Combustion Control of an HCCI Diesel Engine with Cool Flame Phenomenon

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Abstract— **In this paper, we propose a control strategy to improve stability of the combustion of HCCI engines during sharp transients. This approach complements existing airpath and fuelpath controllers, and aims at accurately controlling the end of the cool flame phenomenon. For this purpose, injection time is adjusted based on an auto ignition model and a cool flame model. Experimental results are presented, which stress the relevance of the approach.**

I. INTRODUCTION

Over the last 20 years, pollution standards for automotive engines have become more and more stringent. Engine pollutant emission reduction has then become a topic of major interest for engine development, and, more generally, for the car industry. Besides after-treatment technologies (which are pricey for Diesel engines), a particular emphasis has been put in developping cleaner combustion modes such as the Highly Premixed Combustion modes (HPC), including Homogeneous Charge Compression Ignition (HCCI) (see [1]).

Consider a Diesel engine with an exhaust gas recirculation (EGR). This engine can be used in various combustion modes ranging from conventional to homogeneous. In this paper, we focus on the HCCI mode which is the most challenging from a control perspective (see [2]). HCCI combustion requires the use of high EGR rates. The air charge admitted in the cylinder is significantly diluted, which reduces nitrogen oxides (NO_x) emissions by lowering the temperature peak during combustion. This combustion takes place according to the timeline detailed in Figure 1. There are two main phases corresponding to the airpath subsystem (which involves the intake and exhaust pipes, the intake and exhaust manifolds, the turbocharger(s), and the valve train) and the fuelpath subsystem (which consists of the injectors). This combustion mode is highly sensitive. Accurate airpath and fuelpath controllers are thus required to manage the HCCI combustion. Airpath controllers have long been proposed (see [3], and its references). They result in efficient tracking of the intake manifold variables (reference total mass, composition, and temperature of the intake charge) even during transients. The main actuators employed are the EGR actuators (external circuit valve or variable valve train), the intake and exhaust throttles and the turbocharger(s).

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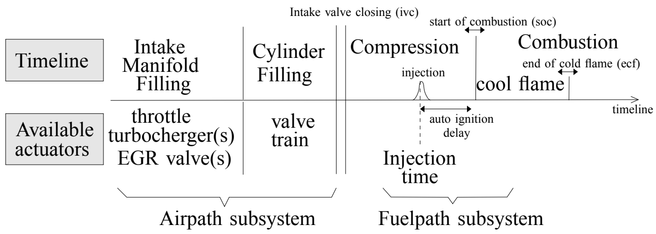

Fig. 1. Timeline of Low Temperature Diesel engine cycle with direct injection

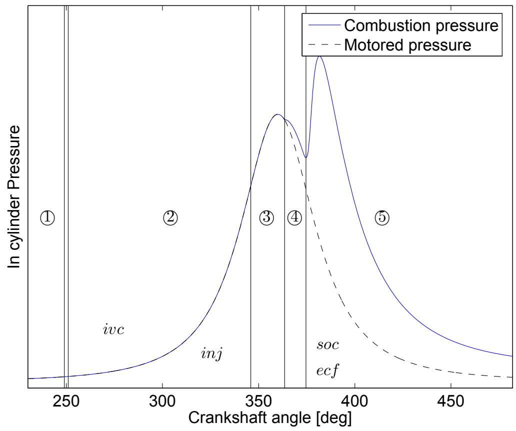

Fig. 2. In-cylinder pressure during one cycle: ① filling the cylinder, ②: compression, ③: auto ignition, ④: Cool flame, ⑤: Combustion and expansion, ivc: intake valve closing, inj: injection, soc: start of combustion, ecf: End of Cool Flame. (360 corresponds to the Top Dead Center)

Classic fuelpath controllers can be described as follows. During the cylinder compression phase, fuel is injected and mixed to the compressed air and burned gas mixture. The fuel vaporizes and, eventually, auto-ignites after the so-called ignition delay. At first, a cool flame takes place (see [4], [5], [6]), then a standard combustion comes into play (see Figures 1 and 2). Standard fuelpath control strategies focus on controlling each cylinder fuel mass. This is usually sufficient to track the setpoints at steady state or at quasi-static transients.

As we will now discuss it, these controllers are not sufficient to stabilize the HCCI combustion mode during transients. In fact, and by contrast to conventional Diesel combustion mode, slight offsets of cylinder initial conditions (e.g. pressure, temperature, or composition) makes the combustion phasing shift in time because they are not taken into account in the injection time controller. Details are given in Figure 3 where the impact of intake manifold conditions variations (pressure and composition) on combustion phasings are reported (θ_{soc} is the start of combustion, CA_X is the crankshaft angle at which X percent of the fuel has been

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burnt). These offsets can be caused by natural airpath regulation errors or malfunctions such as valve clogging/ageing.

In turn, these combustion phasing errors have impacts on the torque produced and the pollutant generation. In details, Figure 4 presents a combustion phasing variation obtained on a four cylinders HCCI Diesel engine, and its effects on pollutant, noise and torque production. The only parameter varying during the presented experiment is the combustion phasing. The BGR, intake manifold pressure, intake manifold temperature, and injected fuel mass are kept constant and correspond to an optimal trade-off. These figures clearly show that the combustion phasing has a major impact on pollutant, noise, and torque production. Our focus is on developing a method to coordinate the airpath and fuelpath subsystem in order to minimize combustion phasing shifts.

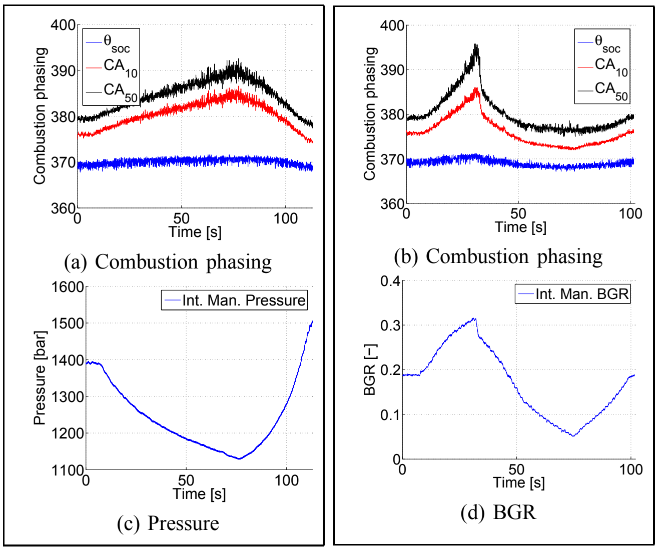

Fig. 3. Experimental results obtained on a Diesel Engine at 2330rpm: Left: influence of the Intake manifold Pressure on combustion phasings Right: influence of the Intake manifold BGR on combustion phasings

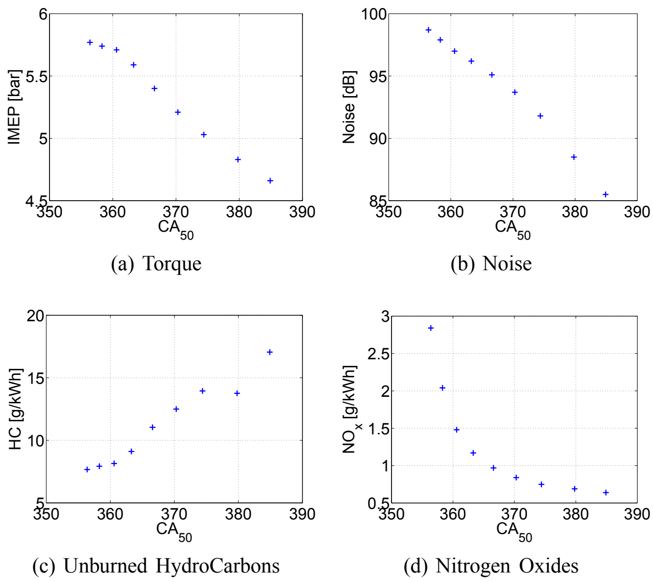

Fig. 4. Experimental results obtained on a four cylinder Diesel Engine: influence of the combustion phasing on engine pollutant, noise and torque production.

To address the discussed issues, i.e. to circumvent changes in the cylinder initial conditions, we propose to use the *injection time* as an actuator to control the combustion phasing. To take into account the most important phenomena in terms of combustion phasing, we propose to control the end of the cool flame (*ecf*). For that purpose, we develop a simple controllers coordination method. Along with suportive experimental results. This is the main contribution of this paper. *A noticeable point of our approach is that the control variable we use (injection time) can be used on all commercial line engines without requiring any hardware upgrade.*

In details, we consider three parameters as cylinder initial conditions: these are the pressure, the temperature and the composition. We assume that the auto ignition can be modeled by a Knock Integral Model (KIM) (see [7], [8] or [9]), and we propose a cool flame duration model. In practical implementation, the initial conditions are either inferred from direct measurements or estimated by an external observer. To guarantee that the *ecf* occurs at a desired setpoint, the injection time is updated according to a first order development of the auto ignition and cool flame models. This approach complements the one presented in [10], [11] in which the cool flame phenomenon was not accounted for.

The paper is organized as follows. In Section II, we detail existing combustion control technologies that we wish to complement and present our approach. In Section III, we present the models we base our study on, along with the main physical assumptions underlying our work. In Section IV, we formulate the control problem, and propose a solution at first order in Section V. Experimental results obtained on a 2.2 liter four cylinder direct injection engine are reported and discussed in Section VI. Conclusions and future direction are given in Section VII.

TABLE I
NOMENCLATURE

Symb.	Quantity	Unit
θ	Crankshaft angle	-
V	Cylinder volume	m^3
$P(\theta)$	Cylinder pressure	Pa
$T(\theta)$	Cylinder temperature	K
X	In-cylinder burned gas rate (BGR)	-
V_{ivc}	In-cylinder volume at ivc	m^3
P_{ivc}	Cylinder pressure at ivc	Pa
T_{ivc}	Cylinder temperature at ivc	K
ϕ	Air/fuel ratio	-
θ_{soi}	Injection crankshaft angle	-
θ_{soc}	Start of combustion crankshaft angle	-
θ_{ecf}	End of cool flame crankshaft angle	-
m_{inj}	Injected fuel mass	mg
γ	Ratio of specific heat	-
T_q	Torque	Nm
N_e	Engine speed	rpm

II. CURRENT COMBUSTION CONTROLLERS AND PROPOSED IMPROVEMENT

A complete nomenclature of engine variables is given in Table I. In generally observed engine setups, airpath and fuelpath controllers are used to guarantee that engine variables (pressures, temperatures, and injected fuel mass among others) track reference values. These controllers are used in the context of actual vehicle implementation which implies frequent transients due to varying driver torque demands ($\overline{T}_q$) and engine speed (N_e). In turn, these demands result in frequent transients for reference airpath and fuelpath variables. Although both controllers work simultaneously, combustion stability is jeopardized during transients due to the differences of the airpath and fuelpath closed-loop performances. Insight into this phenomenon are discussed in [10].

Controlling the combustion is the key to HCCI engine control. To this end, all the controllers presented in the literature (e.g. using Variable Compression Ratio [12], dual fuel [13], or variable valve train [14], [15] or [16]) use high frequency in-cylinder sensors and/or additional actuators. These solutions are costly. Rather, to control the combustion phasing, we propose a solution requiring only devices available on all commercial-line engines, *i.e. requiring no cylinder pressure sensors*.

Consider the fuelpath subsystem whose dynamic behavior is inconsistent with the dynamic behavior of the airpath subsystem (see [10]). This subsystem is controlled by the fuel injection. The injected mass is used to produce a reference torque, and it cannot be changed without harming performance. Therefore, only one degree of freedom of the fuelpath remains as possible additional control variable. It is the injection time (start of injection) which appears in the combustion timeline in Figure 1.

Figure 3 shows the evolution of combustion phasing while varying intake manifold pressure and BGR. It appears that the auto ignition delay and cool flame duration are affected differently:

1) The start of combustion is slightly delayed by a later injection time. The auto ignition delay is thus affected by thermodynamic conditions.
2) The duration between the start of combustion θ_{soc} and CA_{20} has a large sensitivity.
3) The duration between the CA_{20} and the CA_{50} is hardly affected.

The previous remarks lead to consider that the cool flame and the auto ignition are the most sensitive phenomena, and that controlling the end of the cool flame (*ecf*) is sufficient to provide an effective control of the CA_{50}.

Thus, in our approach, we propose to cascade the control of the *ecf* onto the injection time θ_{soi} variable. This implies that, instead of constant values corresponding to references, θ_{soi} has non-constant values during transients. Two particular phenomena have to be accounted for in the control law: the auto ignition and the cool flame. We now model these.

III. Modeling

A. Auto ignition modeling

Auto ignition duration of fuel/air/burned-gases mixture is usually modeled with a Knock Integral Model (KIM) (see [7], [8] or [9]). This model gives an implicit relation between θ_{soi}, θ_{soc} and the physical in-cylinder parameters such as $P(\theta)$, $T(\theta)$, X (additionally an air/fuel ratio ϕ can be considered), usually found under the implicit integral form

$$\int_{\theta_{soi}}^{\theta_{soc}} \mathcal{A}^{ai}(p(\theta)) \frac{d\theta}{N_e} = 1 \qquad (1)$$

where $\mathcal{A}^{ai}$ is an Arrhenius function, and $p(\theta) = (P(\theta), T(\theta), X)^T$. Our control strategy can be used with any (smooth) function $\mathcal{A}^{ai}$. A prime example is (see [17])

$$\mathcal{A}^{ai}(p(\theta)) = \frac{A_1}{1 + C_1 X} P(\theta)^{n_1} \exp\left(-\frac{T_1}{T(\theta)}\right) \qquad (2)$$

where A_1, C_1, n_1, and T_1 are known constant positive parameters. This formula is used throughout the paper and experiments. Numerous other possible choices could be considered as well, using the same approach.

B. Cool flame modeling

The cool flame is a combustion with a very low reaction rate. Several chemical processes occur simultaneously and lead to the real combustion. By analogy with the auto ignition modeling, the cool flame duration can be represented with a KIM. The main in-cylinder parameters impacting on the cool flame phenomenon are the pressure, temperature, composition (BGR), and the available fuel mass (which is the injected fuel mass since no injection is performed during the cool flame). The proposed model is

$$\int_{\theta_{soc}}^{\theta_{ecf}} \mathcal{A}^{cf}(p(\theta), m_{inj}) \frac{d\theta}{N_e} = 1 \qquad (3)$$

where $p(\theta) = (P(\theta), T(\theta), X)^T$ and $\mathcal{A}^{cf}$ is an Arrhenius function

$$\mathcal{A}^{cf}(p(\theta), m_{inj}) = \frac{A_2\ (m_{inj})^{x_2}}{1 + C_2 X} P(\theta)^{n_2} \exp\left(-\frac{T_2}{T(\theta)}\right) \qquad (4)$$

where A_2, C_2, n_2, x_2, and T_2 are positive parameters.

C. Relating the Knock Integrals to available measurements

The proposed models (2) and (4) are expressed in terms of in-cylinder thermodynamics variables ($P(\theta)$, $T(\theta)$, and X) which are not directly measured on commercial line engines. Therefore, it is necessary to rewrite them in terms of different parameters.

During the compression and auto ignition phases, no combustion occurs. Thus, we can simply assume that the transformation is isentropic (see e.g. [18]). This thermodynamic assumption is supported by the fact that, during compression, the gas temperature is much lower than it is during combustion. In short, during the compression phase, wall heating losses are neglected.

Further, during the cool flame, since the Rate Of Heat released (ROHR) is low, we also assume that the compression is isentropic. This assumption drastically simplifies the modeling effort since there is no need for a cool flame ROHR model. At the light of the presented results (see Section VI), it is a reasonably valid assumption.

Thus, during auto ignition and cool flame, PV^γ and $TV^{\gamma-1}$ are both constant. In these relations, V represents the cylinder volume, which is perfectly known as a function of the crankshaft angle θ. We consider the ivc (intake valve closing) values as initial conditions for the isentropic transformation. The values (P_{ivc}, T_{ivc}) can be directly related to intake manifold measurements [11] (pressure and temperature sensors). These considerations yield

$$P(\theta) = P_{ivc} v_{ivc}(\theta)^\gamma \tag{5}$$

$$T(\theta) = T_{ivc} v_{ivc}(\theta)^{\gamma-1} \tag{6}$$

$$\text{with} \quad v_{ivc}(\theta) \triangleq \frac{V(\theta_{ivc})}{V(\theta)}$$

Finally, in models (2) and (4), X is constant and is equal to the intake manifold BGR. Its value can be estimated through an observer (see e.g. [3]). This observer only uses commercial line engine sensors, namely, the intake manifold pressure sensor, the intake manifold temperature sensor, and the exhaust air/fuel ratio sensor.

Substituting equations (5) and (6) into (2) and (4), yields

$$\mathcal{A}^{ai}(p(\theta)) \triangleq \mathcal{A}^{ai}_{ivc}(p_{ivc}, \theta) \tag{7}$$

$$\mathcal{A}^{cf}(p(\theta), m_{inj}) \triangleq \mathcal{A}^{cf}_{ivc}(p_{ivc}, m_{inj}, \theta) \tag{8}$$

where $p_{ivc} \triangleq (P_{ivc}, T_{ivc}, X)$. In particular, the cool flame Arrhenius function $\mathcal{A}^{cf}_{ivc}$ becomes

$$\mathcal{A}^{cf}_{ivc}(p_{ivc}, m_{inj}, \theta) = \frac{A_2\ (m_{inj})^{x_2}}{1 + C_2 X} {P_{ivc}}^{n_2} v_{ivc}(\theta)^{n_2\gamma} \exp\left(-\frac{T_2}{T_{ivc} v_{ivc}(\theta)^{\gamma-1}}\right) \tag{9}$$

Eventually, the auto ignition and cool flame model are thus expressed under the following form

$$\int_{\theta_{soi}}^{\theta_{soc}} \mathcal{A}^{ai}_{ivc}(p_{ivc}, \theta) \frac{d\theta}{N_e} = 1 \tag{10}$$

$$\int_{\theta_{soc}}^{\theta_{ecf}} \mathcal{A}^{cf}_{ivc}(p_{ivc}, m_{inj}, \theta) \frac{d\theta}{N_e} = 1 \tag{11}$$

IV. Control problem

At steady state, all the $p_{ivc} \triangleq (P_{ivc}, T_{ivc}, X)$ parameters are stabilized by the airpath controller to their reference values ($\overline{p}_{ivc}$). Further, the injection timing θ_{soi} is directly set to its reference value ($\overline{\theta}_{soi}$) by the fuelpath controller. A reference combustion takes place. The corresponding reference $\overline{\theta}_{soc}$ and $\overline{\theta}_{ecf}$ satisfy

$$\int_{\overline{\theta}_{soi}}^{\overline{\theta}_{soc}} \mathcal{A}^{ai}_{ivc}(\overline{p}_{ivc}, \theta) \frac{d\theta}{N_e} = 1 \tag{12}$$

$$\int_{\overline{\theta}_{soc}}^{\overline{\theta}_{ecf}} \mathcal{A}^{cf}_{ivc}(\overline{p}_{ivc}, m_{inj}, \theta) \frac{d\theta}{N_e} = 1 \tag{13}$$

During transient, $\delta p \triangleq p_{ivc} - \overline{p}_{ivc} \neq 0$. If fuel is injected at the reference time $\overline{\theta}_{soi}$, then the end of the cool flame differs from the reference combustion one. We propose to compensate any such known error δp with a corrective offset $\delta\theta_{soi}$ on the injection time reference $\overline{\theta}_{soi}$ so that the actual θ_{ecf} can equal $\overline{\theta}_{ecf}$. We can summarize this in the following problem (shooting problem):

Problem 1: Given $\overline{\theta}_{soi}$, $\overline{p}_{ivc}$,$\overline{\theta}_{soc}$, and $\overline{\theta}_{ecf}$ satisfying (12) and (13), consider $\delta p \in \mathbb{R}^3$. Find $\delta\theta_{soi}$ such that

$$\int_{\overline{\theta}_{soi}+\delta\theta_{soi}}^{\overline{\theta}_{soc}+\delta\theta_{soc}} \mathcal{A}^{ai}_{ivc}(\overline{p}_{ivc} + \delta p, \theta) \frac{d\theta}{N_e} = 1 \tag{14}$$

$$\int_{\overline{\theta}_{soc}+\delta\theta_{soc}}^{\overline{\theta}_{ecf}} \mathcal{A}^{cf}_{ivc}(\overline{p}_{ivc} + \delta p, m_{inj}, \theta) \frac{d\theta}{N_e} = 1 \tag{15}$$

V. Solution at first order

It might be difficult to find an explicit solution to Problem 1 when considering general Arrhenius functions of the form (2). A simple way to proceed is to look for a first order solution. Linearizing (14) (see [10] for more details on the linearization) around references values $\overline{\theta}_{soi}$, $\overline{p}_{ivc}$, and $\overline{\theta}_{soc}$ yields

$$\alpha^{ai}_{soc}\delta\theta_{soc} + \alpha^{ai}_{soi}\delta\theta_{soi} + \alpha^{ai}_{p}\delta p = 0 \tag{16}$$

where

$$\begin{aligned} \alpha^{ai}_{soc} &= -\mathcal{A}^{ai}_{ivc}(\overline{p}_{ivc}, \overline{\theta}_{soc}) \\ \alpha^{ai}_{soi} &= \mathcal{A}^{ai}_{ivc}(\overline{p}_{ivc}, \overline{\theta}_{soi}) \\ \alpha^{ai}_{p} &= \int_{\overline{\theta}_{soi}}^{\overline{\theta}_{soc}} \frac{\partial \mathcal{A}^{ai}_{ivc}}{\partial p_{ivc}}(\overline{p}_{ivc}, \theta) d\theta \end{aligned}$$

Similarly, linearizing (15) gives

$$\alpha^{cf}_{soc}\delta\theta_{soc} + \alpha^{cf}_{p}\delta p = 0 \tag{17}$$

where

$$\begin{aligned} \alpha^{cf}_{soc} &= \mathcal{A}^{cf}_{ivc}(\overline{p}_{ivc}, \overline{\theta}_{soc}, m_{inj}) \\ \alpha^{cf}_{p} &= \int_{\overline{\theta}_{soc}}^{\overline{\theta}_{ecf}} \frac{\partial \mathcal{A}^{cf}_{ivc}}{\partial p_{ivc}}(\overline{p}_{ivc}, m_{inj}, \theta) d\theta \end{aligned}$$

By eliminating $\delta\theta_{soc}$ in Equations (16) and (17), we get the desired correction

$$\delta\theta_{soi} = \frac{\alpha^{ai}_{soc}\alpha^{cf}_{p} - \alpha^{cf}_{soc}\alpha^{ai}_{p}}{\alpha^{cf}_{soc}\alpha^{ai}_{soi}} \delta p \tag{18}$$

In practice, this correction is easily computable, since all parameters α depend on measured values (p_{ivc}), mapped values ($\overline{p}_{ivc}$,$\overline{\theta}_{soi}$) or calibrated models ($\mathcal{A}^{ai}$,$\mathcal{A}^{cf}$). In view of application, both terms α^{ai}_{p} and α^{cf}_{p} can be simplified using the actual expressions of functions $\mathcal{A}^{ai}$, and $\mathcal{A}^{cf}$.

VI. Experimental results

A. Experimental setup

All experimental results presented here have been obtained on a four cylinder direct injection Diesel engine running in HCCI combustion mode. Exact specifications are reported in Table II. The engine is an upgrade of a turbocharged multi-cylinder commercial line engine. A low pressure EGR circuit is used. It extracts hot burned gases downstream of the turbine, and introduces them upstream of the compressor. A valve allows the EGR rate to be controlled. Finally, both the air and the EGR circuits include a cooler to keep the intake manifold temperature around 330K.

TABLE II
EXPERIMENTAL SETUP

Bore × Stroke	87.0 × 92.0 mm
Number of cylinders	4
Compression ratio	14.0:1
Displacement	2.2 Liters
Injection device	Solenoid
Maximum injection pressure	1600 bar
Piston bowl design	NADI™
Intake Valve Closing	$\theta_{ivc} = 232$deg (360 is Top Dead Center)

B. Controller design

The general control scheme is presented in Figure 5. It includes the strategy proposed in this article to control the *ecf*. This new strategy is included in the dark-grey box "correction calculation" which implements equations (18). In this setup, the injection crankshaft angle θ_{soi} is not simply set to its reference value $\overline{\theta}_{soi}$, but is corrected according to the airpath errors δp.

The controller has been integrated in the complete IFP engine control system developed with Matlab/Simulink. RTW (Real Time Workshop), and xPC target toolboxes are used for real time code generation. The task execution time of the proposed controller is about 20 μs on a 4.8Ghz target.

C. Sensors

The proposed combustion controller does not need any in-cylinder sensor. More precisely, it only uses intake manifold measurements (pressure and temperature) and the exhaust air/fuel ratio sensor (for the BGR observer).

D. Results

Figures 6 reports experimental results. The scenario considers an increasing torque transient at 1500 rpm. For sake of comparisons, the plots contain the same trajectory with (bold blue curves) and without (dark red curves) the proposed correction. The torque trajectory is presented in Figure 6a. As mentioned in Section II, the airpath control regulates the intake manifold pressure and BGR around their reference values in order to meet torque demands (see Figures 6b and 6c). The not-instantaneous tracking of airpath reference values makes the new fuelpath controller correct the injection crankshaft angle reference value (see Figure 6d). The injection crankshaft angle value sent to the injectors is then different from its reference value. The influence of the proposed combustion control on the engine noise is depicted in Figures 6e. Evolutions of the middle of the combustion are then reported in Figures 6f.

During these transients, without any particular combustion control, combustion is drastically delayed (see the second and third transients in Figure 6f). In response, the proposed controller acts on the start of injection to bring the combustion forward. Actually, the correction seems to be filtering the start of injection reference to compensate for airpath dynamics. The result clearly appears in Figure 6f. The middle of combustion closely follows its step reference. The large overshoots have disappeared. Improvements can also be seen on the engine noise in Figure 6e. Noise transients have been smoothed out. This results in significant improvements in acoustic comfort. Similar results were obtained for a decreasing torque trajectory under various load and speed conditions.

These results clearly prove that the strategy proposed in section V, and implemented in Section VI, brings a solution to the control problem 1. In spite of airpath regulation errors, such as slow transients dynamics or hardware malfunctions, middle of combustion occurs at the reference timing. Thus, combustion is now robust to airpath errors. Interestingly, the proposed linear combustion control has another property at steady state. Since the airpath controller regulates the airpath values, the airpath error vanishes at steady state, just as the start of injection correction. The combustion is thus exactly the reference one at steady state (the combustion defined by the optimal trade off).

VII. CONCLUSION AND FUTURE WORK

An improvement for the fuelpath control strategy of HCCI diesel engine has been presented. Instead of directly setting the injection crankshaft angle to its reference value, we propose to synchronize the fuelpath to the airpath. This controller is mainly based on the linearization of an auto ignition delay model (KIM) and a cool flame model. Provided an estimation of in-cylinder conditions at the *ivc* (which in our case is inferred from intake manifold signals), this method is very general. It can be applied to engine with external or internal gas recirculation, to naturally aspirated engine, throttled engines and/or with turbochargers.

The presented experimental results stress the relevance of this new approach. The auto ignition and the cool flame are the critical phenomena for the whole combustion process. Controlling the end of the cool flame permits to improve the robustness of the HCCI combustion towards airpath errors.

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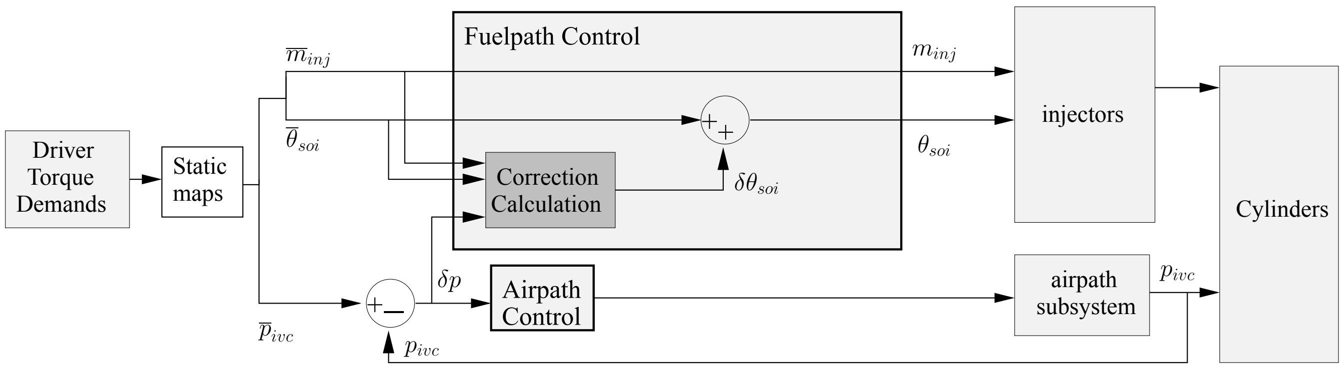
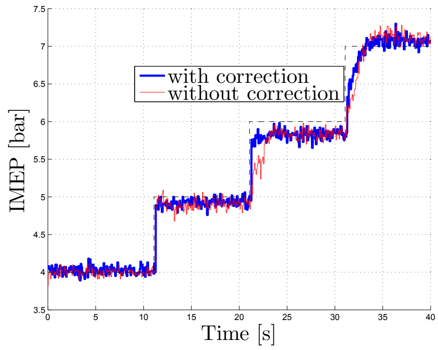
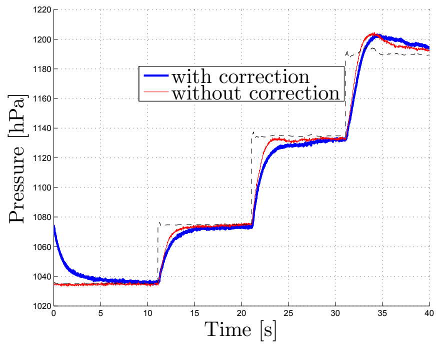


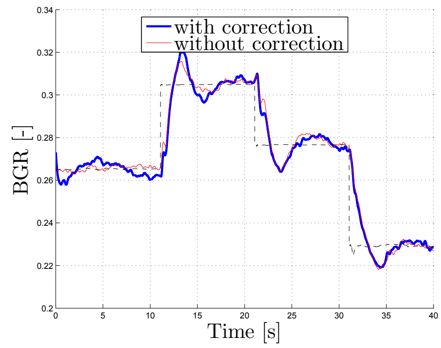
Fig. 5. Design of the new control strategy. The dark grey block has been added, counterbalancing airpath errors δp with an injection crankshaft angle offset.



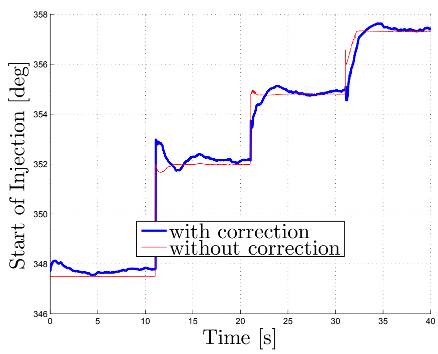
(a) Torque



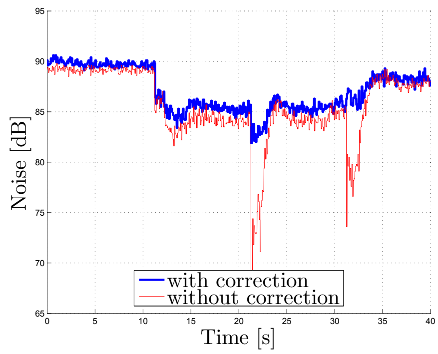
(b) Intake Manifold Pressure



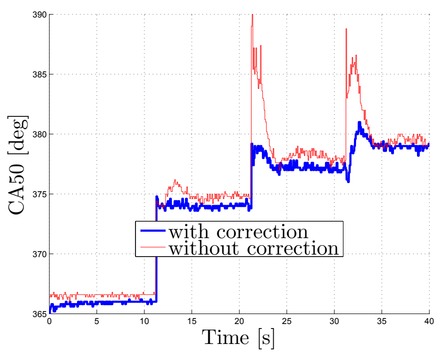
(c) Intake Manifold Burned gas rate



(d) Injection crankshaft angle (360 correspond to the Top Dead Center)



(e) Noise



(f) CA_{50}

Fig. 6. Experimental results on a 4-cylinder HCCI engine with direct injection. Charge increase at constant speed of 1500rpm with and without the new fuelpath control strategy.

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