*Oil & Gas Science and Technology – Rev. IFP Energies nouvelles*, Vol. 66 (2011), No. 5, pp. 855-875

DOI: 10.2516/ogst/2011136

Dossier

*R&D for Cleaner and Fuel Efficient Engines and Vehicles*
R&D pour des véhicules et moteurs plus propres et économes

# Use of Ethanol/Diesel Blend and Advanced Calibration Methods to Satisfy Euro 5 Emission Standards without DPF

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**Résumé — Utilisation d'un carburant Diesel éthanolé à l'aide de méthodes de calibration avancées afin de satisfaire les normes Euro 5 sans filtre à particules** — L'utilisation des biocarburants s'est développée durant ces dernières années de façon importante afin de diversifier les sources d'énergies et de limiter la hausse des émissions de gaz à effet de serre du secteur des transports. L'un des carburants renouvelables les plus adaptés à une production de masse est l'éthanol. Celui-ci est aujourd'hui principalement utilisé dans les moteurs à allumage commandé, alors que la part des véhicules Diesel sur le marché européen est de l'ordre de 60 %. Ce constat nous a incité à proposer une formulation innovante utilisant de l'éthanol pour les applications Diesel.

Les principaux verrous technologiques pour cette utilisation sont la miscibilité, la température d'éclair, la lubrification ou encore l'indice de cétane. Des travaux ont été réalisés pour optimiser la formulation contenant de l'éthanol, des biodiesels de première et seconde générations. Appliqué à un véhicule optimisé pour la norme Euro 4, ce carburant oxygéné a permis une diminution drastique des émissions de particules. Il a été cependant observé que les émissions de HC et de CO peuvent être problématiques tout autant que celles de $NO_x$ si les paramètres du contrôle moteur ne sont pas adaptés. Des analyses de combustion apportent une aide à la compréhension de ces résultats par l'étude du délai d'auto-inflammation ou encore de la combustion de l'injection pilote.

Ainsi, au banc moteur, des méthodes avancées de calibrations telles que les plans d'expériences ont été utilisées pour optimiser l'adéquation moteur/carburant. Une approche de type global a été proposée et comparée à une approche locale plus classique. Cette optimisation à chaud permet une réduction simultanée des émissions de $NO_x$ et de particules. Ces résultats laissent également présager une réduction du bruit de combustion et des émissions de $CO_2$ sur véhicule.

Un banc moteur dynamique a été ensuite utilisé pour étudier plus en détail le potentiel de réduction des émissions polluantes. Le comportement d'un véhicule sur le cycle européen NEDC avec démarrage à froid a pu être reproduit. Différentes méthodologies innovantes, basées notamment sur l'optimisation simultanée des cartographies de base et des cartographies de correction à froid, sont présentées. Elles permettent d'adapter les réglages du moteur par rapport à l'impact que ce carburant alternatif peut avoir sur la combustion et l'activation du catalyseur ou les émissions de HC et CO à la source. Ces méthodes ont permis de limiter le travail sur véhicule au banc à rouleaux à la validation des résultats obtenus sur banc moteur et à l'optimisation finale des cartographies à froid.

Cette étude a prouvé qu'il était possible de satisfaire les normes Euro 5 avec cette nouvelle formulation de carburant tout en conservant une configuration de véhicule Euro 4 sans filtre à particules. En définitive, ce carburant innovant apparaît comme une solution intéressante pour répondre au compromis entre la réduction des émissions polluantes et le coût des futures technologies dans le cadre du développement durable. En outre, les méthodes et outils de calibration présentés permettent d'adapter le couple moteur/carburant dans un temps réduit.

***Abstract — Use of Ethanol/Diesel Blend and Advanced Calibration Methods to Satisfy Euro 5 Emission Standards without DPF*** — *The use of biofuels has been extensively developed in the last years to diversify energy resources and to participate to the transportation greenhouse gas emissions reduction effort. One of the most promising renewable fuels for large scale production is the ethanol which is nowadays mainly used for spark-ignited engines; nonetheless the European market share of Diesel vehicles is around 60%. These issues lead us to propose an innovative fuel formulation using ethanol for Diesel engine applications.*

*The key issues to deal with the use of ethanol in a Diesel blend are the miscibility, the flashpoint, the lubricity and the cetane number. An intensive work has been done to optimise the formulation coupling the use of ethanol, with first and second generations of Diesel biofuels. The application on a Euro 4-compliant Diesel turbocharged engine with high pressure exhaust gas recirculation shows an outstanding decrease of particulate matter emissions thanks to this oxygenated fuel. Nevertheless unburned hydrocarbons and carbon monoxide emissions could be an issue as well as* $NO_x$ *emissions if the engine control settings are not updated. Combustion analysis helps understanding the fuel effect on the resulting auto-ignition delay and the pilot injection combustion behaviour, which leads to modified engine output compared to Diesel fuel.*

*Therefore, the optimisation of the fuel/engine matching is performed using advanced calibration methodologies combined with design of experiments at the engine test bed. First of all, global and mixed approaches are proposed and compared in warm operating conditions. Finally it permits to simultaneously drop nitrogen oxides emissions and particulate matter emissions. Global* $CO_2$ *emissions reduction and noise decrease are also expected.*

*To further investigate engine emissions potential reduction, the engine is set up on a dynamic test bed facility, allowing to reproduce cold New European Driving Cycle (NEDC). Several innovative calibration methods, based on the simultaneous optimisation of engine basic settings and cold correction maps, are introduced in order to better suit to the new formulation impact on combustion and catalyst light-off and to drop off engine-out unburned hydrocarbons and carbon monoxide emissions. This stage allows pushing forward the work on test bed facilities in order to reduce the amount of vehicle tests. Tests on a chassis dynamometer are only used to validate the engine test bed results and to perform final tuning of cold correction maps.*

*This alternative blend shows potential to achieve Euro 5 standard with Euro 4 Diesel vehicle configuration, without any hardware modification and without a Diesel particulate filter in the exhaust line. Such an innovative fuel formulation seems to be an interesting answer to the trade-off in the forthcoming years between cost and emissions reduction to achieve sustainable mobility. The presented calibration methods and tools allow to fully take advantage of this alternative fuel in a reduced time scale.*

## ABBREVIATIONS

| | |
|---|---|
| BHT | Butylated HydroxyToluene |
| BSFC | Brake Specific Fuel Consumption |
| BtL | Biomass to Liquid, biodiesel from the second generation |
| CAxx | Crank Angle for which xx% of injected fraction of fuel has been burnt |
| CN | Cetane Number |
| CO | Carbon monoxide |
| DOC | Diesel Oxidation Catalyst |
| DoE | Design of Experiments |
| DPF | Diesel Particulate Filter |
| ECU | Engine Control Unit |
| EGR | Exhaust Gas Recirculation |
| EN590 | Fuel within European specification limits |
| FAME | Fatty Acid Methyl Ester |
| FT | Fischer-Tropsch |
| HC | Unburned total HydroCarbon |
| HCCI | Homogeneous Charge Compression Ignition |
| HVO | Hydrogenated Vegetable Oil |
| LHV | Low Heating Value |

| | |
|---|---|
| NEDC | New European Driving Cycle |
| $NO_x$ | Nitrogen Oxide |
| OP | Operating Point |
| OS | Operating Space |
| PM | Particulate Matter |
| PME | Pression Moyenne Effective |
| $R^2$ | Coefficient of determination (statistics) |
| RMSE | Root Mean Squared Error |
| SoI | Start of Injection |
| Ti | Injection duration |

## INTRODUCTION

Some of the major challenges for the automotive industry are the reduction of the greenhouse gas emissions, the fossil fuel dependency and the local pollution. The use of renewable fuels is one of the promising ways to answer these issues. Among the various kinds of biofuels, ethanol has a good potential due to its availability in large volume, especially with the second generation process that will be available in the near future. A focus on the European market, shows the large proportion of Diesel cars leading to an imbalance between Diesel and gasoline fuels requirements.

To address these questions, the use of ethanol in a Diesel fuel is considered. To be relevant this approach has to give the possibility to reach high percentage of ethanol incorporation, to avoid any important engine technical modifications and to make the implementation with ease. Moreover, the addition of ethanol to Diesel fuel affects some key properties such as blend stability, Cetane Number or flash point. Consequently, a fuel formulation was developed achieving the incorporation of 20% and even up to 30% of ethanol. Eventually the formulation was improved to fulfill above requirements leading to an innovative ethanol based Diesel fuel formulation. A previous study confirmed the potential of several ethanol based Diesel fuels on both conventional and Homogeneous Charge Compression Ignition (HCCI) Diesel combustion [1].

The biofuels could also have a significant effect on the regulated emissions and on the fuel consumption [2]. One of the key outcome when using oxygenated blends is the large decrease of the Particulate Matter (PM); this could be beneficial to improve the $NO_x$/PM trade-off and to meet future standards. Hence the engine control parameters could be optimised to enhance the engine/fuel matching. The calibration process of these high numbers of parameters, that could be a time-consuming task, requires advanced methods and tools [3].

Therefore after studying the effects of the new ethanol Diesel blend on the combustion, this paper presents the different approaches to set up the engine variables and maps in warm and cold conditions. The goal is to meet Euro 5 emission regulations without modifying the hardware of a Euro-4 compliant vehicle, which means without adding a Diesel Particulate Filter (DPF).

## 1 EXPERIMENTAL SET UP

### 1.1 Engine Test Bed and Vehicle Description

The engine used during this study at the test bed and into the vehicle is a 1.6 litre four-cylinder Diesel engine (DV6 ATED 4 from PSA Peugeot-Citroën). This engine has a maximum power output of 66 kW. The bore and the stroke of this engine are respectively of 75 mm and 88.3 mm. The compression ratio is 18:1. It is a turbocharged engine with a high pressure cooled exhaust gas recycling circuit. It has a second generation injection system using Bosch CRI 2.2 injectors with injection pressure as high as 1 600 bar. The exhaust line has a conventional Euro4 Diesel Oxidation Catalyst (DOC) and is not equipped with a Diesel Particulate Filter (DPF).

A dynamic test bed is used to perform transient operations (New European Driving Cycle (NEDC)) as well as steady-state operations and to reproduce the vehicle behaviour. The main issue is to be representative of the vehicle concerning the engine speed and load tracking and also regarding the thermal behaviour (for example coolant temperature). Thus the cycle reproduction at test bed uses engine in the loop methodology. The only hardware elements are the engine and its Engine Control Unit. The driver behaviour, the road profile and the vehicle parameters are taken into account through real time simulation. The simulated driver manages the engine operation, via the accelerator pedal provided by the engine control. It interacts with the simulated vehicle, via the clutch pedal, the brake and the gear box position. The vehicle model, based on the actions of the driver, is able to calculate the engine speed setpoint applied by dynamometer machine. Finally, the simulated vehicle speed allows the driver to adjust the engine load to follow a defined speed setpoint.

Concerning the temperature tracking, an optimisation of the cooling circuit and its volume was carried out to perform a similar warming behaviour from 20°C until 90°C to the vehicle *(Fig. 1)*. Added pumps and heat exchangers enable to achieve fast cooling and soak phase between two cold cycles (it takes from 2 to 3 hours instead of waiting at least 8 hours between two tests).

The test bed is equipped with specific measurements devices to suit to transient requirements for example concerning fuel consumption (AVL KMA 4000) and opacity (AVL 439). Gas emissions and opacity could be measured upstream and downstream DOC.

The engine is also equipped with cylinder pressure transducers to provide data for heat release analysis and combustion noise calculation.

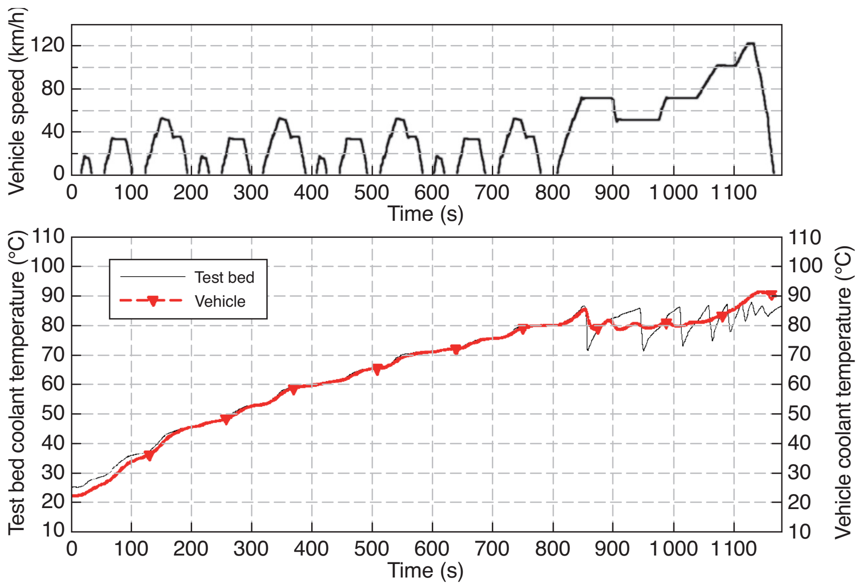


Figure 1

Coolant temperatures comparison between engine test bed and vehicle.

A Citroën C4 without any hardware modification and with a similar engine and initial calibration was tested on a chassis dynamometer. The vehicle inertia is 1 360 kg. It is a Euro 4-compliant vehicle.

## 1.2 Fuel Formulation and Properties

The addition of ethanol to Diesel fuel affects some key properties such as blend stability, Cetane Number (CN) and flash point as well as viscosity and lubricity or energy content.

Indeed, ethanol is not miscible in a Diesel fuel over a wide range of temperatures. In this study, a Fatty Acid Methyl Ester (FAME), another renewable fuel, is successfully used as an amphiphile (a surface-active agent) to stabilize ethanol and Diesel together. The Biodiesel is a Rapeseed Methyl Ester (RME), which is additivated with 1 000 ppm (mg.kg$^{-1}$) of anti-oxidant (BHT Butylated hydroxytoluene) in order to improve its oxidation stability. The addition of ethanol also lowers fuel lubricity, that can affect significantly the lubrication process of the fuel injection system. This low lubricity is compensated by the high lubricity of FAME.

Moreover, ethanol has poor auto ignition properties, which lead to a low Cetane Number of the blend. Consequently, in order to partly compensate the auto-ignition property, the diesel fuel from fossil origin has been substituted by a Fischer-Tropsch (FT) fuel well-know for its high propensity to auto- ignite (high Cetane Number). For this study, the FT fuel is a GtL (Gas to Liquid), as no biofuel from the second generation BtL (Biomass to Liquid) was available. But, in the future, a formulation with ethanol, FAME and BtL or Hydrogenated Vegetable Oil (HVO) could be considered in order to have a “100%” biofuel formulation.

Furthermore, Diesel blends containing ethanol are highly flammable with a flash point temperature around ambient temperature (about 13°C). The flash point of the fuel affects shipping and storage classification of fuels and precautions that should be used when handling. Solutions to solve this trouble could be vehicle hardware modifications (flame arrestors in the filler neck or dry lock) or fuel formulation improvements. This latter solution has been chosen here. The principle is to add a volatile hydrocarbon to the fuel blend, in order to produce an over-rich fuel/air mixture in the empty space that will not ignite if exposed to a potential ignition source. Several products have been selected as potential flash point improvers. Main advantages and drawbacks of many liquid hydrocarbons were analysed and, in a first time, the iso-pentane was chosen.

In a previous work, an innovative ethanol based diesel fuel, formulated with 20%v/v of ethanol, FAME, Fischer-Tropsch and iso-pentane has been tested in single-cylinder and multi-cylinder engines [1]. The main advantage of this innovative ethanol based Diesel fuel was to generate low smoke level. However, the iso-pentane, used to improve the flash point of the fuel, has lead to some phenomenon of vapour lock during warm starts. For this work, the iso-hexane has been preferred in order to solve the flash point and to limit vapour lock issues. Even if its volatility is less important than iso-pentane, iso-hexane is up to now, the best compromise between flash point improvement and vapour lock propensity. The flash point is thus decreased at –8°C *(Tab. 1)*.

TABLE 1

Main properties of base fuels, reference fuel B0 and the innovative 20% ethanol based Diesel fuel. Values in italic are estimated or from literature

| | | | Ethanol | FT | RME | Iso-hexane | Reference fuel B0 | Innovative ethanol based Diesel fuel |
|---|---|---|---|---|---|---|---|---|
| Flash point | °C | EN ISO2719 | *~ 13* | 89 | 175 | *-32* | 67 | -8 |
| Cetane Number | | EN ISO5165/ EN51595 | *~ 8* | 82.5 | 56.0 | *28* | 53.5 | 36.6 |
| Density at 15°C | $kg.m^{-3}$ | EN ISO12185 | 794.4 | 770.2 | 883.2 | *~ 660* | 834.0 | 809.6 |
| Carbon content | %m/m | DIN 51732 | 52.2 | 84.8 | 81.0 | 83.7 | 86.4 | 70.1 |
| Hydrogen content | %m/m | | 13.0 | 15.4 | 11.9 | 16.3 | 13.5 | 12.3 |
| Oxygen content | %m/m | | 34.7 | - | 7.1 | - | < 0.2 | 9.5 |
| LHV | MJ/kg | ASTM D240 | ~26.8 | ~44.1 | ~37.3 | unknown | 43.0 | 36.1 |
| Energy content | w/w% | | - | - | - | - | - | -15.9 |

Consequently, for this study, the final formulation of the tested innovative ethanol based diesel fuel is based on 20%v/v of ethanol, RME, FT and some percents of iso-hexane. This innovative ethanol based diesel fuel is compared to a reference diesel fuel (B0) without any biofuel and complying with the EN590 specification.

Table 1 summarises the main properties of the reference fuel B0 and the innovative ethanol based Diesel fuel. The Cetane Number is measured following the EN ISO 5165 method (CFR engine) or the IQT method (EN51595). The Low Heating Value (LHV) is obtained following the ASTM D240 method. The energy content is calculated from tested fuel LHV and density, and compared to the reference fuel.

The use of ethanol and FT fuel induces a decrease of the density; which is partially compensated by the FAME. Consequently, the density of the innovative fuel is equal to 809.6 $kg.m^{-3}$, compared to 834.0 $kg.m^{-3}$ for the reference fuel. Furthermore, by using ethanol and FAME, the gross heat content decreases. This trend should be softened by the Fischer-Tropsch base fuel and by the iso-hexane, thanks to their better lower heating values (mass) of such products. Finally, that leads to a lower energy content compare to the reference fuel (– 15.9w/w%).

Concerning the ignitability on the innovative fuel, despite the use of FT fuel, the Cetane Number remains low: 36.6. It is lower than expected by a linear calculation. This low Cetane Number induces an increase of the auto-ignition delay.

## 2 ADVANCED CALIBRATION METHOD DESCRIPTION

### 2.1 Context

Since the increasingly stringent Diesel emissions legislation and the use of sophisticated common rail turbocharged engines to fulfill it, the number of engine control parameters has drastically increased. Consequently, Design of Experiments (DoE) methods are now usually used for the engine calibration process in order to reduce cost and turnaround time. Several approaches can be used for this purpose. The local approach is commonly used in the automotive industry. This approach is briefly described in this paper. To ensure a good understanding of the following section, it is recommended to read previous References [3, 4].

This section presents some alternatives methods to the local approach with two different advanced calibration methods. Both approaches were used to calibrate a vehicle running an ethanol/Diesel blend described in the previous section. The methods and the results are presented in this paper, and they are also compared to the local approach results.

The local approach consists in cutting the driving cycle into Operating Points (OP) defined by engine speed and a load. Then it aims at optimising, for each OP, the engine settings using local engine models issued from DoE. As soon as the optimal settings are obtained for each Operating Point, a smoothing stage takes place in order to obtain final maps. The synopsis of the method is presented in Appendix.

One of the main advantages of this method consists in its simplicity to be implemented. Indeed, each step of the workflow is well controlled and the analysis of the results is easy. However, this method has several significant drawbacks. Firstly, the definition of the emission target for each OP is very tricky. Then the map smoothing step is time consuming, and worsens the benefits of the optimisation by changing the optimised settings in order to keep smooth maps. Besides, the local approach considers each Operating Point independently, while the goal of calibration is to fulfil the standards for pollutants emissions, *e.g.* to minimize the cumulated emissions over a cycle. At last, when a large number of applications should be tuned, the global duration of such an approach is long since each configuration is tuned independently. The following mixed and global approaches avoid the main drawbacks of the local one. These methods are described in the following sections, while the main results are shown in Section 3.

## 2.2 Mixed Approach

### 2.2.1 Global Description

As stated previously, the mixed approach avoids the local approach drawbacks while keeping the main advantages. Indeed, the workflow is similar until the optimisation step. The synopsis of the method is presented in Appendix. The difficult step of the definition of the targets for each OP is removed (they are substituted by global targets on cumulated responses), whereas the workflow to obtain the local models is maintained. Therefore only the optimisation step will be described in this section.

### 2.2.2 Optimisation

The optimisation process consists in directly optimising the cumulated engine responses over the cycle. For this purpose, distortions of the engine maps are generated instead of optimising individually the selected OP and building afterwards the engine maps by the smoothing step [3]. In case of the mixed approach, the models of engine responses are available only for a limited number of representative Operating Points as in the local approach. Therefore the cumulated engine responses over the cycle are approximated by the weighted sums of the local models at the chosen representative OP. The mixed map optimisation problem is then formulated as follows:

$$\begin{cases} \min\limits_{m^P \in R^{N_p}} \sum\limits_{l=1}^{N_{OP}} w_l F_i^{(l)}[m^{p_1}(r_l,c_l), m^{p_2}(r_l,c_l), \ldots, m^{p_{N_p}}(r_l,c_l)] \\ \text{subject to} \\ l(r,c) \le Am^p(r,c) \le u(r,c) \\ \sum\limits_{l=1}^{N_{OP}} w_l F_j^{(l)}[m^{p_1}(r_l,c_l), m^{p_2}(r_l,c_l), \ldots, m^{p_{N_p}}(r_l,c_l)] \le S_j \end{cases} \tag{1}$$

where:
- $(r_l; c_l)$ indicate the chosen OP;
- $F_i^{(l)}$ is the local model of the engine response $i$ associated with the OP $l$;
- $m^{Pi}$ are the 2D engine maps of the control parameters in the engine (speed, load) operating domain.

The previous formulation requires an adapted parameterisation of the engine maps in order to limit the total number of unknowns in the optimisation process [3]. This parameterisation has to be also flexible enough to model the different shapes of engine map surfaces. LoLiMoT (Local Linear Mode Tree) models [3, 5, 6] seem to be a good compromise between flexibility, accuracy and complexity: some very simple local models (linear or bilinear) are combined using a weighted sum. These models were more deeply explained and an example of application of this method was presented in a previous reference [3].

In this calculation, different kinds of constraints can be introduced for the optimisation:
- global constraints on cumulated emissions;
- constraints on the parameters (limits of Operating Space);
- local constraints on the responses;
- smoothing constraints on the parameter maps;
- robustness constraints (to take into account parameter dispersions).

## 2.3 Global Approach

### 2.3.1 Global Description

Unlike the local and mixed approaches, the global method directly considers the trace of the standard driving cycle with speed and load as input of the models of the engine responses. Thus no OP have to be chosen, and the steps of OP weight calculation and target definition are removed. Figure 2 shows the speed and load trace, as well as the operating area considered for the calibration. The synopsis of the approach is presented in Appendix. The different stages of this method were further described in a previous Reference [3].

### 2.3.2 Operating Space Definition

The Operating Space (OS) determination becomes the crucial step of the calibration process since it will determine the potential of the optimisation. Indeed it was chosen to only use the models in the domain covered by experiments (to avoid extrapolation of the models). Thus the optimised settings have to be inside the domain defined during this first

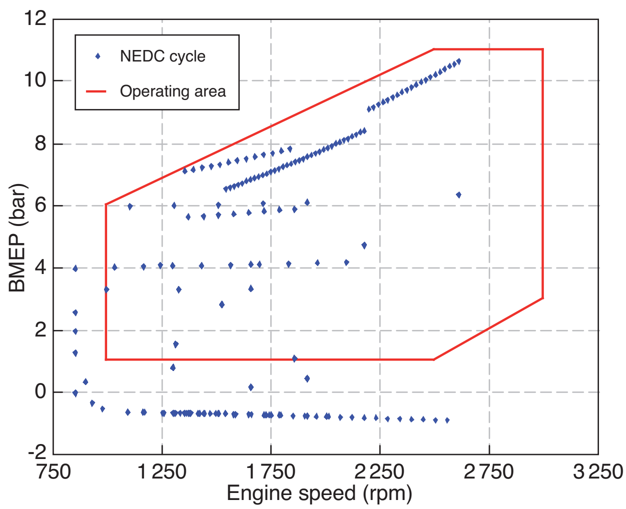


Figure 2

Speed and load trace of the application on a NEDC cycle and operating area.

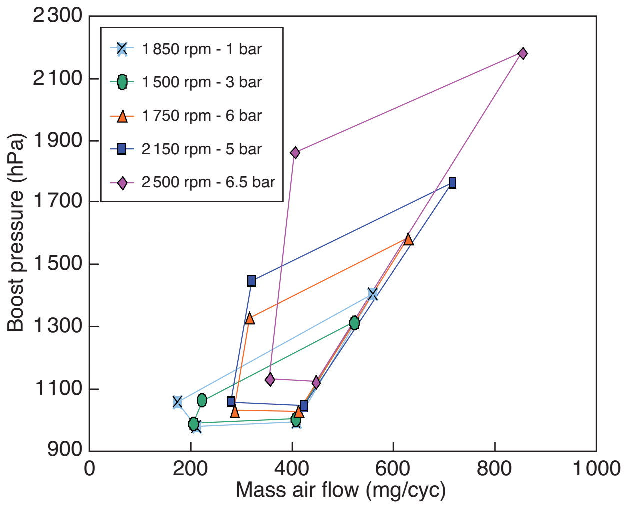


Figure 3

Evolution of air loop limits with engine speed and load.

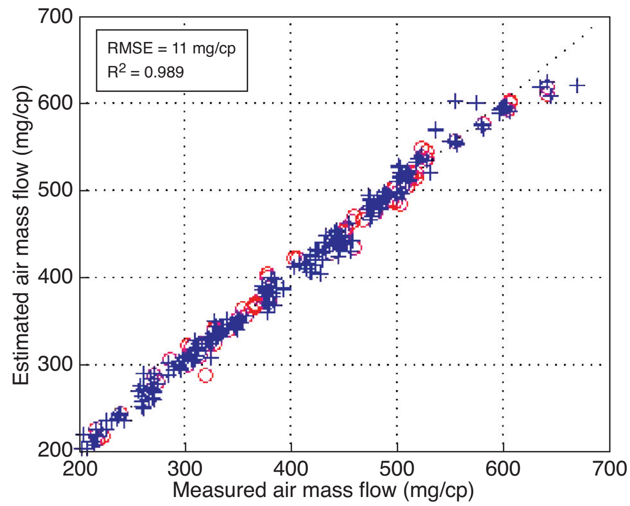


Figure 4

Correlation diagram for minimal air mass flow limit.

step. The goal is to define a wide Operating Space, close to the real physical domain. Hence the potential of optimisation is high, and the optimal settings will necessarily be inside the validity domain of the model. Furthermore, with a wide operating range, especially concerning the air loop parameters, it is possible to consider the optimisation with several applications (configurations of aftertreatment *e.g.* presence or not of $NO_x$ conversion device), without any additional test. Moreover, the step concerning the Operating Space determination has to be low time consuming, and the domain should be easy to model since it will be used for further calibration steps.

A pragmatic approach has been developed [7] to meet these goals. Regarding the injection parameters, some fixed boundaries are defined over the speed and load area to optimise. It is then possible to get lower and upper limit maps for each injection parameter, with a good engine behaviour ensured between these maps. Afterwards, the limits of air loop parameters have to be defined. These limits depend on the injection parameter values as well as speed and load, as shown in Figure 3.

There are four air loop parameter limits. The minimal and maximal boost pressure limits as well as the maximal air mass flow limit are the three physical limits (permeability, maximum cylinder pressure, etc.) that can be easily identified. The last one, the minimal air mass flow limit, is a subjective limit and has to be defined by a criterion (stability, AFR, etc.). An automatic test has been developed to define these domain constraints within a limited amount of time. The air loop limits are then modelled with low degree polynomials. Figure 4 presents correlation diagrams for the model of minimal air mass flow limit. The crosses represent experiences used for modelling, and the circles represent the one used for validation. Despite the fact that this limit is the most difficult to model, since it is not a physical limit, the results are very good ($R^2$, Root Mean Squared Error (RMSE)). The quality of the model depends on the relevance of the criterion used to define this limit. The results for the models of other boundaries (physical limits of the air loop) have equivalent or even better quality than this one.

### 2.3.3 Design of Experiments

For classic local approach, domains are basically hypercubic but some linear constraints can be added. Thus, it makes it easy to use standard algorithms (space filling, D-optimality, etc.) to design the experiments. However, in the case of global approach, the number of parameters increases and more complex constraints (*e.g.* non-linear) can be encountered as shown in previous section. Thus, a specific tool for the design of experiments has been developed to take into account these constraints.

The proposed method to choose a feasible point consists in proceeding in successive steps. First of all, the limit in the 2D space speed and load has to be defined *(i.e. Fig. 2)*. Then, within this 2D domain, some speed and load values are chosen. It is then possible to calculate the maximum and the minimum of each injection parameter which are modelled with maps depending on speed and load. And once acceptable values for injection parameters have been chosen, the air loop limits which depend on the other parameters can be defined.

The method to choose an experimental point is quite similar to this process: firstly, a couple of values for speed and load

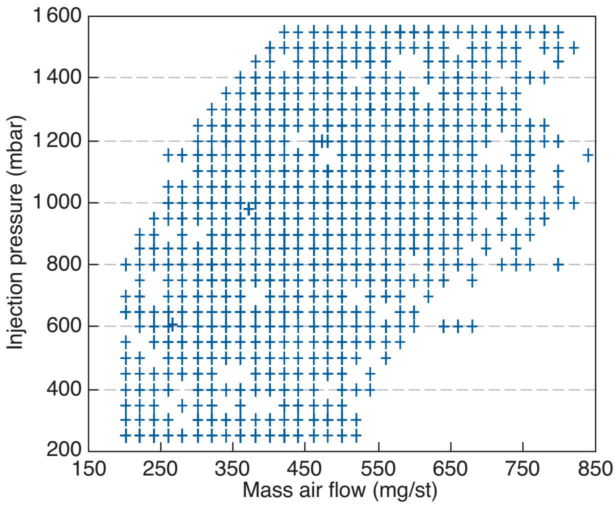


Figure 5

Global design of experiments in a 2D space projection.

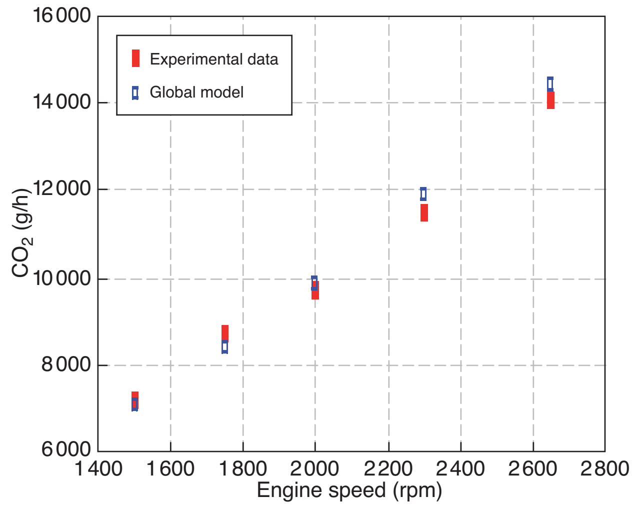


Figure 6

Comparison of measured and estimated $CO_2$ emissions value for engine speed variation.

is defined, and secondly, six other parameters are fixed within their acceptable range. A *Maximin* criterion is then used to choose the most valuable experience.

The final global design of experiments contains the corners, some monoparametrics variations as well as a space filling, added by running iteratively the steps described previously by random sampling. Finally, a total of 1 500 experiments were planned. 200 experiments were used only for validation and 100 points were used to determine the experimental repeatability. Figure 5 presents the 1 500 experiences shown in the 2D space Mass Air Flow (MAF)/Injection pressure. It is noteworthy that the 1 500 experiences (1 200 used to obtain the model) cover the entire Operating Space of a NEDC. They were performed in about 90 hours.

### 2.3.4 Modelling

Standard polynomial models are not sufficient for modelling the complex engine responses surfaces against control parameters in case of global approach. Advanced statistical modelling methods must be used for this purpose [8]. It was decided to use kriging modelling, a method for spatial interpolation based on the theory of Gaussian processes. Kriging takes into account the spatial dependence of data, which means that close points will have close predicted values. It is very flexible and allows the measurement errors to be taken into account. In our case, 100 experiences among the 1 500 are used to provide the experimental repeatability for each engine response to be modelled. A modelling tool has been developed which includes a specific algorithm to avoid overfitting. Validation data, which represents an important part (13%) of the total number of experiences, are used to check this point.

Several engine responses were modelled: HC, $NO_x$, CO, $CO_2$ and PM emissions (g/h), as well as combustion noise (dB) and fuel consumption (kg/h). All the models, except HC and PM emissions, provide very satisfying results with good statistical indicators. The poor quality of the HC emissions model is due to a bad experimental repeatability, which can be attributed either to the engine or to the measurement system. The poor quality of PM emissions model is related to the use of the innovative ethanol/Diesel fuel formulation. Indeed, soot emissions were too low to be accurately measured.

The modelling tool contains also many charts to validate the model quality in terms of accuracy and predictability. As explained in the previous section, the global DoE contains some monoparametric variations which are used to validate the models with a single trend. Figure 6 presents the comparison of the experimental results and the estimated value from the global model of $CO_2$ emissions (engine speed variation). The results are satisfying both qualitatively and quantitatively.

### 2.3.5 Optimisation

The final step of the calibration process is the optimisation.

The optimisation step of the global approach is very similar to the optimisation step of the mixed approach: similarly, this method consists in directly optimising the cumulated engine responses over the cycle via distortions of the engine maps. It allows the entire engine operating field to be directly optimised at once, in order to avoid a smoothing step.

Unlike the mixed approach, the engine responses are modelled by global models defined on the whole operating domain. Therefore, the cumulated engine responses over the cycle are directly the integrals of these global models allowing a more accurate optimisation over the cycle than the mixed approach.

An other interesting feature of this approach is the possibility to optimise different cycles or different application problems (different vehicles...) without any additional tests at the test bench. In this case, the whole engine operating area (for the cycle) of the different applications should be covered during the experimental tests.

The map optimisation problem is formulated as given below:

$$\begin{cases} \min\limits_{m^p \in R^{N_p}} \int_0^T F_i(r(t),c(t),m^{P_1}(r(t),c(t)),\, m^{P_2}(r(t),c(t)), \\ \qquad \ldots, m^{P_{Np}}(r(t),c(t)))dt \\ \text{subject to} \\ l(r,c) \le Am^p(r,c) \le u(r,c) \\ \int_0^T F_j(r(t),c(t),m^{P_1}(r(t),c(t)),\, m^{P_2}(r(t),c(t)), \\ \qquad \ldots, m^{P_{Np}}(r(t),c(t)))dt \le S_j \end{cases} \tag{2}$$

where:
- ($r(t)$; $c(t)$) indicate the trace of the driving cycle within the engine speed-load domain;
- $F_i$ is the global model of the engine response $i$ depending on the engine control parameters but also on the speed and load;
- $m^{Pi}$ are the 2D engine maps of the control parameters in the engine (speed, load) operating domain.

The same parameterisation of the maps by LoLiMoT functions is introduced in order to control the number of optimised parameters and the smoothness of the maps.

The different kinds of constraints that can be introduced in the optimisation are:
- global constraints on cumulated emissions;
- constraints on the parameters (limits of Operating Space);
- local constraints on the responses;
- smoothing constraints on the response maps or on the parameter maps;
- robustness constraints (to take into account parameter dispersions).

## 2.4 Synthesis

The two methods described in this section propose alternative ways to counteract the main drawbacks of the local calibration process. The mixed approach avoids the tricky steps of determination of emissions target for the OP as well as the smoothing stage with an advanced optimisation tool which directly optimise the parameter maps considering the cumulated emissions. This approach keeps the same workflow as the local approach and involves the local model, which makes it a very simple approach. The global approach is different with specific developments. The determination of the global Operating Space is very important for the process. Then, the cumulated emissions are directly optimised with the cycle trace and no more OP weight is required. The map optimisation tool manages many constraints, like the smoothing constraints on the engine response maps. This is particularly useful for the combustion noise, as shown in Section 3.3.

Beyond the technical improvement brought by the use of advanced calibration methods, it also allows to reduce the global duration of the calibration process. Figure 7 presents

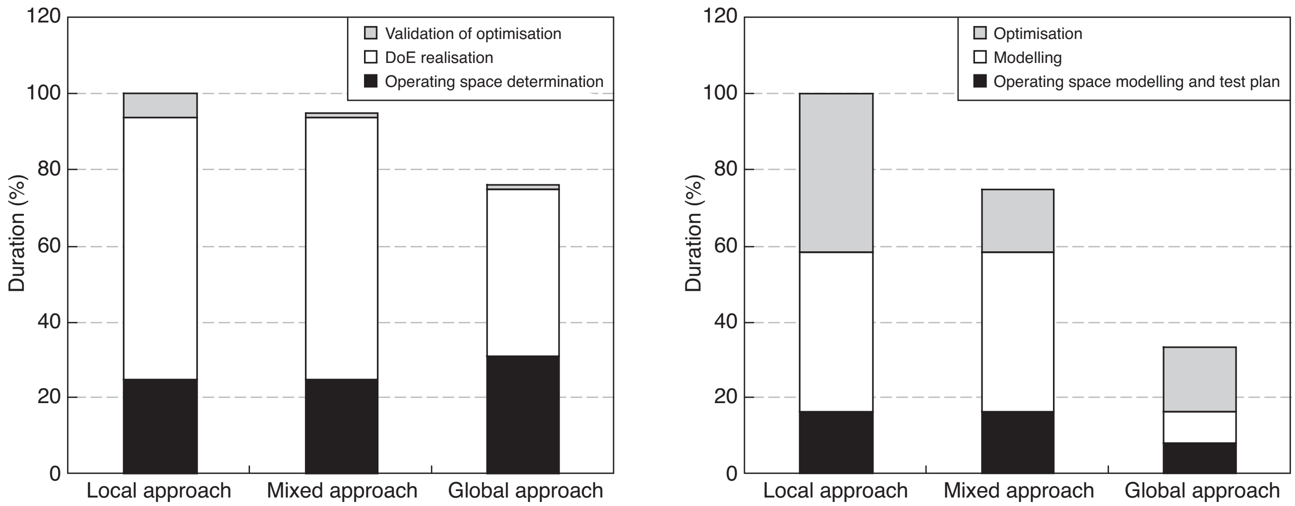


Figure 7

Test duration (left) and post-processing time (right) for several calibration approaches.

the difference in duration for the test realisation as well as for the post processing work. The mixed approach has the same duration for Operating Space determination and modelling. However, the optimisation work is significantly reduced as well as the validation of optimisation. Indeed, if the optimisation work is more complex, it is performed only once, while local optimisation has to be performed for each OP. Furthermore, the local optimal settings have also to be validated for each OP, while some steady state tests (or cycle reproduction if dynamic test bed is available) are enough to validate a new calibration. The global approach reduces the duration of calibration by more than 20%. Even if the Operating Space determination is longer, the global DoE will be shorter than the multiple local DoE. The validation of optimisation is the same as for the mixed approach. For the post-processing work, it is also very interesting, with a reduction of duration up to 65%.

## 3 ENGINE AND FUEL MATCHING ENHANCEMENT IN HOT CONDITIONS

### 3.1 Fuel Impact on Emissions and Combustion

The effects of biofuels and especially of ethanol on combustion behaviour and engine-out emissions have been extensively described in the literature [2, 9-10].

Table 2 illustrates the raw results of the proposed fuel formulation at 2 000 rpm, BMEP 4 bar, comparing the ethanol based Diesel fuel and a reference Diesel one. The injection strategy is composed of a pilot injection and a main injection for this Operating Point.

It can be noticed that soot emissions are drastically reduced. The higher oxygen content of the alternative blend owing to the introduction of ethanol and Biodiesel associated to the lower soot precursors concentrations as explained in Section 1.2 explains the improvement of smoke emissions [11, 12].

The nitrogen oxides ($NO_x$) emissions are increased due to the higher air mass flow required by the engine control on each Operating Point. The lower energy content and density of the new blend “mislead” the engine control about the setpoint which is defined by the engine speed and the injected quantity. Therefore a lower EGR rate is achieved leading to higher $NO_x$ emissions.

HC and CO emissions are significantly increased when changing the fuel. The high oxygen content can also lead to an over-lean mixture at low load when global equivalence ratio is already low. In this case the strong latent heat of vaporization can be an added negative point leading to a slow vaporisation, a lower flame temperature [13] and so an incomplete combustion and consequently higher HC and CO emissions [14-18]. A longer mixing process could also occurred due to the modified auto-ignition properties (poor Cetane Number of ethanol).

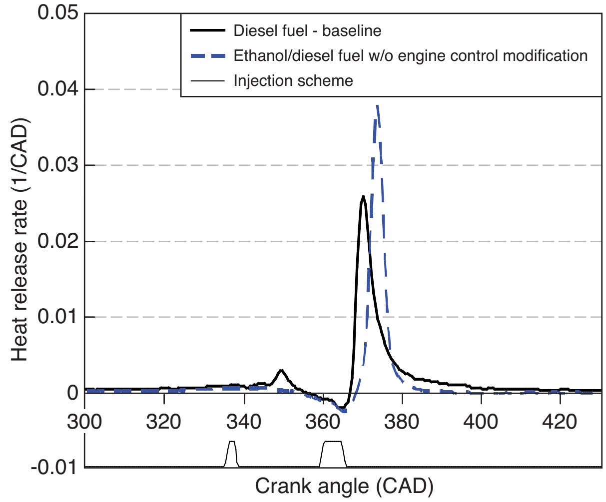


Figure 8

Heat release rate comparison of reference Diesel fuel and ethanol based Diesel fuel at 2 000 rpm, BMEP 4 bar.

The Brake Specific Fuel Consumption (BSFC) is raised up owing to the lower energy content and density of the alternative blend.

Finally the combustion noise is increased due to higher cylinder pressure gradients and higher heat release rates presented in Figure 8 [18]. This is explained by the delayed combustion of the pilot injection which partially burns or even burns within the main injection.

### 3.2 Preliminary Study to Optimise the Engine Calibration

The effects of the alternative blend on the combustion and the engine-out emissions show that optimising the engine settings is mandatory to take advantage of the new formulation properties [8, 19]. The large PM reduction paves the way to a new optimisation of the $NO_x$/PM/$CO_2$ trade-off [12].

The first stage to improve the engine behaviour is to counteract the lower density and lower heating value of the new fuel by changing the fuel density set in the engine control. It was observed that for a similar driver demand, the required injected quantity is higher with the ethanol based Diesel fuel than with the standard Diesel fuel. Hence, as the injected quantity is one of the input of the engine calibration maps, it leads to an increase of the air mass flow setpoint and consequently a drop of the EGR rate of about 5%. By modifying the engine control variable concerning the fuel density and taking into the lower density and the lower calorific value, the engine control is back into the standard Diesel inputs for the calibration maps. Figure 9 illustrates this approach.

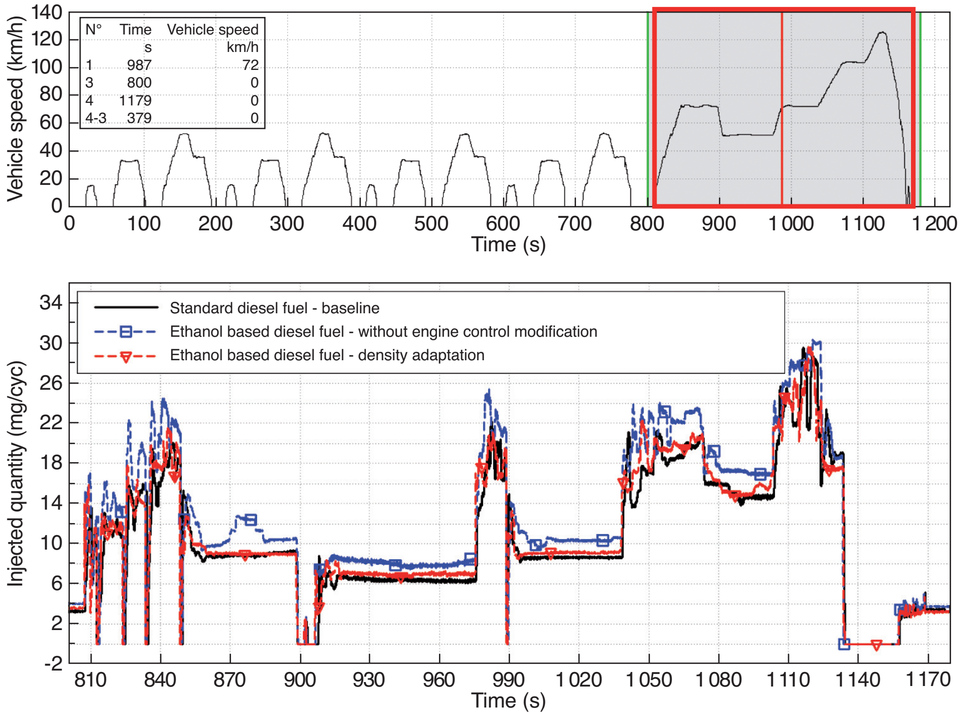


Figure 9

Injected quantity comparison before and after engine control density adaptation.

It was also noticed in the previous analysis that the combustion phasing is shifted closer to the expansion stroke. Then advancing the start of injection could be a solution to deal with the increase of the auto-ignition delay. However, the noise whose level is already high, might further increase. The combination of increased EGR rate and decrease of injection pressure seems necessary to manage this issue as well as decreasing further $NO_x$ emissions. The main characteristics of the injection (scheme, duration of each injection, dwell time) should be modified in order to ensure a proper pilot injection combustion for example, as explained also in previous publications [16, 19, 20].

Therefore, to illustrate this last statement it was decided to focus on some dwell times sweep between pilot and main injections (with a reduction of 750 μs until an increase of 500 μs of this time); they are performed at 2 000 rpm, BMEP 4 bar. Table 2 and Figure 10 display the results of these tests. It could be observed that the pilot combustion occurs only if it gets closer to the main injection owing to better thermodynamics conditions to burn. Thus the air/fuel mixing is enhanced and the combustion takes place earlier during the cycle; it leads to lower HC and CO emissions. The PM emissions are still low whatever the dwell time is chosen and $NO_x$ emissions are barely influenced.

Some further analysis and studies concerning injection scheme or pilot injection timing effects at different engine speeds and loads were presented in previous references [10, 16, 19, 20]. Beyond the PM emissions reduction partly due to the oxygenated content of this alternative fuel, these analyses lay down guidelines for recalibration and prove the potential benefits of updating the engine settings to improve the engine/fuel matching.

### 3.3 DoE and Modelling

This section presents the theoretical results obtained with both methods described in Section 2. These results are then compared to the experimental results performed on the engine test bed. Table 3 provides the theoretical cumulated emissions with the optimised parameter maps. The results with reference maps and with the ethanol based Diesel blend are different from an approach to another, due to the different ways of calculation for the cumulated emissions. It is a weighted sum of emissions based on several OP for the mixed approach, while the emissions are evaluated at each instant during the cycle with the global method, the cumulated emissions are then obtained by integration. In order to compare the results, they are dimensionless with a 100 value for reference maps with ethanol/Diesel fuel. The thresholds for the cumulated emissions are chosen in concordance with the goal of the project which consists in fulfilling the Euro 5

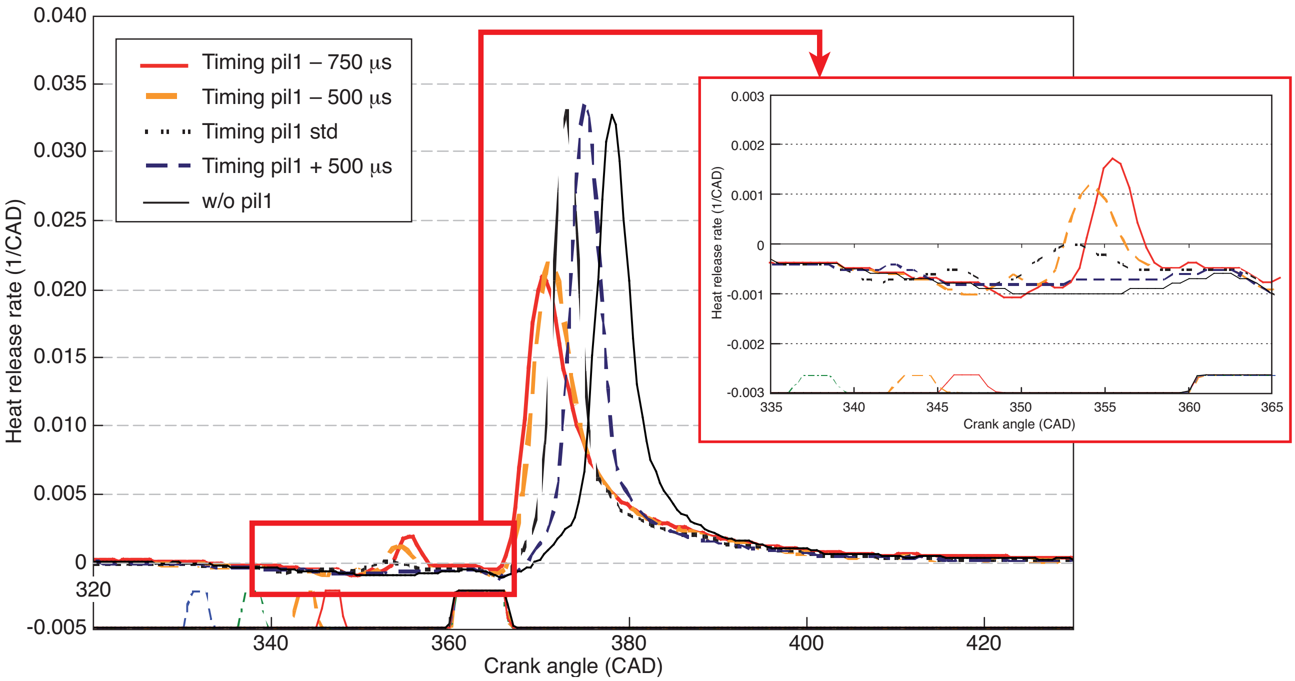


Figure 10

Effect of pilot injection timing on heat release rate at 2 000 rpm, BMEP 4 bar.

TABLE 2

Effect of pilot injection timing on engine-out emissions, fuel consumption and combustion noise at 2 000 rpm, BMEP 4 bar

| | | Ethanol/Diesel Blend | | | | | |
|---|---|---|---|---|---|---|---|
| | | | Engine control: updated fuel density | | | | |
| | Diesel fuel - baseline | Standard settings | Standard Pil1 timing | Standard Pil1 timing + 500 µs | Standard Pil1 timing – 500 µs | Standard Pil1 timing – 750 µs | Main w/o pilot |
| $CO_2$ | 100 | 98 | 97 | 97 | 97 | 97 | 99 |
| CO | 100 | 202 | 177 | 260 | 97 | 78 | 507 |
| $NO_x$ | 100 | 137 | 101 | 104 | 101 | 104 | 117 |
| HC | 100 | 162 | 102 | 160 | 80 | 77 | 485 |
| PM | 100 | 1 | 3 | 1 | 7 | 11 | 0 |
| BSFC | 100 | 114 | 114 | 116 | 113 | 113 | 122 |
| Combustion noise (dB) | 89.3 | 92.9 | 92.2 | 93.4 | 88.4 | 88.2 | 90.9 |

standards. In both cases, a strong reduction of $NO_x$ emissions is obtained, as well as an important decrease of HC and CO emissions. No penalty is observed for the $CO_2$ emissions, which is also a target of the recalibration. The PM emissions significantly increase, since no threshold was used for this response due to the low confidence in local and global PM emissions models.

The first step of the validation of new calibration is the realisation of tests in steady-state conditions. Thereby a variation of load is done for several engine speeds. It is then possible to analyse engine outputs over the operating area. The results are very close for both mixed and global approaches, so the main trends of the results are available in both cases. Figure 11 presents the relative difference for the

TABLE 3

Theoretical results for mixed and global optimisation

| | Reference maps | Mixed optimisation | Global optimisation |
|---|---|---|---|
| $CO_2$ | 100 | 100 | 100 |
| CO | 100 | 65 | 60 |
| $NO_x$ | 100 | 64 | 65 |
| HC | 100 | 70 | 64 |
| PM | 100 | 780 | 600 |

CO emissions before and after optimisation with global approach. These emissions are largely reduced for low load values (up to 60%) where they are the most critical. For higher loads, where the values are lower and the catalyst is generally activated, the emissions are similar. The trend is the same for HC emissions, which also drop of 60% for low loads. $NO_x$ emissions decrease over the whole operating area, while PM emissions raise. Finally, $CO_2$ emissions are slightly modified, varying about 2% in comparison with the initial values.

A second way to use these results for calibration validation is to estimate cumulated emissions over a normalised cycle, with quasi-static hypothesis. Indeed, using a trace of a cycle, it is feasible to estimate the instantaneous emissions values by interpolation within the engine response maps, assuming steady state engine behaviour during cycle. The cumulated value can be obtained by integration. Table 4 shows the cumulated results estimated from quasi-static tests. As stated before, the results are very close for both approaches. HC and CO emissions drastically decrease by approximately 50%, which is more than estimated during the optimisation. Moreover, $NO_x$ emissions decrease slightly less than the model expectation, while $CO_2$ emissions are contained. As expected, PM emissions significantly increase. Finally the relative gains predicted by the optimisation tool are quite well correlated with experimental results.

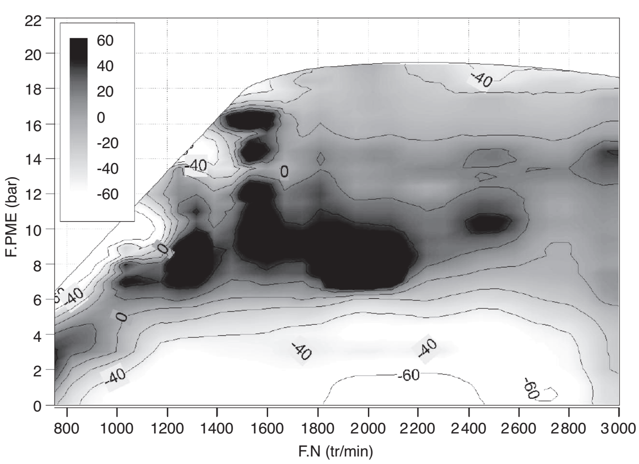


Figure 11

Relative difference of CO emissions before and after optimisation.

TABLE 4

Results of cumulated emissions from quasi-static tests and trace of cycle

| | Reference maps | Mixed optimisation | Global optimisation |
|---|---|---|---|
| $CO_2$ | 100 | 99 | 99.6 |
| CO | 100 | 52.5 | 48 |
| $NO_x$ | 100 | 73 | 78 |
| HC | 100 | 54 | 52 |
| PM | 100 | 510 | 500 |

The last operation regarding the quasi-static validation of engine calibration results consists in performing an acoustic analysis on the entire driving field. Indeed, if the decrease of absolute noise level is important for driveability, the homogeneity of noise level against engine load is also very important, in order that the driver does not feel any discontinuity when accelerating. Therefore, in addition to a maximum combustion noise level constraint, a constraint on the combustion noise level homogeneity has been implemented in the optimisation tool for the global approach.

Figure 12 displays the combustion noise level for each load at several engine speed values for several configurations. With the reference maps in ethanol/Diesel blend (top left), for a given engine speed value, the heterogeneity of noise level is important with differences up to 12 dB at 2 000 rpm, in comparison to the standard Diesel fuel (top right) which shows much more homogeneous combustion noise values. The optimised maps with the mixed approach (bottom left) present lower values of combustion noise. This is consistent with the constraints of maximum threshold used during the optimisation for each OP. However, there are still some areas of heterogeneity, for example at low loads (1 and 2 bar of BMEP). These points should be improved by modifying the parameter settings. At last, the acoustic analysis results after engine maps optimisation with global approach (bottom right) present combustion noise values much more homogeneous against engine load, with differences of less than 5 db over almost the entire engine speed domain. The noise level homogeneity against engine load is close (or even better) to the one obtained with standard Diesel fuel. This is the consequence of the smoothing constraint on the combustion noise maps which is managed by the map optimisation tool. This point is a significant advantage of the global approach in comparison to the mixed one.

As presented in Section 1.1, a dynamic test bed was used. It allows to reproduce standard driving cycles (NEDC).

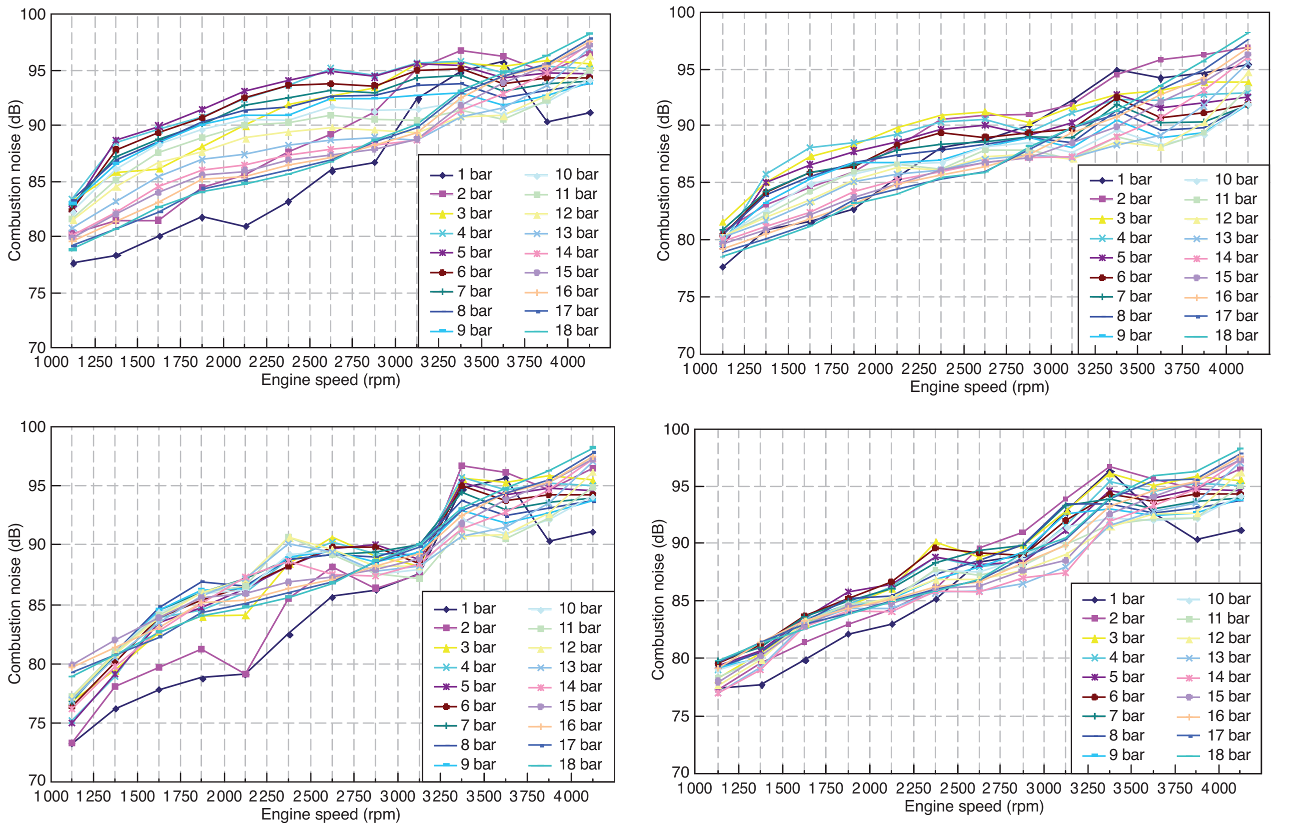


Figure 12

Acoustic analysis for the several configuration. Top left: reference maps with ethanol/Diesel blend. Top right: reference maps with standard Diesel fuel. Bottom left: optimised maps (mixed approach) with ethanol/Diesel blend. Bottom right: optimised maps (global approach) with ethanol/Diesel blend.

Thereby, the optimised engine maps were tested in dynamics during warm NEDC cycles in order to check if the quasi-static gains observed are effective in real operating conditions. The estimated cumulated emissions with quasi-static models are representative, as shown in Table 5. CO and HC emissions show a decrease similar to the estimation of the models, which had already been confirmed with the results of the quasi-static tests. $NO_x$ emissions also drop off significantly, but less than estimated during the optimisation. This is consistent with the quasi-static tests results. Thus, the dynamic test bed appears as an efficient way to validate quickly the calibration results. The vehicle results are presented in the next section.

TABLE 5

Results of cumulated emissions issued from transient tests on engine test bed

| | Reference maps | Mixed optimisation | Global optimisation |
|---|---|---|---|
| $CO_2$ | 100 | 100.4 | 101.7 |
| CO | 100 | 59 | 60 |
| $NO_x$ | 100 | 73 | 80 |
| HC | 100 | 69 | 70 |
| PM | 100 | 97.5 | 230 |

## 3.4 Vehicle Results

The last stage to assess the new calibration settings is to perform some vehicle tests. The NEDC results at the chassis dynamometer, given in Table 6, are obtained downstream the catalyst in warm conditions, which explains the low CO and HC emissions. Concerning the mixed approach, the $NO_x$ emissions are consistent with the transient test performed at the test bed presented in Table 5. With the global optimisation, the $NO_x$ emissions are even slightly lower than the expected decrease issued from the test bed. It can be noticed that the $NO_x$ emissions decrease, from the standard Diesel fuel to the alternative fuel with an optimisation of the calibration, is consistent with the difference between Euro 4 and Euro 5 standards (28%).The absolute PM emissions are close to zero whatever the calibration and therefore a slight absolute increase leads to a large relative difference. The $CO_2$ emissions with both approaches are close to the standard Diesel ones.

TABLE 6

Results of cumulated emissions on warm NEDC cycle from chassis dynamometer

| | Reference maps | Mixed optimisation | Global optimisation |
|---|---|---|---|
| $CO_2$ | 100 | 97 | 98.5 |
| CO | 0 | 0 | 0 |
| $NO_x$ | 100 | 80 | 67 |
| HC | 100 | 74 | 73 |
| PM | 100 | 180 | 180 |

## 4 COLD CONDITIONS IMPROVEMENT

### 4.1 Introduction

The calibration approaches presented in the previous sections are available for warm operating conditions. However, European standards are based on cumulated emissions over a normalised cycle with a start in cold conditions at 20°C. This significantly impacts the engine behaviour and the pollutants emissions. Furthermore, the engine behaviour during the rise of the coolant temperature, and also the aftertreatment (catalyst light-off), has to be taken into account in order to optimise the cumulated emissions. This section deals with advanced calibration methods applied to a cold normalised cycle. These methods depend on the control algorithms to manage the cold conditions in the ECU. For this application, a correction map is used for cold operation, and it is applied via a correction factor to the warm operation maps as described in Figure 13.

#### 4.1.1 Global/Local Approach

The first method consists in mixing global and local approach. The global one is used to calibrate warm operation while local approach is used for cold operation. In theory, it would be possible to realise a global cold and warm model, including the coolant temperature as parameter. However, this kind of test would be very difficult to perform:

- the number of experiences would be important;
- the Operating Space would be difficult to determine;
- the coolant temperature control is slow, particularly when two successive experiences are made with a significant difference for this parameter;
- the risk of engine drift (fouling, wear) is high, especially with a low coolant temperature (this could impacts the repeatability and the engine stability).

Indeed, the results would not be completely exploited since this global model would supply a continuous information for the coolant temperature dimension, while this parameter is strongly “discreticised” in the ECU. A pragmatic approach intends to use the main advantages of the local method for the cold calibration:

- local DoE are short, so the risk of engine drift during the test is limited;
- the number of OP is reduced, so the duration of the process is shortened;
- the accuracy of local models is important for cold calibration, especially for HC and CO models.

DoE are performed in cold conditions (40°C for coolant temperature) on several OP already used for warm operations as shown in Figure 14, leading to local models. The cold correction maps are optimised using these models with the local process presented in Appendix.

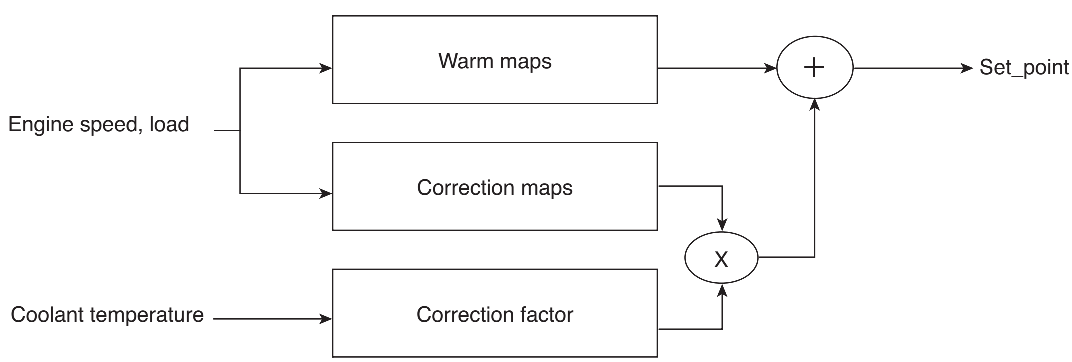


Figure 13

Cold management in ECU.

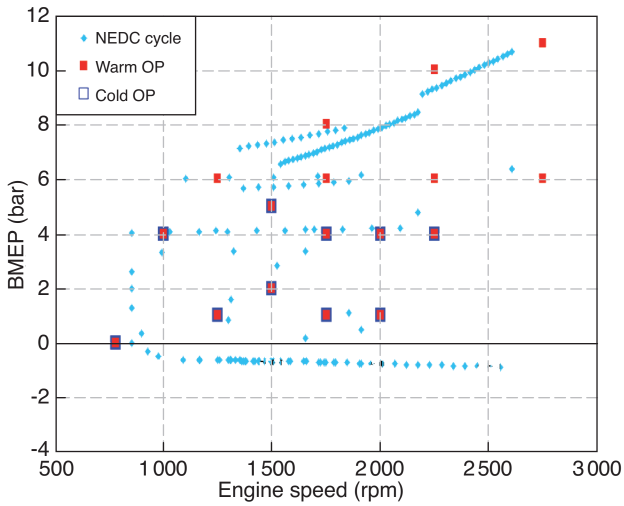


Figure 14

Cold and warm OP definition for NEDC cycle.

#### 4.1.2 2.5D Mixed Approach

Like the method described previously, the calibration of cold operation is usually performed after the warm calibration. By applying a sequential approach, the main issue is to define the optimisation targets (*i.e.* mainly to fix the limits on the cumulated emissions over the cycle) for each step without influencing the results of the other step. Nonetheless, cumulated emissions are significantly different from a cycle in warm conditions to a cycle in cold conditions, especially when considering the emissions downstream the catalyst. Hence it was decided to optimise simultaneously standard settings and cold corrections, by directly optimising the cumulated emissions over the whole cold cycle. Besides, the optimisation tool has been improved, including the introduction of area management, a feature originally developed to manage the several injection strategies over the cycle operating area. Indeed, the cold operating area can be considered as an operating area, with its own OP. This method is described in this section.

Cold and warm parameter maps (or cold correction maps) are directly optimised from local emissions models (obtained from DoE in cold and warm conditions) with objectives on cumulated emissions over NEDC. A key step of this approach is the calculation of the OP weights. Indeed, the weights are very important in order to get some representative estimated cumulated emissions over a normalised cycle. A calculation method has been developed and is illustrated in Figure 15. At each second of the cycle, a weight is attributed to three warm OP and three cold OP. The distribution of the attributed weight to cold or warm OP is calculated from an evolution law depending on the coolant temperature. The main issue is to determine a relevant evolution law in order to be representative of the engine behaviour during the engine warm up.

Some preliminary tests were performed in order to consider the impact of engine warm up on the pollutants emissions. After the engine starts, the engine parameters are fixed, and the evolutions of the pollutants emissions are observed. An example at 2 000 rpm and 1 bar of BMEP is given in Figure 16. The tests were performed on several OP and the engine

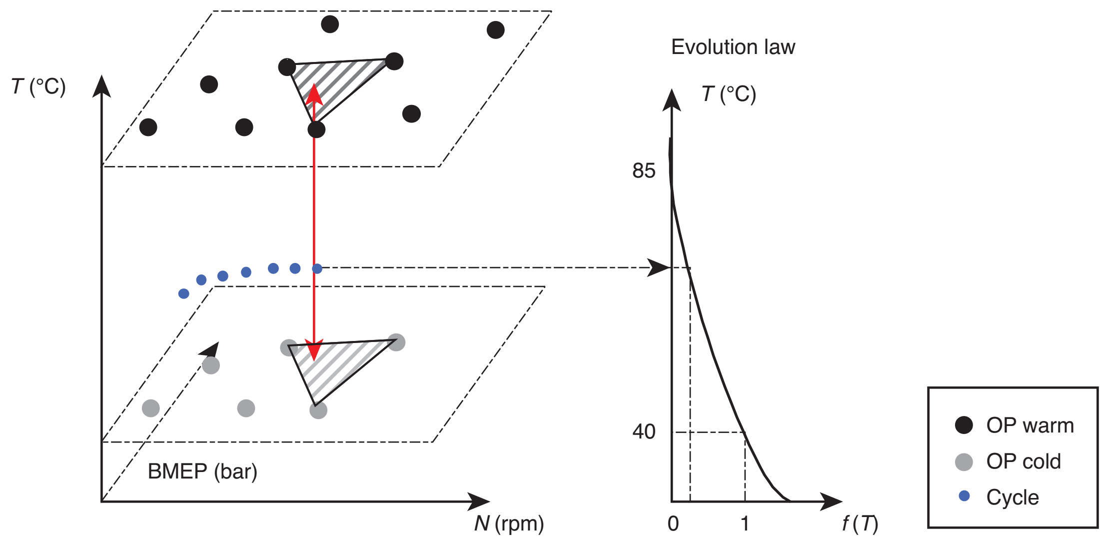


Figure 15

Principle of OP weight calculation with temperature.

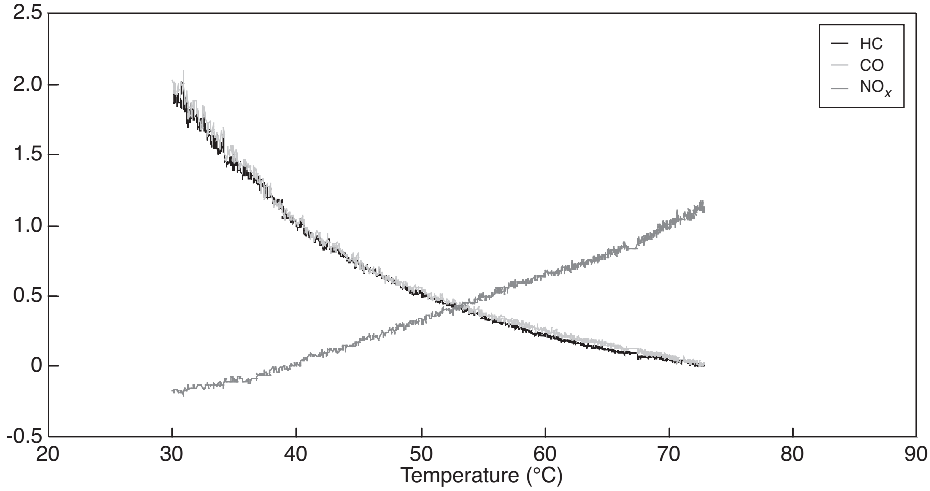


Figure 16

Evolution (dimensionless) of pollutant emissions at 2 000 rpm and BMEP 1 bar.

behaviour is similar in all cases. Thus, it is possible to define general evolution law for each engine out emission to be implemented in the calculation of OP weight. These laws will be the only piece of information relative to the coolant temperature during the calibration process.

For this application, a total of 26 OP (17 warm OP and 9 cold OP) were chosen, and the weights were calculated as described earlier. Table 7 presents the theoretical results obtained after optimisation on a cold NEDC cycle. The results are very similar to the warm results, with an important decrease of $NO_x$, HC and CO emissions. Table 7 gives engine-out emissions *i.e.* upstream the catalyst.

TABLE 7

Theoretical results for the 2.5D mixed approach

| | Reference maps | Optimised maps |
|---|---|---|
| $CO_2$ | 100 | 100 |
| CO | 100 | 70 |
| $NO_x$ | 100 | 65 |
| HC | 100 | 75 |
| PM | 100 | 533 |

In addition to this process, it would be interesting to consider the DOC efficiency concerning the cold settings. Indeed, it is not relevant to further decrease the HC or CO emissions during the EUDC part of the cycle, when the DOC is activated. Therefore, the future developments of the calibration method could be oriented in order to implement the DOC efficiency in the evolution law in order to be representative of the pollutants emissions downstream the catalyst.

## 4.2 Vehicle Results

Figures 17 and 18 present the vehicle results with the different approaches presented in this section concerning the cold conditions optimisation. All the results are obtained with the ethanol based Diesel fuel. The pollutants emissions are measured downstream the catalyst (unlike the test bed results (upstream the catalyst)). It can be noticed that there is only a minor effect of the calibration on $CO_2$ emissions. HC and CO emissions are dropped of 30% to 45% compared to reference maps. PM emissions are also reduced in cold conditions because they are mainly composed of the Soluble Organic Fraction (SOF) with the ethanol based Diesel fuel [21]. Therefore the HC emissions change was a reliable information to know PM emissions trend. This explains also the difficulty to have reliable measurement at the test bed with the opacimeter compared to roller test bed measurement (measuring the mass of PM on a filter). $NO_x$ emissions are slightly increased with the optimisation of the cold corrections (upstream catalyst) (adjusted mainly for HC and CO emissions) and the margin compared to the reference maps is still consistent with a Euro 5 objective over NEDC. It can be noticed that HC and CO emissions are decreased significantly by using the mixed approach during the first cold urban cycle; however the cumulative emissions over NEDC show that there is no gain obtained with the mixed approach due to the higher quality of the global method in warm conditions.

Table 8 sums up the final NEDC results obtained at the chassis dynamometer. The different stages are presented. The standard Diesel fuel used on a Euro-4 compliant vehicle permits to obtain already low CO, HC and PM emissions compared to Euro 4 standard. HC and CO emissions already

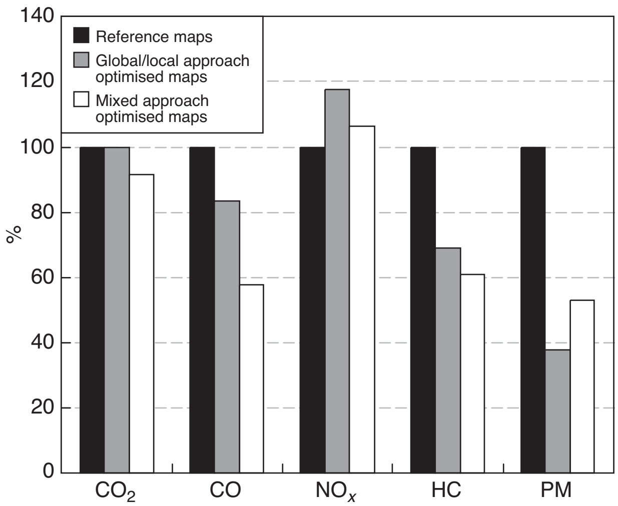


Figure 17

Vehicle results during first urban cycle with the different approaches.

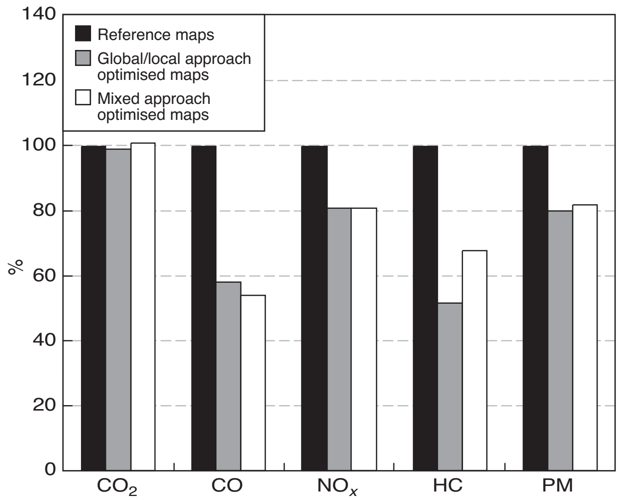


Figure 18

NEDC vehicle results with the different approaches.

TABLE 8

Final vehicle results on cold NEDC

| | Diesel | Ethanol/Diesel | | | Standards | |
|---|---|---|---|---|---|---|
| | | | Modified density | | | |
| | Reference maps | Reference maps | Reference maps | Optimised maps | Euro 4 | Euro 5 |
| $CO_2$ (g/km) | 119.0 | 116.9 | 119.7 | 119.0 | | |
| CO (g/km) | 0.135 | 0.606 | 0.602 | 0.322 | 0.500 | 0.500 |
| $NO_x$ (g/km) | 0.183 | 0.293 | 0.177 | 0.143 | 0.250 | 0.180 |
| $NO_x$ + HC (g/km) | 0.203 | 0.423 | 0.308 | 0.210 | 0.300 | 0.230 |
| PM (g/km) | 0.008 | 0.004 | 0.005 | 0.004 | 0.025 | 0.005 |
| Fuel consumption (l/100 km) | 4.48 | 5.22 | 5.35 | 5.29 | | |

satisfy Euro 5 regulations and $NO_x$ and PM emissions are close to Euro 5 standard. $NO_x$ emissions show a margin of around 20% compared to the Euro 4 standard.

By changing only the fuel, unlike PM emissions, the $NO_x$ emissions increase dramatically as explained in Section 3.2, as well as HC and CO emissions.

The modification of density detailed in Section 3.2 permits to bring back the $NO_x$ emissions to the reference Diesel fuel level; nonetheless, other pollutants emissions are not meeting the Euro 5 standards; this highlights the need to perform a new calibration of the engine parameters to fully take advantage of this alternative fuel formulation.

The optimised maps enable to meet Euro 5 stringent regulation without any hardware modification starting with a Euro-4 compliant vehicle and more specifically not being equipped with a DPF. The $CO_2$ emissions are similar to the Diesel fuelled Euro-4 vehicle over the cold NEDC. Since the volumetric energy content of the ethanol based Diesel fuel is somewhat lower than standard Diesel fuel, it was expected that the (volumetric) fuel consumption would be slightly higher. $NO_x$ emissions are still largely below the Euro 5 limit permitting to have a margin regarding production dispersion. HC or PM (mainly SOF in cold conditions) emissions are closer to the limit and some minor hardware enhancements could answered this issue. An optimisation of the formulation of the DOC washcoat [15] could be a solution as well as an update of the injection system.

## CONCLUSION

To manage some of the key priorities of the transportation area, an innovative fuel formulation has been proposed using ethanol incorporated in a Diesel fuel. The use of a renewable fuel allows reducing well-to-wheel $CO_2$ emissions [22]. It is also an answer to limit the Diesel import in Europe by balancing the Diesel and gasoline requirements without modifying the fuel distribution infrastructure [9].

After solving the issues concerning the blend miscibility or the Cetane Number, some heat release analysis allowed identifying the formulation effects on the combustion process. It has been noticed that the pilot injection combustion occurs in the same time as the main injection for example and it leads to higher HC and CO emissions as well as high combustion noise. Moreover the main advantage of biofuels consists in the dramatic PM emissions reduction due to the high oxygen content and low soot precursors concentrations.

These statements lead us to perform a new calibration of the engine parameters. Firstly, the low energy content of the new fuel shifts the engine control Operating Point. Modifying the density recorded by the control software allows fixing this problem. Secondly advanced calibration methods have been developed in order to adapt the engine control settings to the features of this fuel formulation. An exhaustive report of these methods is presented in this paper. The global method in warm conditions provides interesting results in a time reduced scale. To manage the warm-up and catalyst light off, a mixed approach has also been used. Besides the use of a transient test bed was helpful to quickly assess and validate new settings and was in good agreement with the vehicle results.

Finally starting with a Euro-4 compliant vehicle and without any hardware modification, so without being equipped with a DPF, the combination of an innovative fuel formulation and advanced calibration methods allows to meet Euro 5 standards. The absence of DPF enables to decrease the global powertrain cost and to reduce the calibration time by removing the DPF regeneration settings.

The processes presented in this paper could be used to perform the calibration of the vehicle with another fuel but it requires to perform the whole process for each blend formulation. The current development of fuel quality sensors might be an opportunity to integrate a fuel parameter in the global model (for example fuel density or cetane index, etc.). Therefore a dedicated map in the engine control software could take into account small blend variations to adapt the engine parameters (start of injection, etc.) to the current fuel properties. However if the blend formulation changes, it will require to extend the global domain during the calibration process and this could affect significantly the model quality and accuracy.

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*Final manuscript received in July 2011*
*Published online in November 2011*

## APPENDIX

### Synopsis of the different advanced calibration methods