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Dossier

Diesel Engines and Fuels: a Wide Range of Evolutions to Come
Motorisation et carburants Diesel : un large spectre d'évolutions à venir

HCCI Diesel Engine Control Design using Advanced Simulation with Real Time Capabilities

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Résumé — Développement d'un contrôle moteur Diesel HCCI basé sur la simulation avec des performances temps réel — Un simulateur moteur est un outil très flexible pour réaliser un grand nombre de tests sur une large plage de fonctionnement et pour accéder à des informations supplémentaires à celles mesurables expérimentalement. Dans ces conditions, l'utilisation de la simulation pour le développement des lois de contrôle moteur peut réduire significativement la durée des essais au banc tout en donnant accès à des grandeurs utiles à la compréhension du système. Basé sur des modèles physiques avancés permettant, par exemple, de représenter la combustion Diesel HCCI, un simulateur moteur est en mesure d'aider à la conception du contrôle, du développement des lois de commande jusqu'à la calibration sur plateforme temps réel.

Le but de cet article est de démontrer la pertinence d'utiliser la simulation pour le développement du contrôle moteur dans les contextes technologiques modernes. Cette démonstration est réalisée à travers l'application au cas complexe du contrôle d'un moteur Diesel HCCI, mode de combustion considéré comme l'une des voies les plus prometteuses mais également les plus délicates à contrôler dans la perspective d'une forte réduction des émissions polluantes à la source.

Abstract* — *HCCI Diesel Engine Control Design using Advanced Simulation with Real Time Capabilities — *An engine model is a very flexible tool to perform a large amount of tests over a wide range of operating conditions and to access to more detailed information than those available with experiments. Therefore, the use of the numerical engine simulation in the control development process can significantly reduce the test time needed on real hardware while providing more information for understanding the system physics. Based on high-performance physical models such as HCCI combustion model, the virtual engine simulator is able to support the control design from controller development to real-time calibration.*

The aim of this communication is to show the relevance of using such a model-based approach for current control issues. The demonstration is achieved through the challenging case of Diesel HCCI combustion engine. As an alternative to expensive and complex after-treatment solutions, Diesel HCCI engine takes benefit of high exhaust gas recirculation rate to limit in-cylinder pollutant production. Therefore, it is recognized as one of the most promising ways for new generation of CI engines but also one of the most complex engine technologies to control due to the sensitiveness of HCCI combustion mode to small perturbations.

ABBREVIATIONS

AFR	Air Fuel Ratio
ATDC / BTDC	After / Before Top Dead Center
BGR	Burnt Gas Ratio
BMEP	Brake Mean Effective Pressure
CA	Crank Angle
EVC / IVC	Exhaust / Intake Valve Closing
EVO / IVO	Exhaust / Intake Valve Opening
IMEP	Indicated Mean Effective Pressure
MBT	Maximum Brake Torque
Ne	Engine Speed
SI	Spark Ignition
VVT	Variable Valve Timing
atm	Atmosphere
man	Manifold
in	Intake
super	Supercharging

INTRODUCTION

The engine control design is becoming an important part in the development of new engines. It has to accurately control more and more complex engine technologies while dealing with the NOx/PM trade-off imposed by stringent emission standards. The control issues represent a crucial goal for the success of an innovative technology such as HCCI engine. The conventional control conception methodology is mainly based on a small part of off-line development using simple modelling approaches such as transfer functions or maps in the early stage of the design process and a large part of development and tests directly on the testbed which is quite expensive. This paper presents another methodology for advanced control design based on engine simulation. It consists in using the engine model as a virtual bench for a large part of the control conception process before or in parallel with the bench tests. An engine model allows to perform easily a large amount of operating conditions and gives access to detailed information on the engine behaviour which are not necessary easy to measure on the testbed. Therefore, the engine simulation is an essential tool to improve control law design efficiency in such a context. As a matter of fact, extending the use of numerical engine models in the control development process can significantly reduce the test time needed on real hardware while providing more information for understanding the system physics. Based on high-performance physical models such as HCCI combustion model, the virtual engine simulator is able to support the control design from controller development to real time calibration. The aim of this communication is to demonstrate the relevance of using such an approach with the example of the current technological challenge of the HCCI engine control design.

After a generic presentation of this methodology, the way the engine simulator has been used for the control design in a HCCI Diesel engine development project is presented step by step and illustrated with both simulated and experimental results.

1 HCCI ENGINE CONTROL ISSUES

1.1 Experimental Set Up: the NADI™ Engine

IFP has developed a combustion system able to reach near zero particulate and NOx emissions while maintaining expected performance of DI Diesel engines. The dual mode engine application called NADI (Narrow Angle Direct Injection) applies HCCI combustion at part load and switches to conventional Diesel combustion to reach full load requirements. Figure 1 shows the experimental set up used in this study. The engine is a four cylinder light-duty direct injection Diesel engine working in both conventional combustion mode and HCCI mode. The engine is a standard production 2.2 litre (87 mm bore and 92 mm stroke). The combustion chamber has been designed following the NADI™ concept specifications with a specific piston bowl and a compression ratio of 14:1 [1]. The intake ducts of the cylinder head are designed in order to adapt the swirl motion to the required swirl number of 1.3 at BDC. All the results shown in this paper, including full load conditions were obtained with the same swirl number, without any inlet duct closure. The HCCI Diesel engine application presented here is an engine equipped with a variable geometry turbocharger (VGT). This latter was chosen in order to reach the power target fixed at

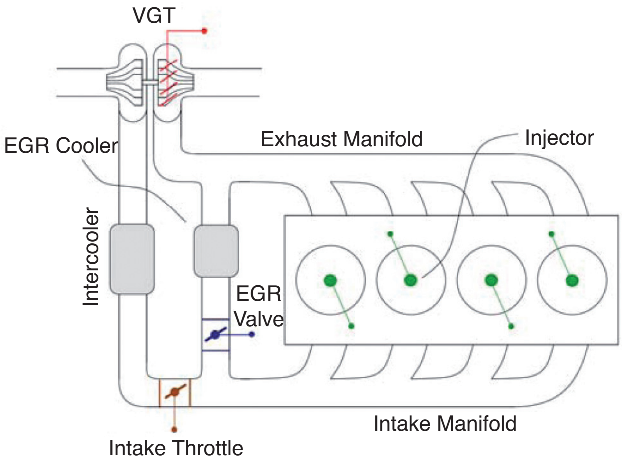

Figure 1

Four cylinder NADI™ HCCI engine.

full load. The air path architecture includes an air cooler to maintain the intake manifold temperature below 50°C at full load. A high pressure EGR circuit was chosen for this application in order to reach fast settling time in transient operations. This type of EGR circuit extracts burnt gas in the exhaust manifold upstream of the turbine. The introduction of the exhaust gas is made downstream of the compressor, close to the intake manifold, by means of a Venturi device to improve burnt gas mixing with fresh air. The EGR circuit includes a cooler which is supplied by an external low temperature water circuit. The EGR cooler may be by-passed to increase the intake manifold temperature. This limits the HC and CO emissions at very low engine loads.

1.2 Air Path Control Issues

The engine management system controls the engine actuators in order to produce a fast torque response while maintaining low emission levels. The exhaust gas recirculation is used to limit the flame temperature and reduce the NOx emissions at low engine load. During HCCI operations, the engine works with very high levels of burnt gas ratio (BGR). The engine runs under conventional conditions at higher loads with lower BGR levels (part load) or no BGR (high load). The turbocharger works mainly at high engine loads, where boost pressure is needed to reach the expected IMEP. From the air path point of view, the estimation and control of intake manifold variables have to be considered. The intake manifold pressure and the BGR are the standard feedback variables for HCCI combustion control. It would obviously be desirable to separate the effects of the turbocharger and the EGR vane and to deal with these actuators with two separated control loops. However, this proves to be difficult since the air loop and the EGR loop are strongly linked. On one hand, the energy taken by the turbine on the exhaust gas depends on the mass flow, which is directly linked with the EGR flow. On the other hand, the EGR flow depends on the pressure difference between the exhaust and intake manifolds, which in turn depends on the turbocharger operating conditions. The air intake system must be considered as a multivariable system, with two control inputs (EGR vane and VGT positions) and two state variables (air and EGR mass flows) as represented in Figure 2. Several approaches have been published to deal with this control issue [2-5]. A simple solution may consist in the deactivation of the turbocharger control loop when high levels of BGR are needed.

1.3 Fuel Path Control Issues

Since the combustion takes places in a homogeneous mixture the soot production is lower. The 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. However, the potential benefits of HCCI combustion in terms of emissions are counterbalanced by its high sensitivity to in-cylinder thermodynamic conditions since the HCCI combustion is controlled by chemical kinetics. Practical application of HCCI on car engines faces fundamental control challenges. The main HCCI combustion control issues are described below:

- Fuel loop control adaptation to the air path dynamics. The fuel loop dynamics is faster than air loop dynamic. Thus, the engine control must take into account of these two separated dynamics avoiding related perturbations.

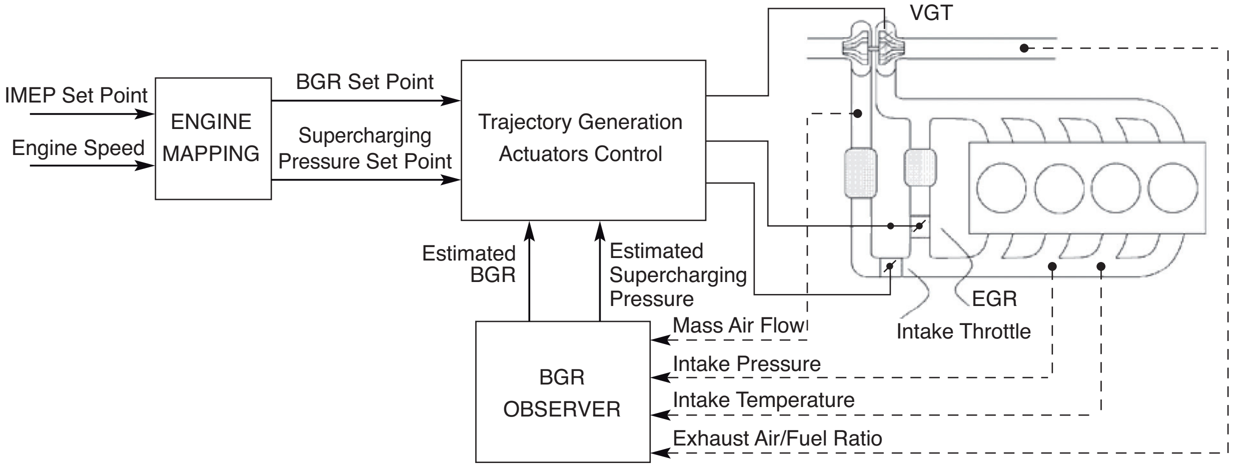

Figure 2

Schematic representation of the BGR observer/controller for HCCI Diesel engine.

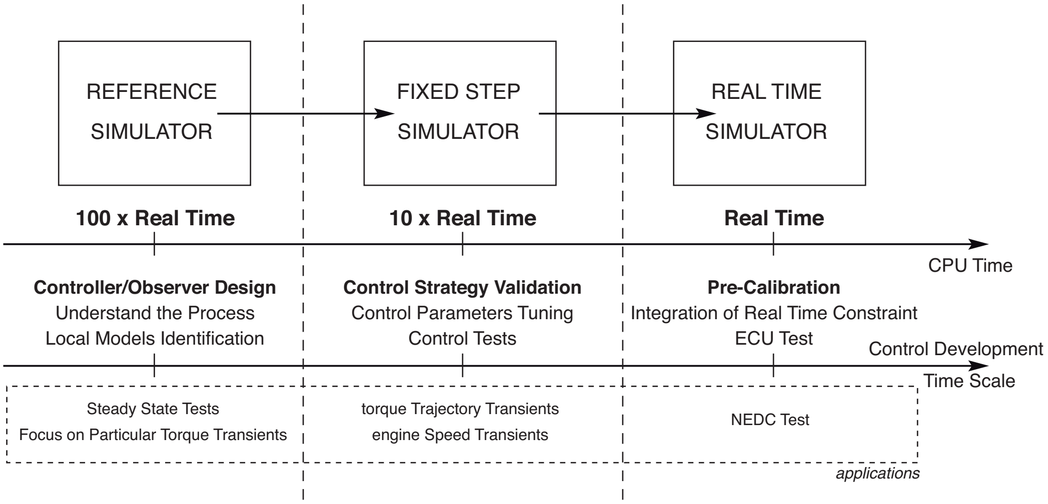

Figure 3

Schematic diagram of the engine simulation based control design methodology.

The HCCI combustion makes the control task much more difficult than with conventional Diesel combustion due to higher BGR variations at combustion mode transition.

- Bad repartition of BGR in multicylinder engines. The intake system design and the aerodynamic phenomenon may lead to some cylinder to cylinder BGR level discrepancies. This tends to increase emissions and to unbalance the IMEP production cylinder to cylinder generating harshness and noise. Moreover, the BGR overshoots may cause misfires and engine instability.
- Cycle-to-cycle coupling. There is cycle-to-cycle coupling through the temperature of the reintroduced air/BGR mixture (fast dynamic) and to the engine block temperature (slow dynamic). This coupling is stronger during mode transition (especially from high loads to part or low loads).
- Stability of HCCI combustion at part load. The HCCI combustion becomes harder to control at part load when the engine operates near to the stoichiometry.

The challenge with HCCI engine is to maintain the adequate thermodynamic conditions for each cylinder by using modern actuators fitted to the engine. This involves applying robust control techniques. The key point to ensure stable HCCI operating conditions is to optimise the injection settings, BGR level and boost pressure. Combustion control is a key point for HCCI engine expansion to mass production vehicles.

2 ENGINE MODELLING FOR CONTROL DESIGN

2.1 Methodology

Three types of modelling levels can be used for engine simulations:

The CFD 2D/3D simulations are dedicated to the local studies and allow to capture the small scale in-cylinder phenomena such as turbulent combustion and chemical kinetics. The characteristic timescale of CFD simulation is to the order of the turbulent timescale. The CPU time of such a simulation allows to run a few engine cycles.

A simplest modelling approach consists in representing the engine with operating condition look-up tables. This level allows to get CPU time lower than real time. This kind of simulator is used to run standard driving cycle with a low dynamic vehicle simulator. The characteristic timescale of vehicle simulation is to the order of 0.1 second.

The engine system modelling is at a turning point between the two previous approaches. This approach involves phenomenological or empirical models and allows to represent the complete engine with a characteristic timescale to the order of 0.1 crankshaft degree. Thanks to dedicated code optimisation to limit the CPU time cost, the engine system model allows to reproduce accurately the behaviour of the engine during transient such as driving cycle while reaching the real time in certain conditions. Therefore, this approach is well adapted to a wide range of use for the engine development [6], notably as a support for engine control design from the control development to the hardware-in-the-loop validation [7].

During the model based control development cycle, the control requirements of engine simulation do evolve. Different versions of the engine simulator take part in the control algorithm development and testing. The methodology used in this paper is based on [7] and uses three engine simulators as presented in Figure 3.

The engine simulation takes part of the control laws development at each step of the control design with the three following simulator versions:

– *The reference engine simulator (Software-in-the-Loop platform)*. The goal of this simulator is to provide an engine model with a high representative capability of the real engine. It allows to investigate engine behaviour and to better understand the phenomena involved on the processes to control. From the control point of view, this simulator is mostly used for steady state and short transient tests to support controller/observer design. This simulator is also very important for the engine simulation part because it is used as the reference accuracy level for the others simulator versions. It is therefore the most CPU expensive version (about 20 times the real time with a standard 3 GHz PC in our case). The reference engine simulator is run in the AMESim/Simulink co-simulation environment.

– *The fixed step simulator (Software-in-the-Loop platform)*. Once the basis of the control components is defined, the engine fixed step simulator is required to achieve further tests and to validate the control algorithms under a wider range of operating conditions before experimental validation. This simulator has to be able to be computed with a fixed time step solver and is supposed to run the engine tests with a reduced CPU time (about 4 times the real time with a standard 3 GHz PC in our case) while preserving the relevant physical behavior obtained with the engine reference simulator.

– *The real time simulator (Hardware-in-the-Loop platform)*. Finally, the real time simulator is used to perform hardware-in-the-loop (HiL) simulations with dedicated platforms such as xPC or dSPACE which allow to test the control in a configuration very close to the bench configuration. The engine simulator is therefore required to run in real time which is achieved thanks to specific adaptations. The control law performances and robustness are then validated under much more realistic conditions than with the off line platform. For example, such a kind of simulations can help pre-calibration of controller parameters (especially when gain scheduling strategies are chosen). In addition, with the HiL platforms, the real hardware can be tested on simulated components. As an example, the HiL tests may consist in coupled simulation of electronic control unit (ECU) and real time simulator.

2.2 HCCI Engine and Vehicle Modelling

The HCCI engine and vehicle simulators are designed in the AMESim's modelling environment edited by LMS IMAGINE.Lab. Thanks to the AMESim's platform coupling facilities, the simulators take mainly benefits from IFP-ENGINE library for engine modelling and IFP-DRIVE and POWERTRAIN libraries for vehicle aspects. Further details about this simulation tool can be found in [8]. The goal of engine simulation is to supply the relevant model level for the specific application. User expectations have to be accurately understood in order to achieve the optimum trade-off between physical description and calculation time cost. The engine control design has strong CPU time constraints because it needs to perform a large amount of engine operating points and is oriented to real time environment. The vehicle simulator including the HCCI engine diagram is presented in Figure 4. A specific dual mode combustion model has been introduced as an additional development in order to improve the model accuracy for both HCCI and conventional combustion modes [9]. The model reproduces the main HCCI combustion features: multi-pulse injection, auto-ignition delay, EGR effect including cool flame. The model fits the heat release behaviour in the whole range of operating set points.

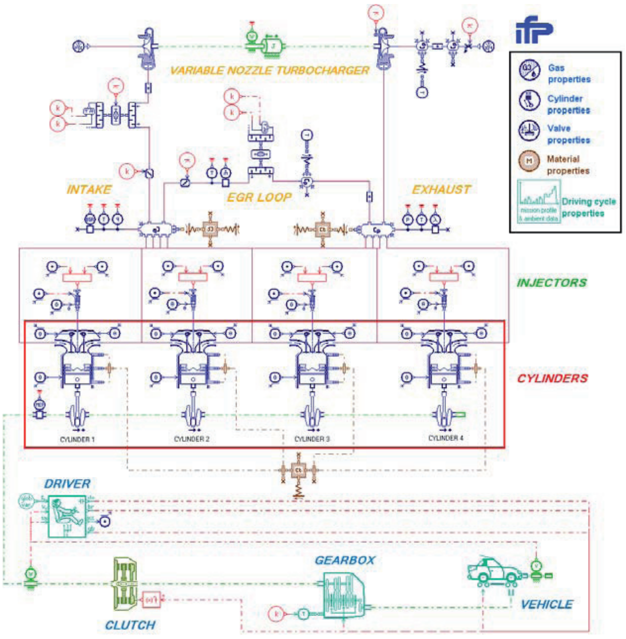

Figure 4

HCCI engine simulator diagram.

2.3 Simulator Validation

Before being used for engine control design, the simulators are calibrated as close as possible to the real engine. As experiment results are available from engine bench, the comparison with test bed results is the best way to achieve and validate this calibration for the HCCI engine simulator. For the complete vehicle simulator, the goal is to be able to test and achieve a first calibration with simulation before the engine is embedded in the prototype vehicle. The vehicle simulator calibration mainly consists in powertrain characterization and qualitative results.

2.3.1 Engine Simulator Results

To be complete, this process shall be performed for a large amount of operating set points, under steady state and transient conditions, with an open loop and a closed loop control. According to the experimental acquisitions, mean but also instantaneous engine characteristic values from simulator and testbed may be compared. Finally, the goal is to obtained satisfactory results from the three versions of the engine simulator (reference, fixed step and real time) with the same parameter set for all the validation tests.

Figure 5 presents the phenomenological combustion model behaviour for various load set points at 1500 rpm. Thanks to a relevant modelling approach and an adapted model parameter calibration, the cylinder pressures from the numerical model fit well with the experimental results. One crucial step in the real time engine simulator development is the accuracy of the knowledge combustion model. The parameters of this model are issued from a learning

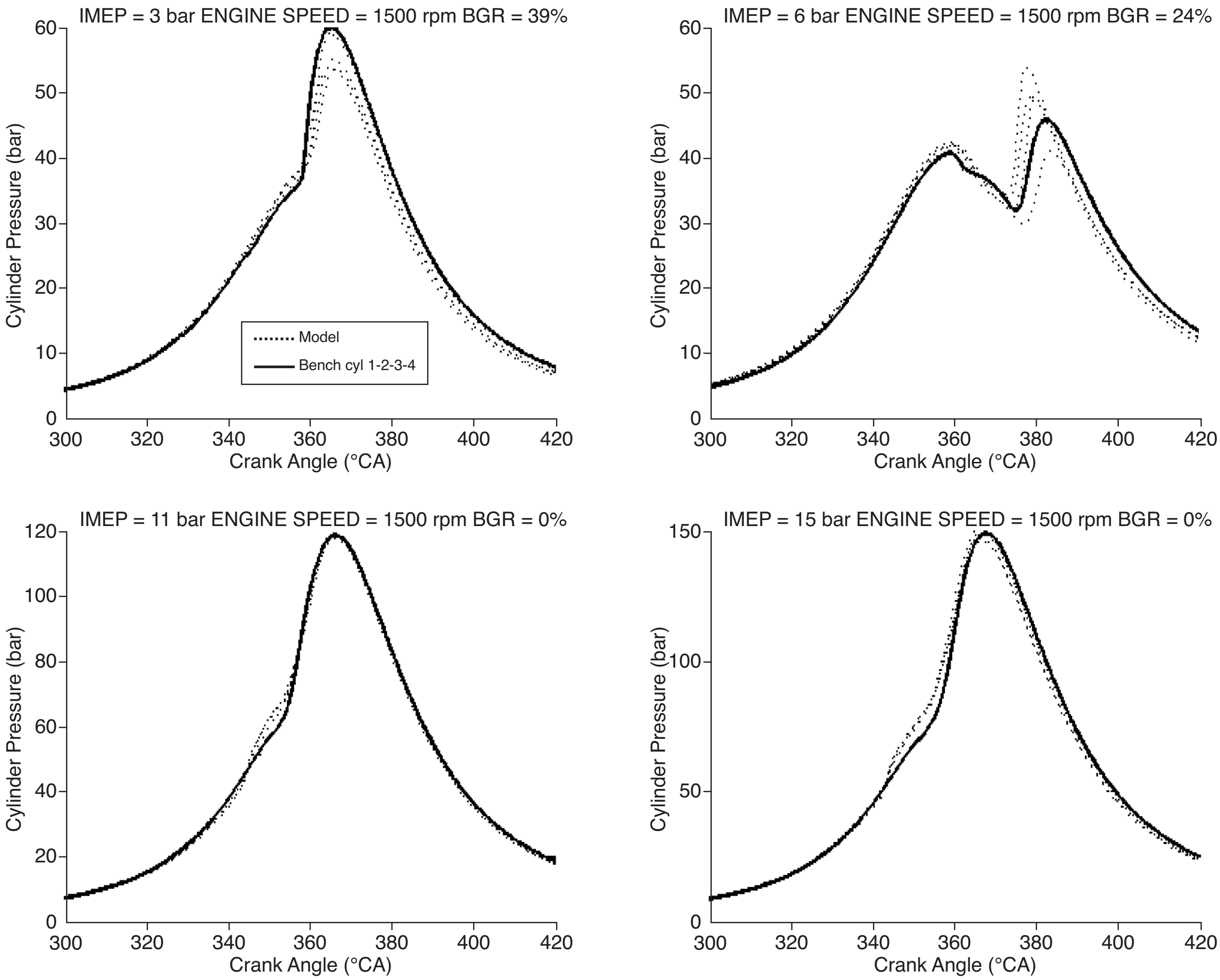

Figure 5

Model/bench cylinder pressure comparison for various loads at 1500 rpm for the phenomenological combustion model.

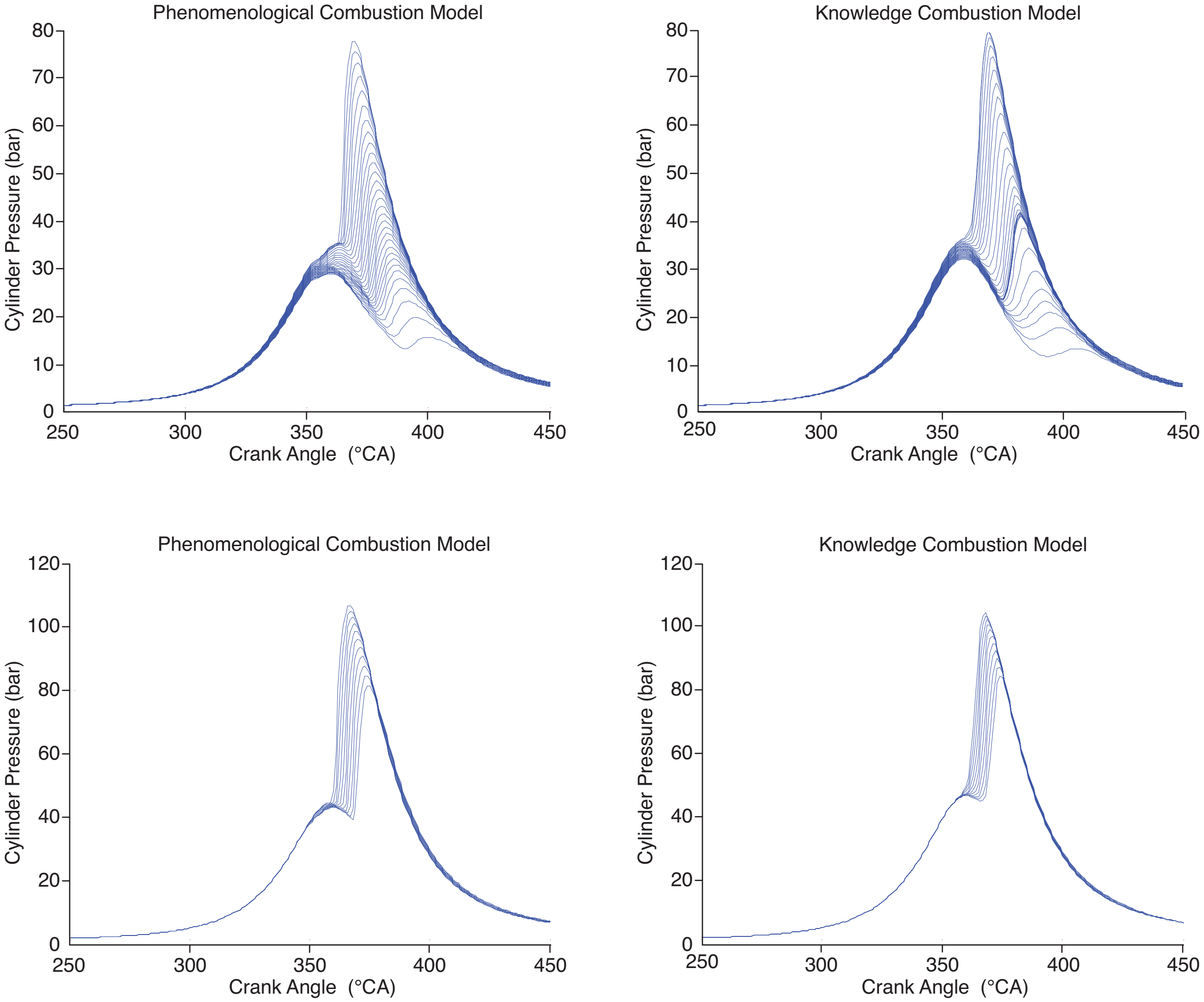

Figure 6

Phenomenological/knowledge combustion model comparison for BGR variation at 5 bar IMEP (top) and for start of main injection variation at 9 bar IMEP (bottom) at 1500 rpm.

process performed on phenomenological combustion model campaign results. Through this specific process, the combustion response sensitivity to the main control variables such as BGR or start of injection has to be preserved. Figure 6 presents two examples of cylinder pressure obtained with the phenomenological and the knowledge model at 5 and 9 bar IMEP (1500 rpm). For the 5 bar IMEP HCCI combustion set point, a BGR variation has been performed and for the 9 bar IMEP conventional combustion set point, a start of injection variation has been performed. These two examples demonstrate the good accuracy of the knowledge model which can be implemented in the fixed step simulator.

Since noise limitation is a crucial control issue for HCCI engine, especially during transient from conventional combustion mode to HCCI combustion mode, a validation of the model representative capability on noise estimation has been achieved too. It consists in comparing the maximum cylinder pressure gradient (noise level indicator) and its angular location (noise location indicator). The results for a complete load variation at 1500 rpm are presented in Figure 7 and show a good agreement with bench data. We can therefore conclude that the combustion numerical result accuracy allows us to use the simulator for the main combustion control issues since it is not crucial to take into account the cylinder to cylinder variations as it is the case for the baseline control development.

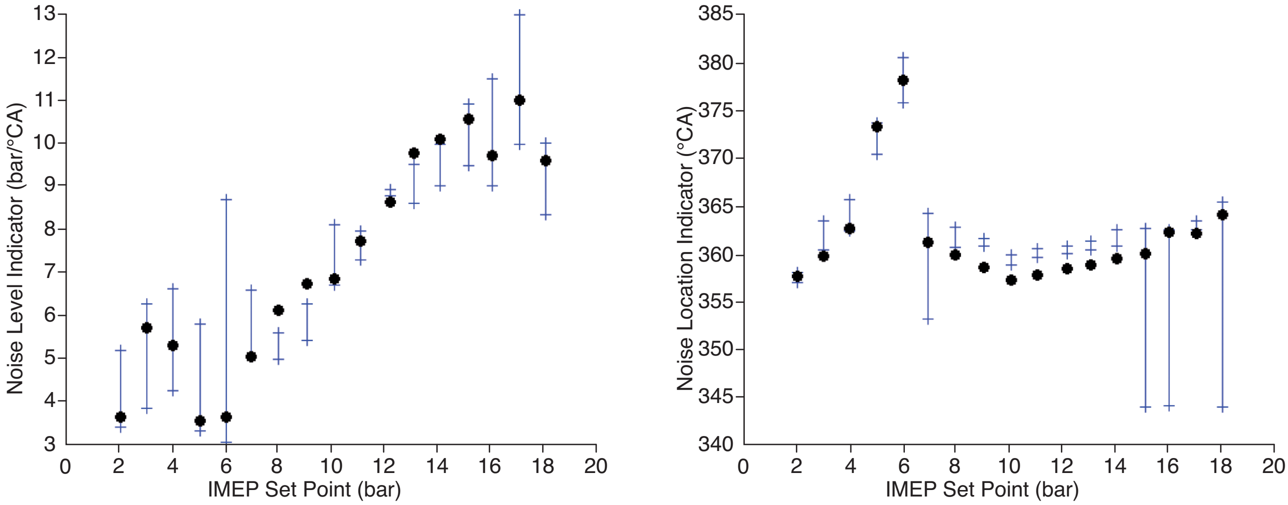

Figure 7

Model/bench noise comparison for a load variation at 1500 rpm (dots for model results, segments for bench response range).

Figure 8 presents the reference simulator results obtained under steady state conditions in a wide range of operating set points. 86 set points have been computed from low load to full load for five engine speeds: 1500, 2000, 2750, 3000 and 4000 rpm. The comparison with the bench data shows the good agreement on IMEP, mass air flow, exhaust temperature and air/fuel ratio (AFR). This validation assesses the load/speed area that can be investigated using the engine simulator.

Since engine control is dealing with transient issues, the engine simulators have to be tested under transient conditions. For example, the fixed step simulator has been run on a load trajectory at 1500 rpm under open loop conditions. That means that the trajectory has been first performed with the control at the test bed. The instantaneous trajectories have been recorded for all the actuators and then applied on the engine simulator. This open loop method is a really severe test for the engine model because any small deviation to real engine test may be emphasised by the absence of engine control. Some comparison between simulator and bench results are plotted in Figure 9. Turbocharger speed, exhaust equivalence ratio and burnt gas ratio predicted by the simulator are in good agreement with the measured engine behaviour in term of dynamics.

2.3.2 Vehicle Simulator Results

The goal of the project is to embed the NADI™ engine in a Renault VelSatis vehicle. At first, the relevant technical properties of the VelSatis have been introduced in the vehicle simulator. Then, a simulation has been run under the 200 seconds of the ECE driving cycle with closed loop control in order to evaluate the qualitative behaviour of the vehicle simulator dynamics. Typical values are presented in Figure 10 and show that the vehicle simulator with a crank angle degree HCCI engine model allows to access a wide range of dynamics, from vehicle velocity to BGR peaks.

3 CONTROL DEVELOPMENT AND VALIDATION USING THE SIMULATORS

3.1 Reference Simulator for Controller/Observer Design

Using simulations for control design enables to test and validate the developed strategies, in order to debug the algorithms and anticipate from a test on the real engine or vehicle. The most important roles of the engine reference simulator are to help understanding of the underlying physics and to provide information about the fundamental aspects that need to be taken into account in the control strategies.

3.1.1 Simulation-Based BGR Observer Design

For example, let's consider the design of a burnt gas ratio (BGR) observer. The amount of burnt gas present in the intake manifold is an important factor for the quality of the

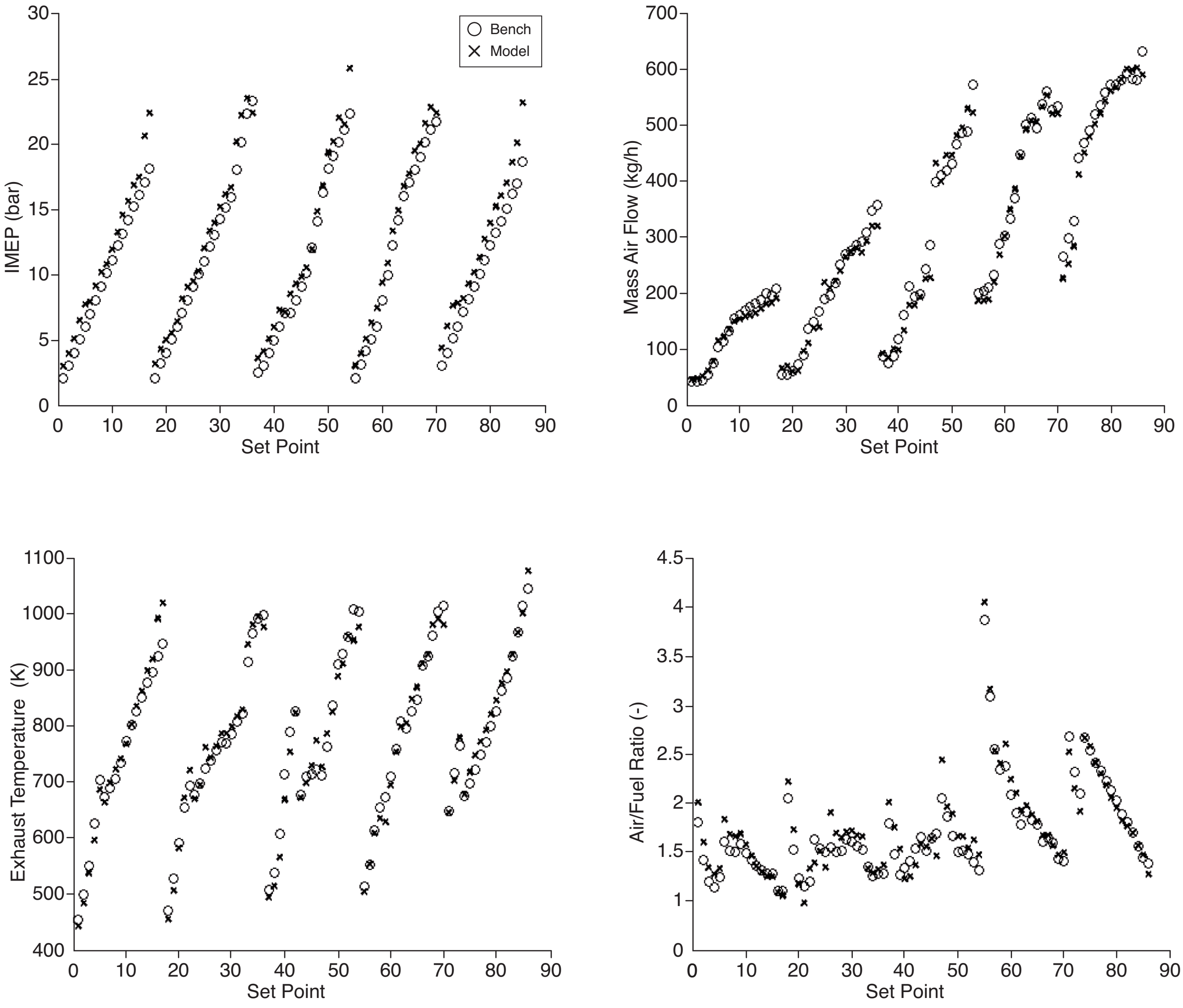

Figure 8

Model/bench result comparison on load and speed steady state set points (reference simulator).

combustion in the cylinder. This variable is difficult to measure on an engine, and has to be reconstructed from other measured variables (fresh air mass flow, intake manifold pressure and temperature). The observer development can be performed only on a reference model because on a real engine, the measurement of BGR in the intake manifold is simply not available under transient conditions. Figure 11 shows a comparison between the BGR value given by the reference model and its estimation computed using an observer based on conservation equations.

3.1.2 Validation on Bench

Figure 12 shows tests results performed on the engine test bench. The presented tests were done on a load transient at 1500 rpm. The top figure shows the IMEP set points, the middle and the bottom figures show the EGR and fresh air mass flow control. The EGR mass flow control is very accurate and fast, while the fresh air flow is much slower. This reflects the difference in the dynamics of the two actuators: the EGR valve acts much faster on the system than the VGT which has mainly an effect on the turbine permeability and

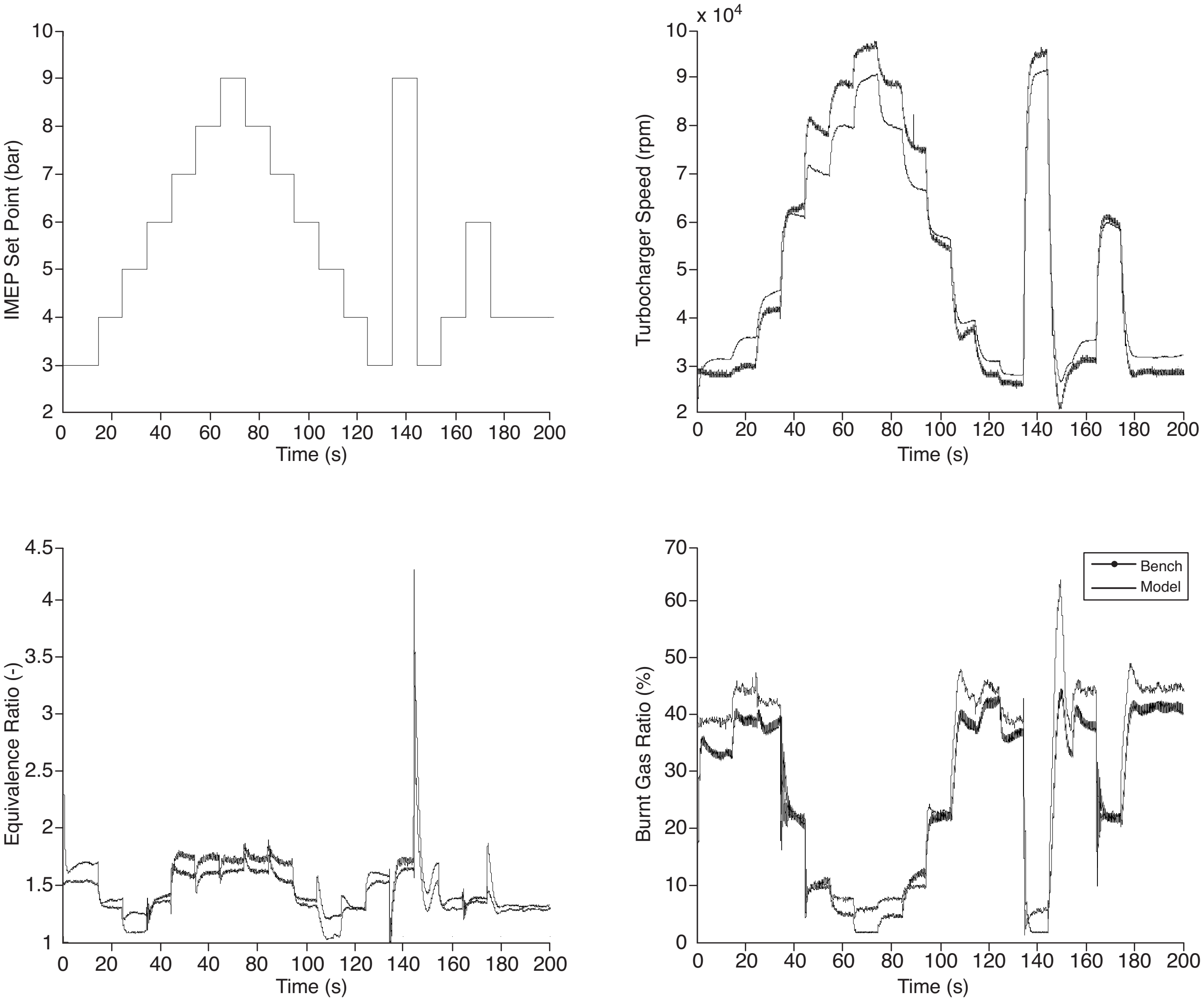

Figure 9

Model/bench comparisons for a load trajectory at 1500 rpm under open loop conditions (Fixed step simulator).

efficiency. In this case, the calibration of the control strategies, including controller and observer, was done entirely on the reference simulator, and tested without any modification on the engine bench, giving good results with a very short test time.

3.2 Fixed Step Simulator for Control Strategy Validation

The typical CPU time of the reference simulator in AMESim is roughly 10 to 100 times the real time, which is reasonable to focus on phenomenon with dynamics of about 10 Hz to 100 Hz. However, the CPU time of the reference model becomes unproductive in the calibration control strategy phase or during their development when low frequency aspects are studied, for example vehicle control (idle speed, cut off) which requires to consider slower dynamics (about 1 Hz to 10 Hz) and long time simulations (as driving cycles).

One efficient way to reduce the CPU time is to run the engine model with a fixed step solver. Depending on the chosen time step, some dynamics may be filtered leading to a loss of information. As a consequence, going from the reference simulator to the fixed step simulator needs an adaptation phase which may include specific modelling approaches. As

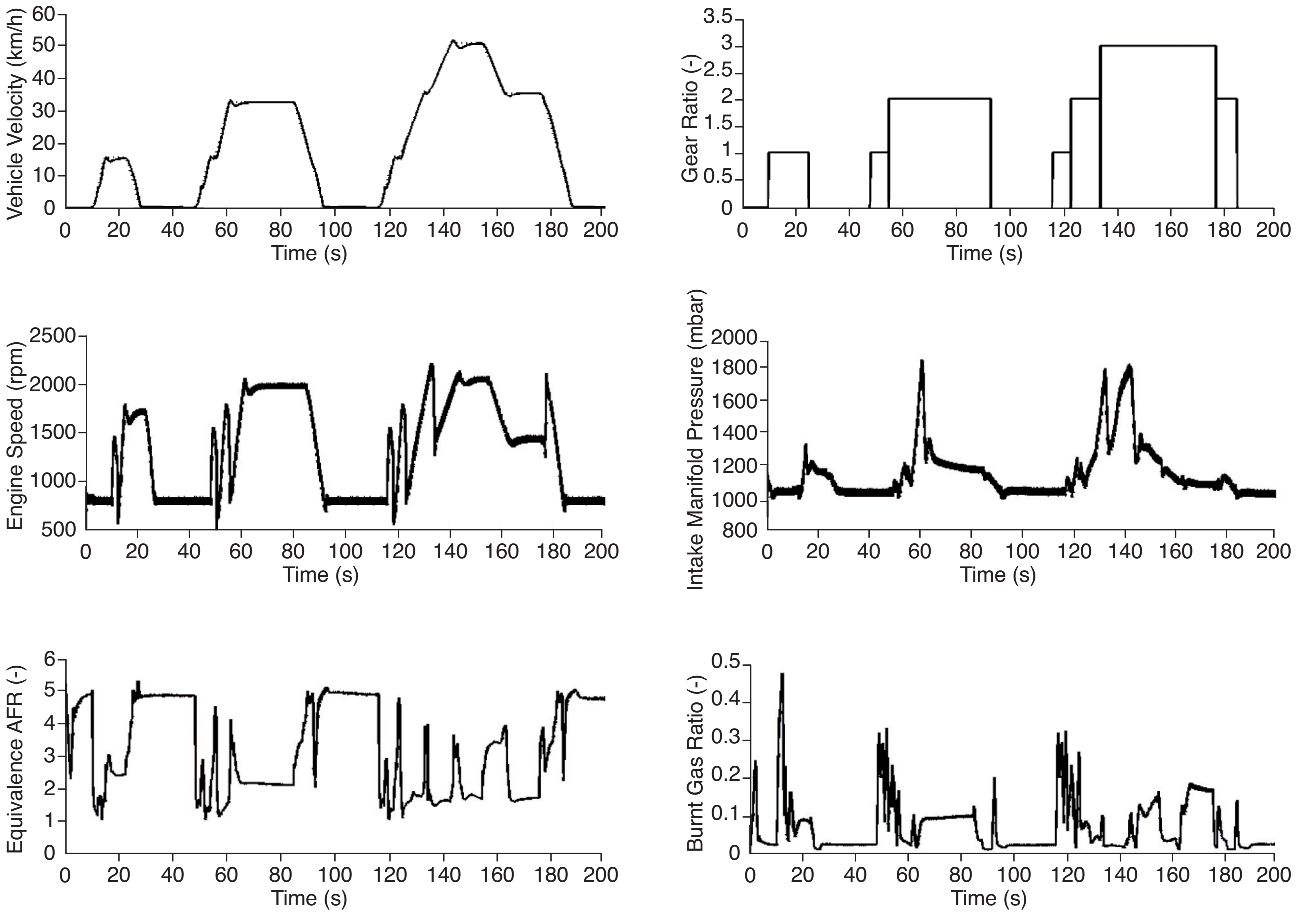

Figure 10

Vehicle simulator results under ECE driving cycle conditions with a closed loop control (Fixed step simulator).

mentioned above, one must keep in mind that the relevant dynamics in the model are the ones that influence the strategies that are being developed. While considering slower strategies, the time step of the model can be chosen longer. This second simulator has to be validated either against real measurements when available, or against results from the reference one, which is sufficient to be used in a predictive way when no real test results are available.

3.2.1 Focus on Injection Strategy

In this section we illustrate the advantage of using a model in the development of fuel control strategy. This example briefly describes a fuel path control strategy to limit the smoke emission during large IMEP transients (tip-in). The smoke emission is mainly caused by a lack of air during transient operating conditions. Concerning the dynamics, the fuel loop is very fast and the air loop related variables (boost pressure, air flow, BGR level) are slower. This effect is increased when the fuel is fed to the cylinder without taking into account of the air dynamics. Without any specific strategy, the engine emissions raise and misfires may occur (in cylinder mixture is saturated).

Here, we use the model to tune the smoke limiter strategy during the tip in transient. The tuned parameter is the in cylinder equivalence ratio limits. The output variable is the exhaust equivalence ratio which is correlated to the smoke emissions.

The best equivalence ratio limit must ensure a good trade off between the exhaust equivalence ratio peak reduction and the engine torque response. Figure 13 displays the IMEP set points computed with several values of AFR limits performed with the fixed step simulator. The smoke limitation strategy acts as a filter on the raw IMEP set point. As a consequence the fuel mass will be injected with a delay depending on the equivalence ratio limit. Here we choose to limit fuel mass in order to operate near to the stoichiometry

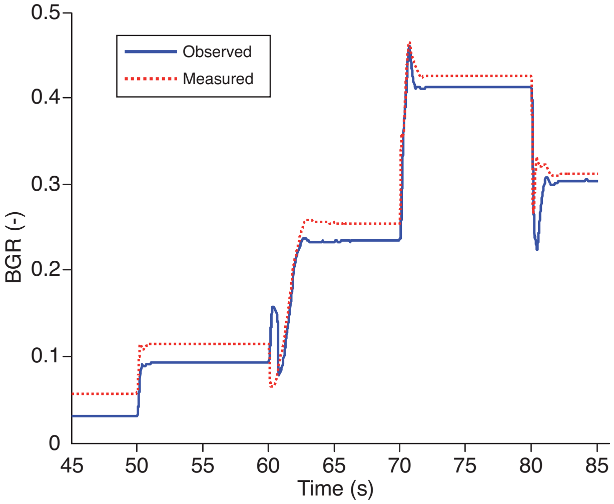

Figure 11

Comparison between BGR observed (bold) and measured on the reference simulator (dotted).

(AFR = 1), this choice leads to a good trade off. For a more stringent limitation, the equivalence ratio peak occurring during the transient can be avoided but the torque response is too slow and may affect the engine agreement.

Figure 14 displays the same kind of test results but performed at the bench. These tests are performed during a torque transient (from 3 bar to 13 bar of IMEP) at constant engine speed (1500 rpm). The top figure shows the IMEP responses which are clearly the same in both cases (with and without smoke limitation strategy). The fuel adaptation to mass air flow dynamics has no effect on the IMEP settling time. The two other figures show the deviation in opacity and exhaust equivalent ratio. The proposed strategy is able to limit the equivalent ratio. As expected, the opacity is reduced accordingly. It should be quoted that the experimental exhaust equivalence ratio agrees with the simulated results obtained during the smoke limiter tuning phase *(Fig. 13)*.

3.2.2 Simulation-Based Control Strategy Validation

The fixed step simulator allows to validate functionally all the parts of the engine control strategies. In the case of the vehicle control part, first calibration values can be tested for the idle speed control, start up and injection cut-off, giving some confidence for the first tests on the real system. The following figures show the most important features of the control strategies tested on a fixed step simulator during the first part of an ECE driving cycle: set points and measurements for BGR and air mass flow, fuel masses of the different injection pulses (pilot, main and

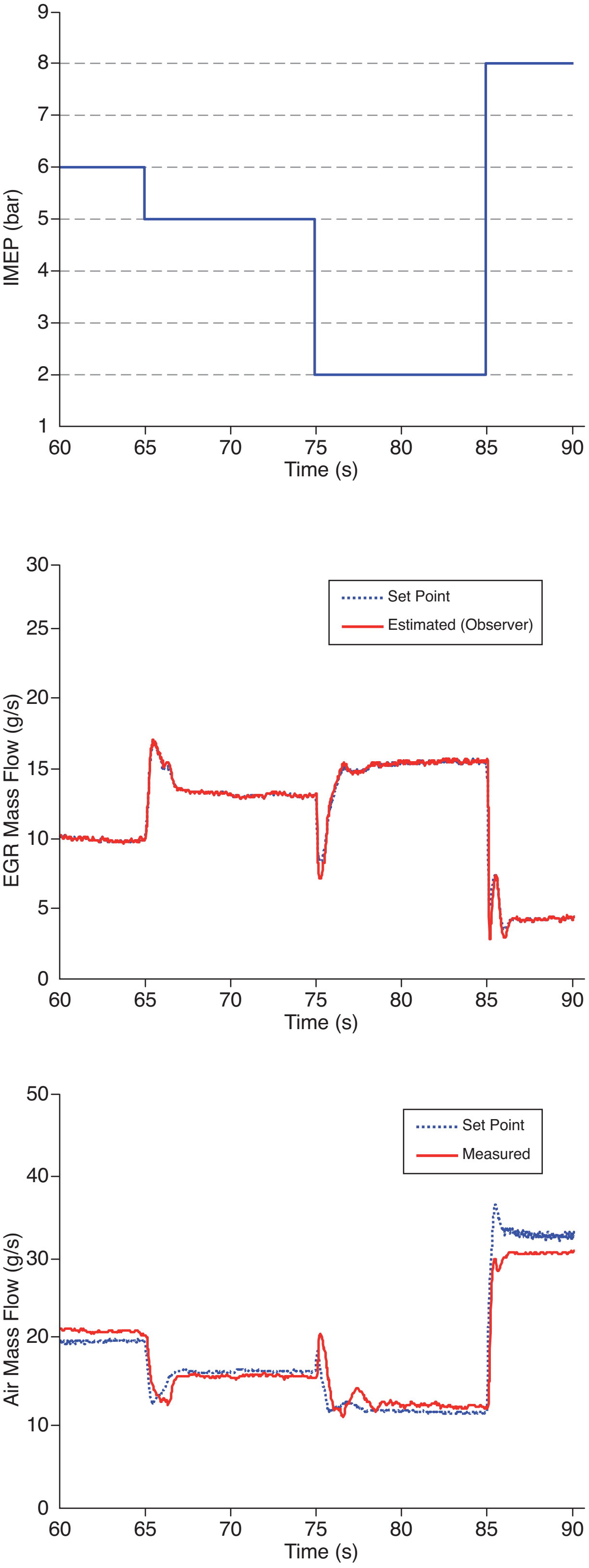

Figure 12

Experimental engine test results at 1500 rpm: IMEP set points (top), EGR flow (middle) and air mass flow (bottom).

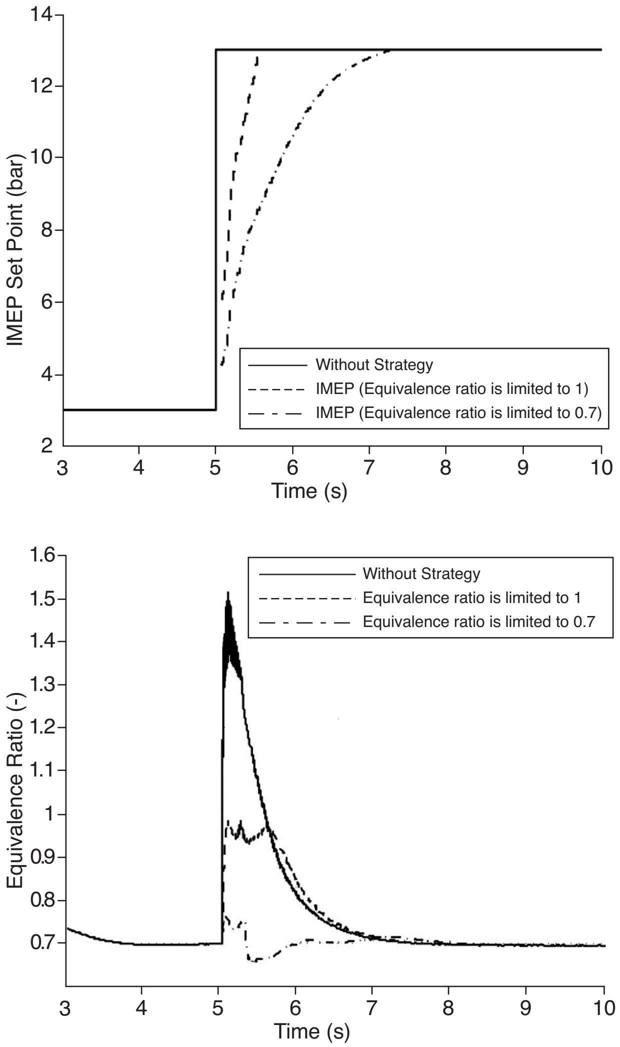

Figure 13

Comparisons of the IMEP set points (top) and exhaust equivalence ratio (bottom) with and without smoke limitation strategy. Simulations are performed with the fixed step simulator.

post injections), and the exhaust equivalence ratio. Such tests prove to be very helpful to validate the injection modes switching strategies used in HCCI or Diesel operating conditions.

3.3 Real Time Simulator for Control Pre-Calibration

The real time simulator is the last phase of the simulator development. The dedicated platform for this simulator is the hardware-in-the-loop platform which is designed to test and validate the complete engine control implemented in a

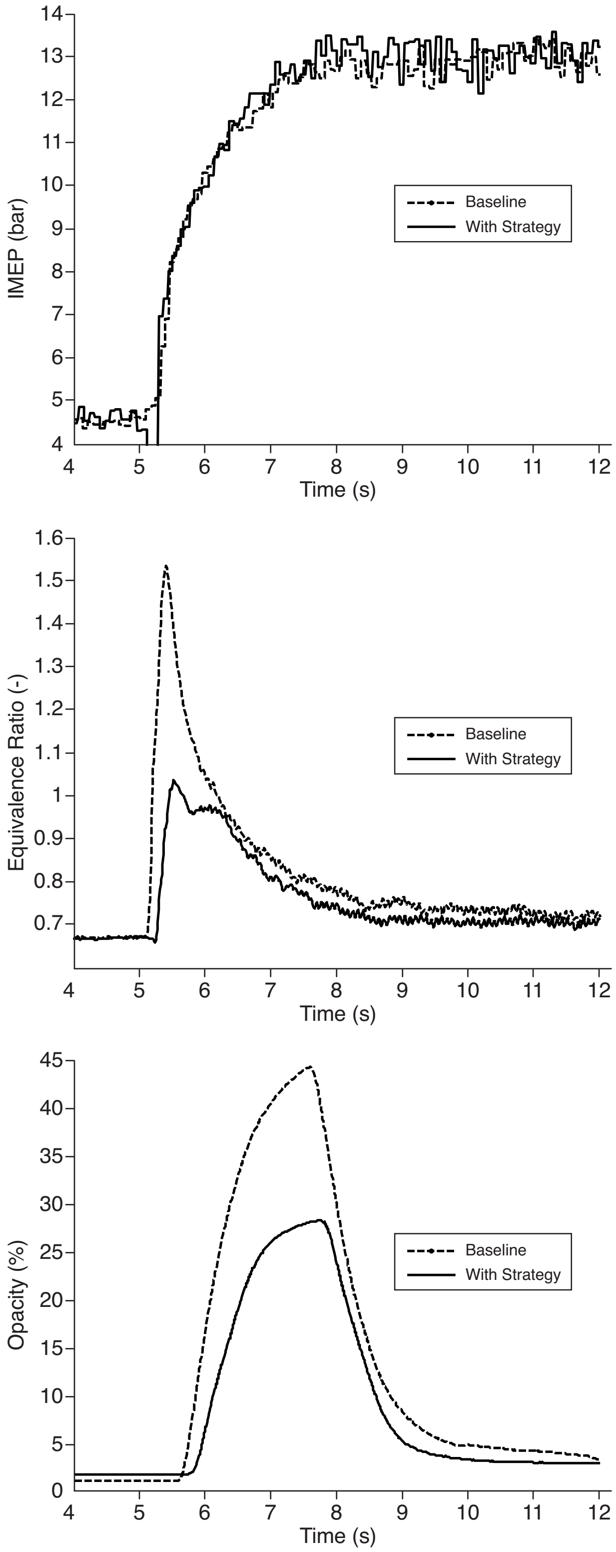

Figure 14

Smoke limiter experimental results: IMEP (top), equivalence ratio (middle) and opacity (bottom). The dotted lines represent the engine behaviour without the smoke limitation strategy and the solid lines exhibit the engine response with the strategy.

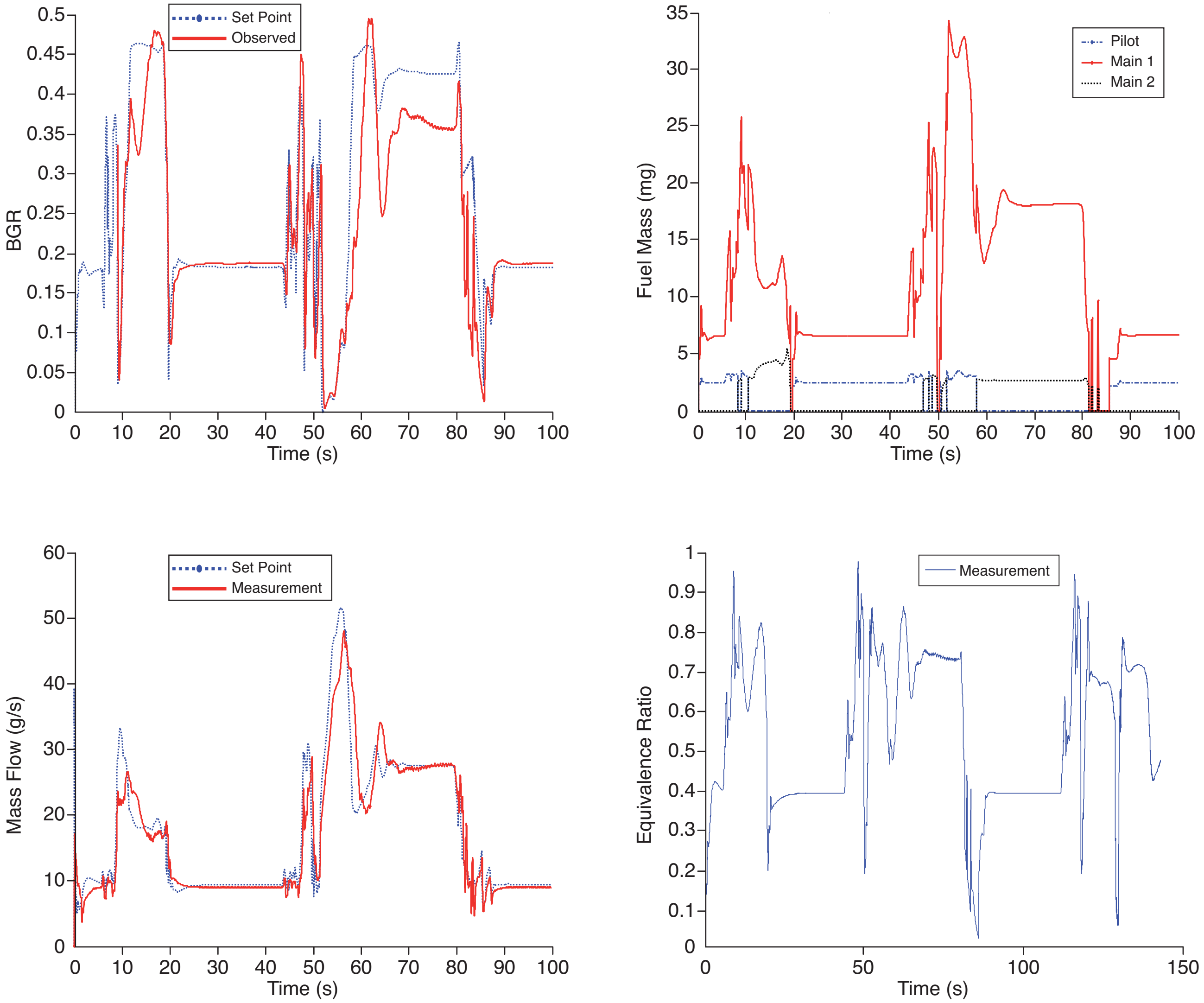

Figure 15

Test results during the first phase of the ECE driving cycle using the fixed step simulator.

Simulink-based framework. The purpose of this HiL platform is not to reproduce all signal conditioning to be compliant with a standard ECU wires, but to be fully plug-and-play with the engine control coming from the testbed or the vehicle to the engineer's desktop. Real-time exchanges with this engine control are ensured with a dual-port shared memory connected between the HiL platform and the engine control. This dual-port is synchronized with time events and engine events, TDC and 6°CA interruptions. Synchronized to these events, all inputs/outputs are exchanged from the engine simulator to the engine control. To avoid any modification of the engine control in Simulink, all drivers for inputs/outputs used at the testbed are overloaded to exchange, in the same conditions, data through the dual-port memory board.

It allows to test the strategies in a real time environment, and even with some real hardware. In addition, the real time simulator can also allow to test slowest control strategies, consisting mainly in the state machines responsible of the transitions between the different operating modes (or warm up).

3.3.1 Simulation-Based Full Control Pre-Calibration

The real time simulator is a very flexible tool to simulate and to pre-calibrate the engine control strategies over longer

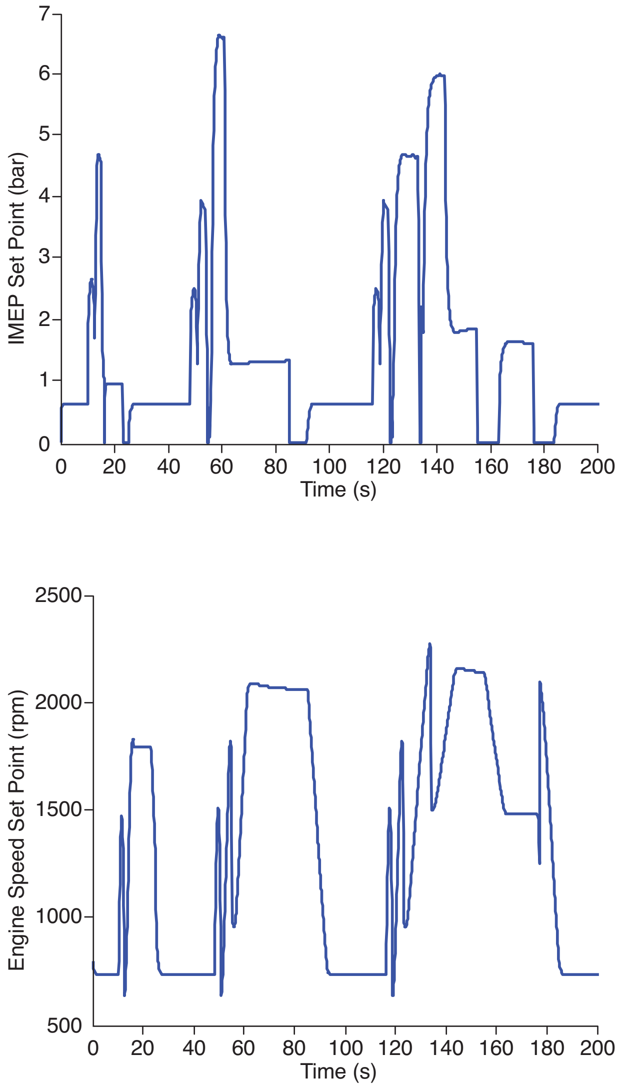

Figure 16

ECE cycle: IMEP and engine speed set points for a Renault VelSatis vehicle.

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transient tests. Here, we choose to evaluate the potential of the smoke limitation strategy during the urban ECE driving cycle with the corresponding torque and engine speed profiles based on a RENAULT VelSatis vehicle *(Fig. 16)*. This transient test lasts 195 s.

We evaluate the validity of the air loop control by observing the EGR and air mass flows in first. The fuel loop controller (including the smoke limiter system) is validated by observing the exhaust equivalence ratio which is strongly correlated with the opacity. The simulated results are shown in Figure 17 and some conclusions can be drawn.

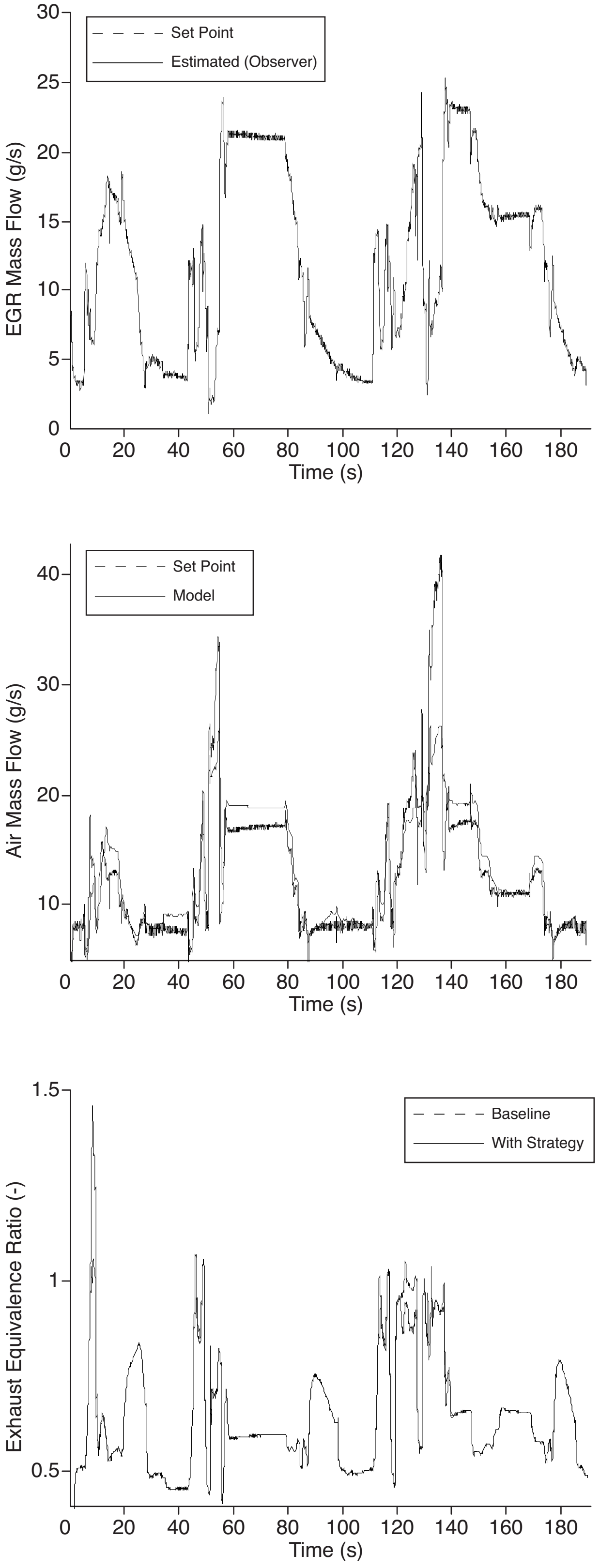

Figure 17

HiL simulation results on the ECE driving cycle: EGR mass flow (top), air mass flow (middle) and exhaust equivalence ratio (bottom).

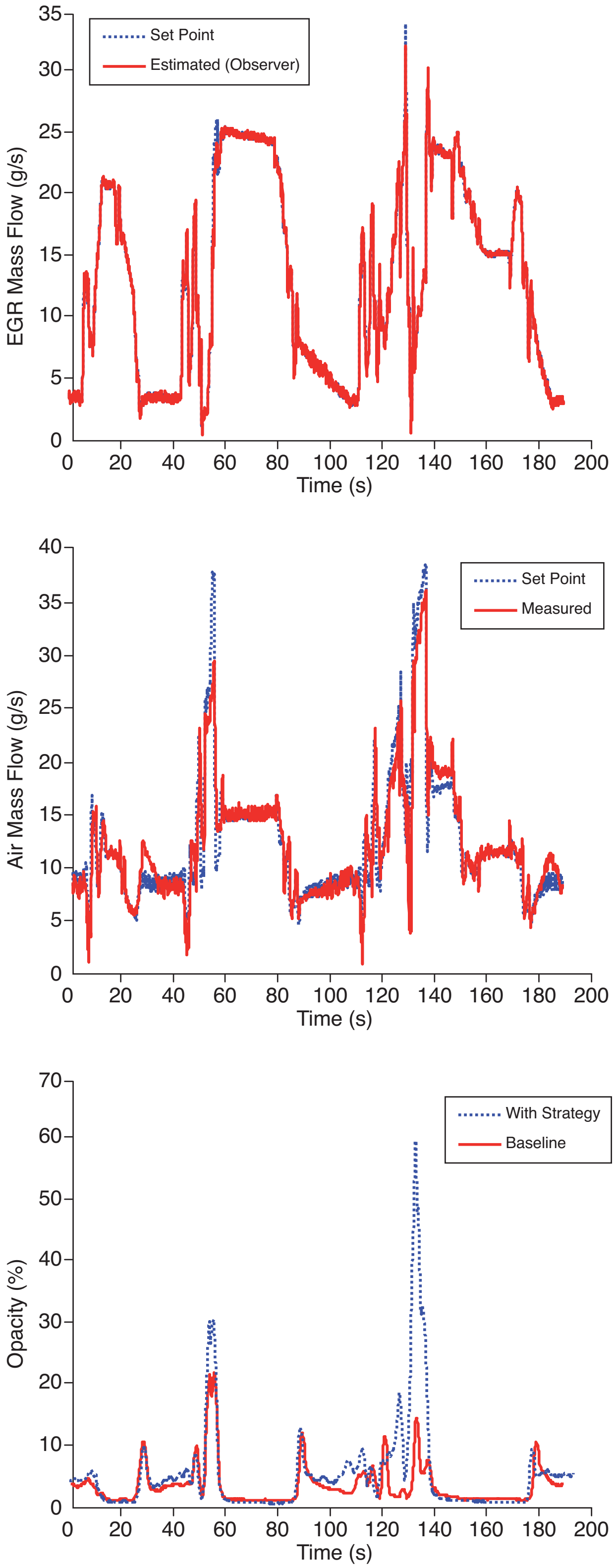

Figure 18

Experimental engine results on ECE driving cycle: EGR mass flow (top), air mass flow (middle) and opacity (bottom).

The EGR control is fast and accurate, the actual controller does not need any other improvement to track the EGR flow set point (the set point and the estimated values are superimposed).

The air flow control is slower and there is a static error during large IMEP transients. This point requires some improvements.

The exhaust equivalence ratio agrees with expected results since its maximum value remains under the settled limit (equivalence ratio which is lower than one).

The global engine behaviour is correct and the engine controllers developed can now be tested on the engine bench.

3.3.2 Validation on Bench

Figure 18 shows results measured on the engine test bench on the same ECE driving cycle as in Section 3.3.1. On the top and middle figures, the control of the EGR and fresh air mass flow are presented, with good performances in the EGR control while the fresh air control remains slower as noticed in Section 1. The bottom figure shows the comparison of the exhaust gas opacity measurement between a test without any specific dynamic strategy on the fuel injection, and a test with the developed strategy. The smoke emissions are reduced during the transient operations.

CONCLUSION

The engine simulation based control design methodology has been presented and illustrated with the case of the NADI™ HCCI engine. The HCCI engine simulator design and the way the three simulator versions have been used for key purposes of HCCI control such as BGR estimation or injection strategy switching have been described. As a matter of fact, it has been demonstrated that the engine simulator has been used at each step of the control development process, from observer/controller design until the real time pre-calibration, since it has allowed performing off-line a significant part of the design work in a flexible and low cost environment. Today, the engine simulation is growing up in engine technological project, especially for control issues, with the constant improvement of this modelling approach and its CPU performance. Mainly focused on engine control, the proposed approach is also applied to others automotive control related issues such as after-treatment management or vehicle calibration.

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REFERENCES

1 Ranini A., Potteau S., Gatellier B. (2004) New developments of the NAD™ concept to improve operating range, exhaust emissions and noise, *Proceedings of the THIESEL 2004 Conference*, Valencia, Spain, 2004.

2 Guzzella L., Amstutz A. (1998) Control of Diesel engines, *IEEE Contr. Syst.* **18**, 2, 53-71.

3 Jankovic M. (2001) Control design for a Diesel engine model with time delay, *IEEE Conference on Decision and Control*.

4 Jankovic M., Jankovic M., Kolmanovsky I. (1998) Constructive Lyapunov control design for turbocharged diesel engine, *Proceedings of the American control Conference,* Philadelphia, 1998, pp. 1389-1394.

5 Stefanopoulou A.G., Kolmanovsky I., Freudenberg J.S. (1998) Control of variable geometry turbocharger Diesel engines for reduced emissions, *American Control Conference*, Philadelphia, USA, 1998.

6 Albrecht A., Le Berr F. (2008) Powertrain simulation: an overview of the use for engine development, *FISITA 2008*, Munich, Germany, submitted.

7 Albrecht A., Grondin O., Le Berr F., Le Solliec G. (2007) Towards a stronger simulation support for engine control design: a methodological point of view, *Oil Gas Sci. Technol.* **62**, 4, http://ogst.ifp.fr/.

8 Ménégazzi P., Aubret Ph., Badin F., Triguy R., Marchand C. (2006) An advanced simulation tool for consumption, emissions, and performance analysis of conventional and hybrid vehicles, FISITA paper F2006P115, 2006.

9 Albrecht A., Chauvin J., Lafossas F.A., Potteau S., Corde G. (2006) Development of highly premixed combustion Diesel engine model: from simulation to control design, SAE paper 2006-01-1072.

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