the intake manifold before the intake valve closing). The goal of this correction is to guarantee that the start of combustion (SoC) will occurs at the desired setpoint. The closed loop control of the measured SoC using cylinder pressure sensors is a possible way to deal the combustion control. This solution is efficient but expansive due to the cost of the sensor. Moreover, the sensor drift can be a limiting factor. Look-up table corrections have been tested but they require a large amount of experimental tests [12]. The IFP approach is a feedforward control law that does not require any cylinder measurement and with a limited calibration effort. In homogeneous combustion, there is no direct trigger to start the combustion. The ignition is only driven by kinetic reaction after the injection. A classical model for the auto-ignition delay [13] has been adapted to be suitable for the HCCI combustion [14]. It writes :

$$\int_{\theta_{SoI}}^{\theta_{SoC}} \frac{k_1}{k_2 + F_1} p_{cyl}(\theta)^n \exp\left(-\frac{T_A}{T_{cyl}(\theta)}\right)\ d\theta = 1 \tag{15}$$

where $k_1$, $k_2$, $n$ and $T_A$ are the model parameters. The cylinder pressure $p_{cyl}$ and temperature $T_{cyl}$ evolution (according to the crank angle $\theta$) prior to the auto-ignition are computed from the intake valve closing conditions assuming an isentropic compression. The in-cylinder conditions are assumed to be equal to the intake condition at the intake valve closing time. During transient, a corrective offset $\delta\theta_{SoI}$ on the reference injection timing $\theta_{SoI}$ compensates any error on intake conditions.

$$\int_{\theta_{SoI}+\delta\theta_{SoI}}^{\theta_{SoC}} g\left(p_1 + \delta p_1, T_1 + \delta T_1, F_1 + \delta F_1, \theta\right)\ d\theta = 1 \tag{16}$$

The explicit solution to find the appropriate SoI correction is not an easy task when considering models with an integral form 16. A simple way to proceed is to approximate the solution of the Arrhenius integral function. The detailed computation is given in [15]. This general solution gives an analytical function of the SoI correction depending on the air system errors, the model parameters and the raw SoI :

$$\delta\theta_{SoI} = h(\delta p_1, \delta T_1, \delta F_1, k_1, k_2, n, T_A, \theta_{SoI}) \tag{17}$$

The main advantage of proposed compensation method is that it does not require any cylinder pressure sensors and the auto-ignition model parameters are estimated with a limited amount of experimental data. Detailed engine results can be found in [16].

*3) Injection model :* The injected fuel mass depends on the integral of the mass flow rate through the injector holes during the injector opening duration. In conventional engines, injection duration is stored in look-up tables as a function of the fuel mass and fuel pressure. This conventional approach is not very practical, since the look-up tables are limited to two or three inputs while the parameters that influence the injected masses are numerous. Instead of using injector maps with tabulated correction terms, the injection model proposed is based on a control-oriented physical model of the process. This model provides an explicit formulation of the injection duration $T_{inj}$ and the injection settings : fuel mass, fuel pressure, SoI and dwell time between two subsequent injections (for multiple injection pattern). This model and the main results are reported in [17].

## IV. Experimental results

The final experimental validations and controller tuning are done on the vehicle. The vehicle tests are performed on roller test bed during European driving cycle. A part of the vehicle results are displayed on figure 5. The top figure represents the vehicle speed profile recorded during the NEDC cycle with the corresponding pedal position and gear number. The figure 6 displays the gas flows (air and EGR), the BGR, the turbocharger speed and the actuators positions. It can be noticed that the EGR flow control is almost perfect. The BGR control is a key variable to drive the HCCI combustion and for NOx reduction. Therefore, the engine operates mostly in HCCI combustion mode during the driving cycle. Here, The BGR is well tracked in spite of small overshoots. The BRG overshoots are due the air flows errors in transient. However, the robust calibration in addition with a dynamic adaptation of the start of injection avoids misfire occurrences. The turbocharger speed control is good in spite of transient errors during tip-in (t=247s). This is mainly due to the turbocharger inertia. In summary, the results on the European cycle are good in terms of air path transient. The driver torque demand is reached, even with HCCI combustion mode to Diesel combustion mode transitions with hight amplitude of BGR variations.

## V. Conclusion

Conventional open-loop controller based on tabulated feedforward and look-up table for linear controller gains are not sufficient for HCCI engine control. Moreover, the calibration time reduction is now a key factor in automotive industry. This paper has presented a full torque control structure developed by taking into account of the system nonlinearities and based on physical models. This torque control structure is relevant for an HCCI engine but it is also suitable for conventional Diesel engines applying lighter HCCI concept and for engines with complex airpath architecture. The controllers have the advantage to be easy to calibrate by the application of model-based controllers.

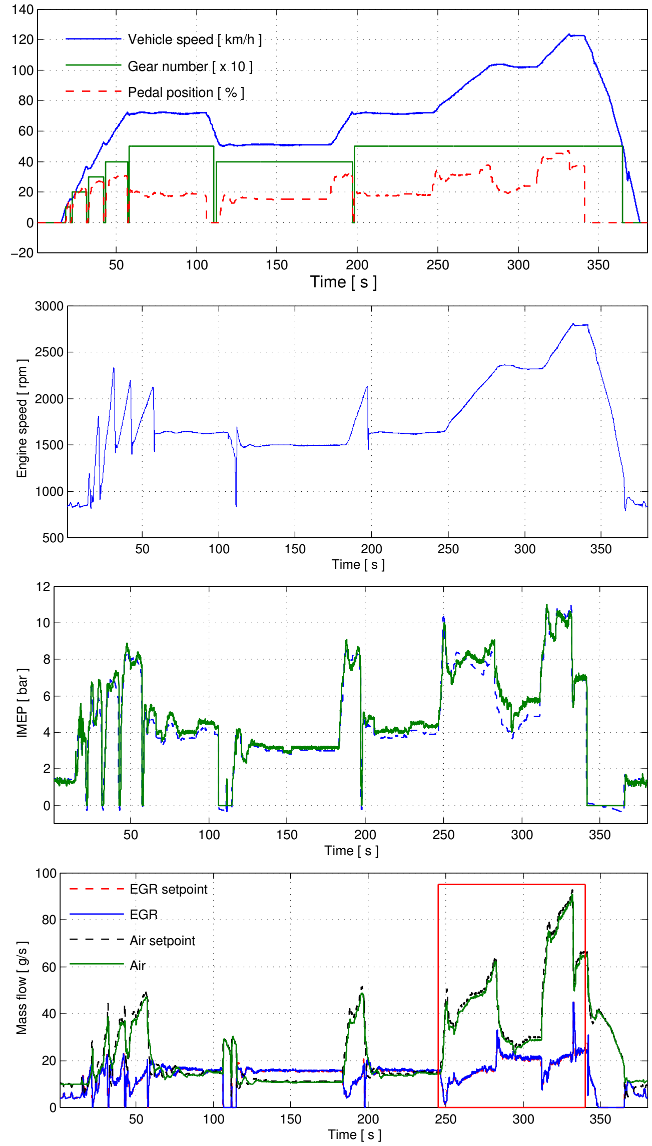


Fig. 5. New European Driving Cycle vehicle results. From top to bottom : vehicle speed, engine speed, indicated mean effective pressure and gas flows.

The airpath control structure is designed such that a limited number of sensors are required. The sensor setup is the same than the one of commercial Diesel engines and the intake manifold gas composition is estimated. The controller calibration effort is also limited since the control laws mostly rely on dynamic feedforward terms computed from the physical equations. The fuelpath control structure is robust to the airpath system errors occurring in real driving conditions. This is achieved without the need for cylinder pressure-based closed loop combustion control system. As a result, the HCCI combustion engine is embedded into a vehicle. The control structure is tested over European driving cycles. The experimental results show good performances in term of air path trajectory tracking and also in term of fuelpath control adaptation according to airpath transient errors. This proves the relevance of the control techniques applied.

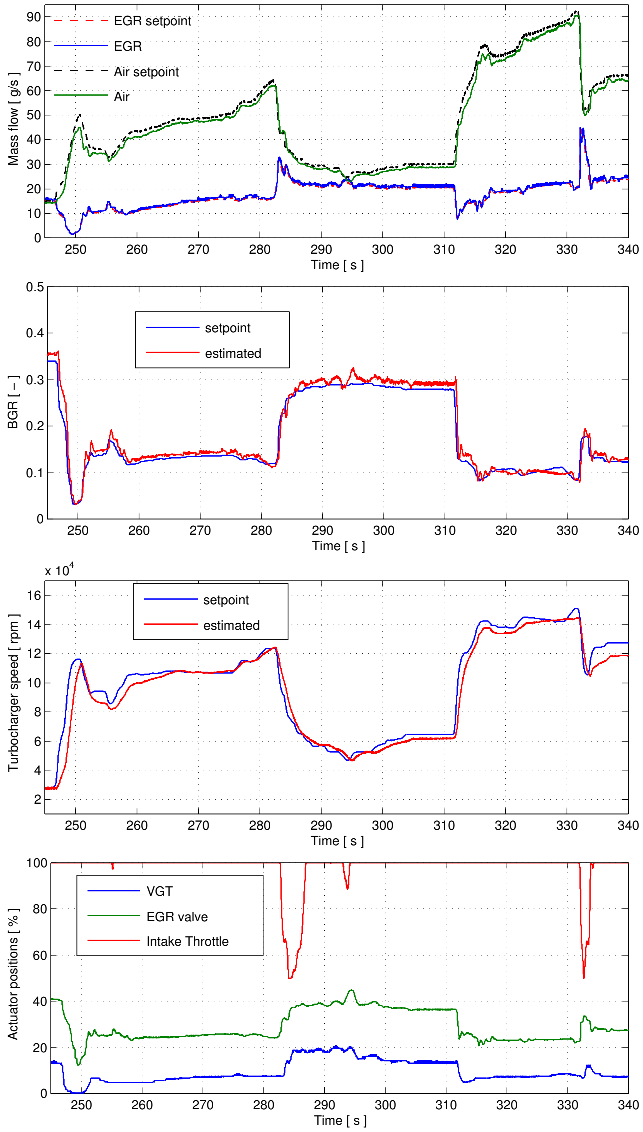


Fig. 6. New European Driving Cycle vehicle result. Zoom on the transient surrounded on the figure 5 (from $t = 245$ $s$ to $t = 340$ $s$). From top to bottom : air-EGR flows, BGR, turbocharger speed and actuators positions.

## VI. ACKNOWLEDGMENTS

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## VII. NOMENCLATURE

| | | |
|---|---|---|
| $F_1$ | Intake manifold burnt gas fraction | - |
| $F_2$ | Exhaust equivalence ratio | - |
| $IMEP$ | Indicated mean effective pressure | bar |
| $\dot{m}_{air}$ | Air mass flow | $kg.s^{-1}$ |
| $\dot{m}_{egr}$ | EGR mass flow | $kg.s^{-1}$ |
| $\dot{m}_f$ | Fuel mass flow | mg/stroke |
| $\dot{m}_{in}$ | Aspirated mass flow | $kg.s^{-1}$ |
| $N$ | Engine speed | $kg.s^{-1}$ |
| $p_1$ | Intake manifold pressure | Pa |
| $p_f$ | Fuel pressure | Pa |
| $R$ | Ideal gas constant | $J.K^{-1}.kg^{-1}$ |
| $T_1$ | Intake manifold temperature | K |
| $V_1$ | Intake manifold volume | l |
| $v_{egr}$ | EGR valve position | - |
| $v_{th}$ | Intake throttle position | - |
| $v_{vgt}$ | VGT position | - |

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