2020-01-2068 Published 15 Sep 2020

Assessing the Efficiency of a New Gasoline Compression Ignition (GCI) Concept

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Citation: Cracknell, R., Bastaert, D., Houille, S., Châtelain, J. et al., "Assessing the Efficiency of a New Gasoline Compression Ignition (GCI) Concept," SAE Technical Paper 2020-01-2068, 2020, doi:10.4271/2020-01-2068.

Abstract

A practical Gasoline Compression Ignition (GCI) concept is presented that works on standard European 95 RON E10 gasoline over the whole speed/load range. A spark is employed to assist the gasoline autoignition at low loads; this avoids the requirement of a complex cam profile to control the local mixture temperature for reliable autoignition. The combustion phasing is controlled by the injection pattern and timing, and a sufficient degree of stratification is needed to control the maximum rate of pressure rise and prevent knock. With active control of the swirl level, the combustion system is found to be relatively robust against variability in charge motion, and subtle differences in fuel reactivity.

Results show that the new concept can achieve very low fuel consumption over a significant portion of the speed/load map, equivalent to diesel efficiency. The efficiency is worse than an equivalent diesel engine only at low load where the combustion assistance operates.

In this work, a detailed, dynamic longitudinal simulation model was created that allows accurate CO_2 emission and fuel consumption predictions for a typical C-segment vehicle in the New European Drivecycle (NEDC), Worldwide Harmonized Light-Duty Test Cycle (WLTC) and Real Driving Emissions (RDE) driving cycles. Modelling and simulation were performed in the GT-Suite simulation environment.

The vehicle simulation for the NEDC with its low load demand shows that the outstanding fuel consumption at higher engine loads is more than compensated by the poor efficiency at lower engine loads. For the WLTC and RDE cycles, the fuel consumption improves relative to the NEDC due to the higher average load. In order to take better advantage of the GCI concept's performance at high loads, a virtual engine map for a 4-cylinder engine was also created that is capable of running in cylinder deactivation mode. Avoidance of low-load operating points in this way leads to a significant improvement in cycle efficiency

Introduction

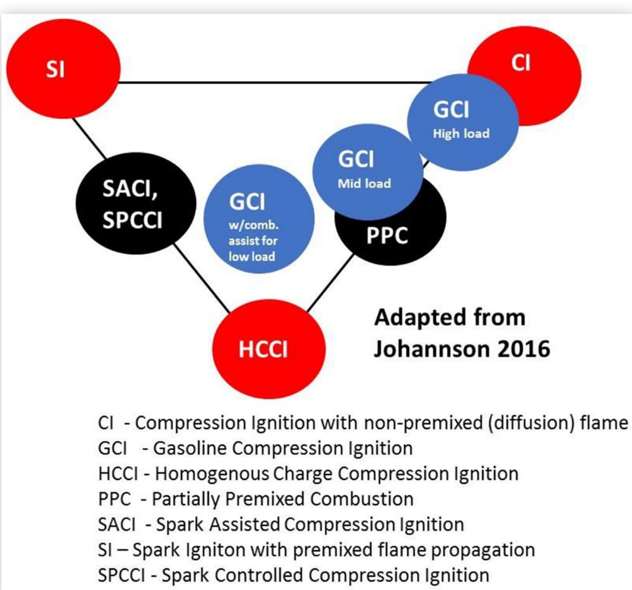
Over the last 30 years or so, advanced compression ignition (CI) combustion has been an active research topic: The general feature of these concepts is that the fuel-air mixture is much more premixed than in conventional CI, with the fuel then combusting under Low Temperature Combustion (LTC) conditions, thereby simultaneously minimising NOx and soot emissions and reducing heat losses [1,2]. The extreme embodiment of this approach is where the mixture is completely homogenous and ignites during the compression stroke as a distributed reaction - Homogenous Charge Compression Ignition (HCCI), with the combustion phasing controlled purely by the autoignition chemical kinetics of the fuel/air mixture [3]. However, "pure" HCCI can be very difficult to control and operate over the whole speed-load range, which has significantly limited its applicability [4].

Johannsson [5] has conceptualised different types of combustion systems as a triangle, in which all embodiments reside (Figure 1). Two of the poles of the diagram represent conventional spark ignition (SI) and compression ignition (CI) systems. The third pole represents "pure" HCCI. Many practical embodiments of advanced combustion systems that have been proposed that represent a compromise between SI, CI and pure HCCI.

Spark Assisted Compression Ignition (SACI) is a concept in which the fuel/air mixture is homogenous, but there is a spark ignition that leads to a propagating flame, which in turn increases the pressure and temperature of the remaining charge so that it auto-ignites [6,7]. In some variants a late second injection is used to create some charge stratification [8]. The key point about SACI is that it is the spark that controls the combustion phasing.

Partially Premixed Combustion (PPC) represents a compromise between pure HCCI and CI [9]. As the name

FIGURE 1 Taxonomy of Combustion Systems [29].



suggests, the fuel is much more premixed than conventional diesel combustion before ignition occurs, although combustion phasing is still controlled by the injection timing. Previous Concawe work has shown that this type of combustion can be achieved on a surprisingly wide range of fuels using a CI engine designed for diesel fuel, as long as there is an adaptive control system to optimise the injection timing in a way that compensates for the different cetane number of the fuels [10,11,12]. However, in the same test series, European market gasoline proved to be too resistant to ignition to operate satisfactorily using a compression ratio suitable for diesel fuel.

Nevertheless, there are advantages in using gasoline in a partially premixed combustion mode [13]. In particular, the resistance to autoignition of gasoline means that a high compression ratio is required, leading to high efficiency. Such a technology is usually referred to as Gasoline Compression Ignition (GCI). Further important considerations are that the ability of GCI concepts to use an already available market gasoline would allow these concepts to enter the fleet without fuel constraints, and a successful GCI vehicle could potentially compete in predominantly gasoline markets in parts of the world without significant penetration of passenger diesel. A further attraction from the perspective of fuel supply is that more gasoline consumption in passenger cars would help to rebalance the gasoline/diesel fuel demand in European refineries and consequently reduce GHG emissions arising from fuel production.

Operating GCI over the whole speed/load range presents significant challenges. At high load, the combustion system approaches that of conventional CI combustion as illustrated in Fig 1, with an increased challenge to control the pressure rise rate and hence the combustion noise [14]. The same study, however identified low load operability as the biggest challenge for GCI given gasoline's intrinsic resistance to autoignition, and a follow-on modelling study indicated that a variable valve timing arrangement combined with an optimally positioned combustion assist device (either a spark plug or glow plug) provided a potential means to facilitate low load operation [15]. Delphi's GDCI technology employs a complex valvetrain arrangement to achieve low load operation [16,17]. Another approach is to use a lower octane fuel, usually a type of naphtha, which is much less resistant to autoignition at low loads than pump gasoline. It is argued that such a fuel gives additional Well to Wheels efficiency and CO_2 benefits, because the fuel requires less processing during its manufacture [18], however it has a significant disadvantage of not being widely available in the market.

Mazda have announced the development of an engine that operates using gasoline in "Spark Controlled Compression Ignition" (SPCCI) mode over much of the operating map [19,20]. The Mazda SKYACTIV-X employs combustion assist in the form of a spark over much of the engine map and undergoes a mode switch to SI at the highest loads, high engine speeds and when the engine is cold. In SPCCI mode, the charge is relatively homogeneous and dilute, either by being ultra-lean or lean with high levels of EGR. According to Mazda, SPCCI combustion is controlled based on ignition timing thereby making it more akin to the SACI approach.

This is a significant development in that it shows the commercial potential of this technology.

The objective of the present work was to develop a robust GCI system than can operate on a standard European gasoline, over a wide speed/load range. At low loads, a spark is employed to assist the autoignition of gasoline. The combustion phasing is controlled by the injection pattern and timing, and a sufficient degree of stratification is needed to control the maximum rate of pressure rise and prevent knock. The importance of the injection timing to control combustion phasing distinguishes the GCI technology presented here from SACI (or SPCCI), although clearly there are similarities between the two concepts for low load operation, as illustrated in Fig 1.

A previous publication [21] reported on the robustness of the GCI technology discussed in the current work: in addition to a reference ULG 95 gasoline, five market representative gasoline fuels were tested with RONs varying from 92 to 102, and with differing volatilities and ethanol contents. The engine was calibrated on the single ULG-95 reference gasoline fuel, but that calibration allowed operation on a range of gasoline fuels, representing RON levels from 92 to 98, as well as summer and winter grade volatility levels and differing ethanol content. Being able to operate in GCI mode over the whole speed/load range represents a considerable achievement. Additional combustion assist was however required at mid-load points for the 102 RON fuel because of the enhanced autoignition resistance. Reference [21] also reported a preliminary investigation of cold start and warm-up, which indicated that the spark assist may be needed to run at more speed/load points where the engine is warming up. However, spark assist appears to provide an easy, practical and efficient approach to cold operability.

The GCI approach presented in reference [21] and discussed further in this work is able to achieve exceptionally low fuel consumption over a wide area of the speed load map, with ISFC values of 200g/kWh or lower (i.e. equivalent to diesel-efficiency). However, the efficiency deteriorates at low load where the combustion assistance operates. The objective of this paper is to show what this would mean in practice for operation in a simulated vehicle over a set of regulatory drive cycles. To that end, a detailed, dynamic longitudinal simulation model was created that allows accurate CO_2 emission and fuel consumption predictions for a typical C-segment vehicle in the NEDC, WLTC and RDE driving cycles. Modelling and simulation were performed in the GT-Suite simulation environment..

The paper is structured as follows: First the basic features of the engine are summarised, and then the longitudinal vehicle simulation technique and results are presented and discussed.

Summary of Key Features of the Engine Design

Following an initial computational fluid dynamics study of the in-cylinder flow behaviour conducted by PSA, the combustion chamber was modified for GCI with combustion assist but kept within the dimensions of a DW10F PSA 4-cylinder, 2 litre diesel engine of bore 85mm and stroke 88 mm. The length of the reinforced con-rod is 145 mm, and the overall displacement is 499cm^3. A diagram of the engine configuration is shown in Figure 2. With the swirl control vane swirl numbers between 2.45 and 3.6 are possible. An EGR intercooler, upstream from the vane, allows a reduction in EGR temperature and therefore an increase in the density of the intake charge. The compression ratio of the engine is 16:1 with a typical centered omega piston bowl.

A fully open Engine Management System supplied by CIRTEM - Aboard Engineering, was used, combining both injection, spark, and EGR control. Its embedded software based on the in-house control named "ORIANNE" [22], can be configured to fit single and multi-cylinder Gasoline or Diesel engines.

The SCE-GCI_DW10 engine has the capacity to control up to 5 injections per stroke. A 7-hole injector is employed. Injection pressures are relatively low, typically less than 600 bar (i.e. much less than a typical diesel); this potentially allows the use of a gasoline DI system in a future development. A standard spark plug was used with a 1mm spark gap, and 103 mJ spark energy. The spark plug location is 16.5mm from the injector and placed in the targeting of one spray.

The engine was manufactured by Danielson Engineering in Magny-Cours, France, and transferred to CERTAM in St Etienne du Rouvray, France, for calibration and testing.

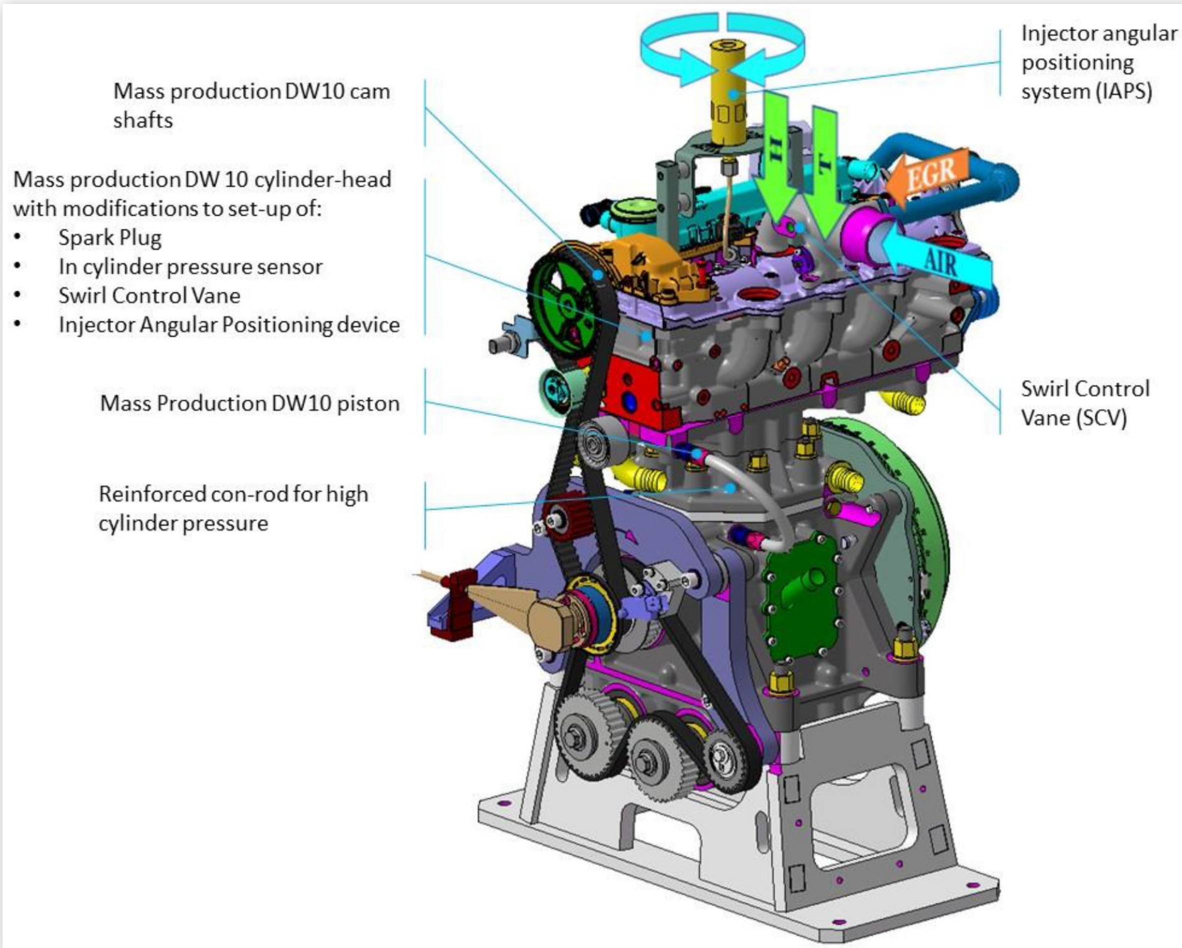
The points chosen on the speed/load map for developing calibration are shown in Fig 3 and Table A1 in the Appendix. The spark assist is used only for points in the lower load ranges with 2 injections per cycle, while 4 injections are used in medium/higher loads. A swirl management strategy, dependent only on engine speed, was employed with a swirl control vane angle of 67.5° between 800 to 1200 rpm, 22.5° between 1350 to 1500 rpm, and 45° between 2000 and 2400 rpm.

The engine was calibrated on the reference fuel described in the following section. The calibration strategy had the following guiding principles:

- The maximum permitted engine-out NOx and PM at each point were chosen such that Euro 6 requirements could be met over the whole driving cycle, assuming the use of a conventional de-NOx system (e.g. SCR) in a production vehicle.
- The timing of multiple injections must control the pressure rise rate to prevent damaging pressure oscillations (aka "ringing" or "knock") that could cause damage [23]. This is less of an issue when the combustion assist is used, because the spark helps reduces pressure oscillations [24]. The challenge of pressure oscillations is discussed in detail in ref. [21].
- Notwithstanding the other two principles, CO_2 emissions were minimised.

The final optimised calibration parameters are given in the appendix. One consequence of the calibration approach to minimise CO_2, whilst tolerating a level of engine-out NOx that can be handled by the after-treatment system is that less EGR is used than in other embodiments of GCI systems. It is assumed that HC and CO emissions can be handled with an

FIGURE 2 GCI Single Cylinder Demonstrator



oxidation catalyst [25], although the assumption would require practical validation.

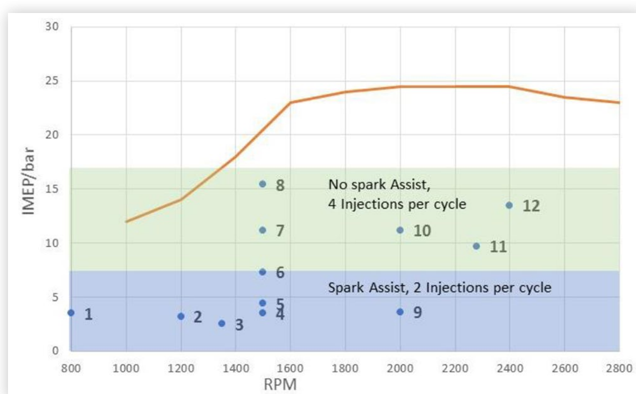
Fuel Selection

The reference fuel was chosen to represent a standard European 95 RON E10 pump fuel meeting the EN 228 specification (Table 1)

The only modification to the pump gasoline was the addition of lubricity improver (300ppm by volume of Infineum R655). No other additives were added. As noted earlier, there is a general consensus that Fuel Injection Equipment (FIE) can be made robust to GCI operation (with modest injection pressure) with regular pump gasoline containing no lubricity improver additive, but the purpose of this project was to investigate the combustion system rather than FIE development and so addition of lubricity improver was precautionary

The overall campaign consisted of testing a range of fuels representing RON levels from 92 to 102 with different volatilities and ethanol content. The (fully warm) engine was able to operate successfully, using the default calibration, on fuels of octane ranging from 92 to 98. No specific effect of volatility or ethanol content could be discerned. These tests on the range of fuels are described in reference 21.

FIGURE 3 Test points used for engine calibration



Longitudinal Vehicle Simulations

Longitudinal vehicle simulations are a well-established technique to convert the performance of an engine on a bench to a vehicle on the road [26, 27, 28]. The total driving resistance force in longitudinal direction is represented by the sum of:

- Rolling resistance (mainly from deformation of the tyres)
- Air resistance (drag)
- Acceleration resistance (i.e. weight dependent inertia)
- Gradient resistance (depends on mass and road gradient)

Vehicle data (Table 2) of a C-segment vehicle in series production with a turbocharged 1.4l gasoline direct injection engine were used as a basis for the simulation model parameterization and calibration within the GT-Suite environment . A plot of model validation is shown in Figure A.1 (appendix).

TABLE 1 EN228 compliant E10 gasoline used in tests

Parameter	Unit	
Density	kg/L	0.7545
DVPE	kPa	60
RON		95.7
MON		85
Total Paraffins Content	% v/v	37.9
Olefins Content	% v/v	10.7
Aromatics Content	% v/v	33.3
Oxygen Content	% m/m	3.4
Ethanol Content	% v/v	9.7

TABLE 2 Parameters for C-Segment Vehicle uses to parametrise the longitudinal vehicle model

Body	
Vehicle weight on test stand	1625 kg
Air drag coefficient	0.27
Projected area	2.185 m^2
Driven wheels	front wheel drive
Engine	
Cylinders	4
Engine size	1,395 L
Power output	103 kW (140PS)
Engine speed at peak power	5000 rpm
Torque	250 Nm @ 1500 rpm
Gearbox	
Type	dual-clutch-transmission
axle ratio (1.- 4. gear)	4.8
axle ratio (5.-7. gear)	3.429
gearbox ratio	
1. Gear	3.5
2. Gear	2.087
3. Gear	1,343
4. Gear	0.933
5. Gear	0.974
6. Gear	0.778
7. Gear	0.653
Tyres	
Manufacturer	Michelin
Tyre size	205 / 55 R16
Tyre pressure	2.5 bar

For this vehicle (with conventional engine technology), IAV have previously obtained extensive data on a chassis dynamometer, running a range of drive cycles. It is therefore possible to calibrate the map-based full vehicle model to reproduce measured CO_2 emission levels.

The indicated specific fuel consumption (ISFC) map produced from the engine tests needs to be converted to a brake specific fuel consumption (BSFC). Since the single cylinder engine´s friction is not representative of conventional multi-cylinder engines, for the simulations, IAV has assumed a friction behaviour that correctly represents the combustion process with high peak pressures and the Diesel-typical layout of the cranktrain.

Both the “New European Driving Cycle” (NEDC) and “Worldwide Harmonized Light-Duty Test Cycle” (WLTC) prescribe an ambient engine temperature at cycle start. This accounts for the significance of the engine cold start phase on the emission behaviour. One of the reasons for the increased CO_2 emissions is the higher friction inside the engine associated with higher engine oil viscosity Within the simulation environment, these circumstances can be accounted for by modifying the Friction Mean Effective Pressure (FMEP) based on the measured engine coolant temperature profile during the driving cycle. For this purpose, a temperature-dependent parameter for modifying the friction map can be derived from the measurements. This sum of the original FMEP (corresponding to warm engine state) and the derived parameter then yields the total break mean friction pressure during the start phase of the cycle. The higher FMEP results in a load increase and at the same time it leads to a shift to higher fuel consumption values.

With the model, it was possible to calculate fuel consumption over a drive cycle for a GCI engine in a C-segment car. In addition to the NEDC and WLTC cycles, a Real-world Driving Emissions (RDE) cycle was also simulated - Figure 5 shows the test trip that has been used in the dynamic reference

FIGURE 4 FMEP map used, showing speed/load points tested

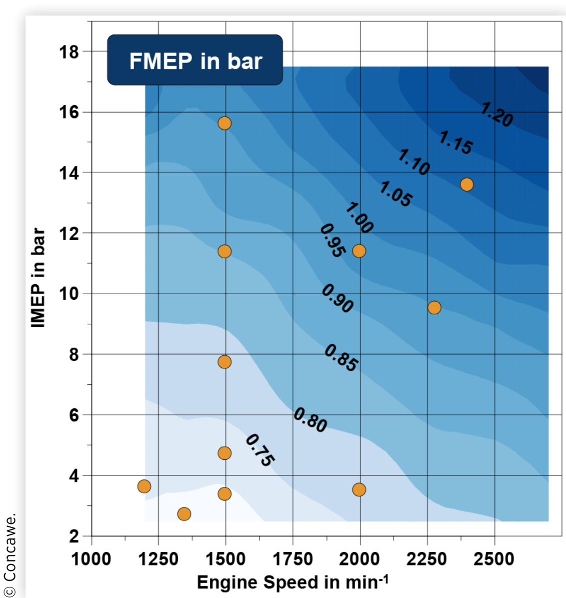


FIGURE 5 Test trip for RDE cycle used in vehicle simulation model

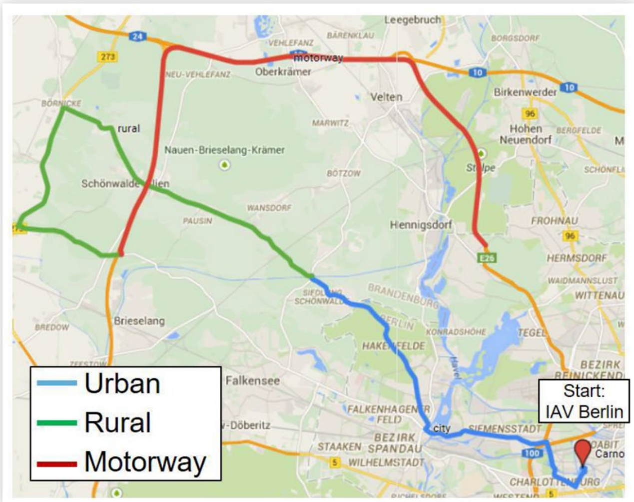


TABLE 3 Parameters relating to the RDE trip

Parameter	Y axis	Z axis	Angle α
Unevaporated fuel mass	13.3%	1.3%	1.3%
Sauter Mean Diameter (droplet)	29.9%	3.9%	11.8%
Turbulence kinetic energy	0.0%	1.0%	0.7%
Energy dissipation rate	2.5%	0.7%	1.3%
Air excess ratio	3.3%	2.3%	3.0%

vehicle model. The total trip distance was 94 km which was driven within 97 minutes (See Table 3)

Results

It was possible to operate the engine in GCI mode for each of the 12 speed/load points shown in Figure 2.

Figure 6 shows a contour plot of ISFC over the whole map. There is a broad region at mid and high load where ISFC is around 200g/kWh or even lower. This represents a very

FIGURE 6 Indicated Specific Fuel Consumption over the speed/load map.

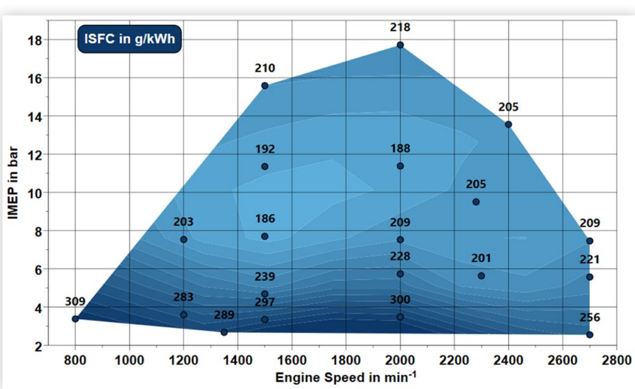
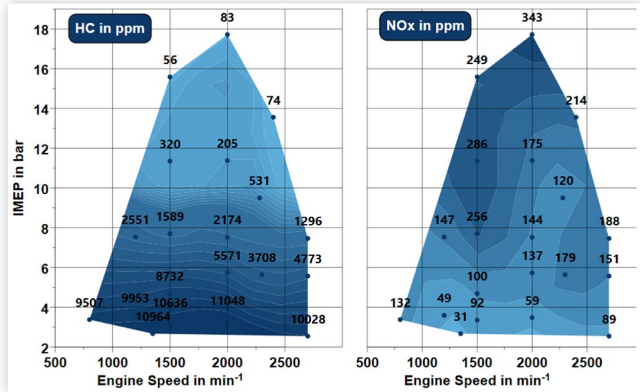


FIGURE 7 HC and NOx emissions over the speed/ load map.



efficient combustion system for gasoline. However, the efficiency is relatively poor at low load where spark assist is employed.

The corresponding engine-out HC and NOx emissions over the map are shown in Figure 7. The very high level of HC emissions are consistent with the poor efficiency in the low load regions.

By taking into account the friction (Figure 4), a fully warmed BSFC map can be calculated. (Figure 8)

Since the GCI engine's part-load fuel consumption in the region below 4 bar BMEP is impaired due to poor combustion efficiency and late combustion phasing, it was instructive to also investigate a four-cylinder engine configuration with cylinder deactivation as a means to reduce the probability of the engine running at low engine loads and thus to improve the cycle fuel consumption. Figure 9 shows the improvement achieved when operating the engine with cylinder deactivation at low engine loads. For example, at 1500 rpm and 4 bar

FIGURE 8 Brake Specific Fuel Consumption over the speed/load map.

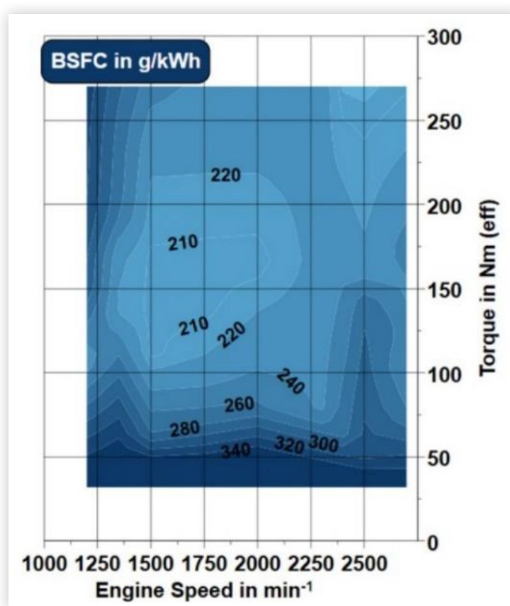
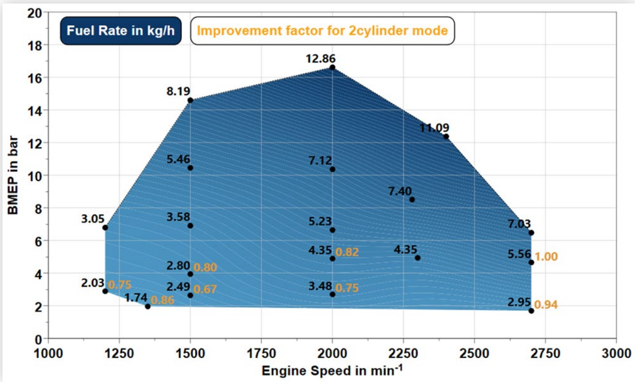


FIGURE 9 GCI fuel consumption in 4-cylinder mode and improvement factor for cylinder deactivation mode.



of BMEP, the improvement factor is 0.8, which means that the fuel consumption in (two) cylinder de-activation mode is 0.8 times the value in 4-cylinder mode, which corresponds to a fuel consumption reduction of 20%. It needs to be stated though, that for the vehicle simulation with cylinder deactivation, a very simple engine operation mode selection approach based only on the engine speed and load has been implemented, where the two-cylinder mode is active at engine speeds higher than 1000 rpm and engine loads below 65 Nm.

The simulation output for the fuel rate (in kg/h) and the engine out NOx emission rate (in kg/h) for the NEDC can be found in the Appendix (Figures A2–A4) for 4-cylinder GCI mode (with and without deactivation mode) as well as the TGDI reference vehicle discussed in the "Longitudinal Vehicle Simulations" section. The performance of the GCI engine relative to the reference vehicle is very load dependent. Figures A2–A4 also show that the engine-out NOx is much lower from the GCI vehicle, than the TGDI reference, although the NOx at λ=1 in the TGDI can easily be treated with a three-way catalyst.

Figure 10 shows the integral results for the three drive cycles investigated for the GCI driven vehicle being operated with and without cylinder de-activation. The NEDC results reveal what has been widely discussed in the previous parts of this work: because of the quite poor fuel consumption of the GCI engine at part load, the average fuel consumption in the NEDC is spoiled by the poor low-load efficiency. If the GCI engine is operated in cylinder deactivation mode, the real advantage of the GCI concept becomes much more apparent and the cycle fuel consumption is drastically reduced.

The picture is slightly different for the WLTC, which tends to be operated at much higher engine loads. Even though the "standard" GCI engine is operated at loads lower than 4 bar of BMEP, at the end the poor part-load fuel consumption is more than compensated by the engine running longer intervals at high loads, so that the 4-cylinder GCI engine shows a clearly reduced fuel consumption compared to the NEDC. If the engine is additionally operated in cylinder deactivation mode, the fuel consumption shrinks to less than 5 l/100km. In the RDE cycle, the same phenomena result in a fuel consumption behaviour which is comparable to the WTLC results, albeit slightly better.

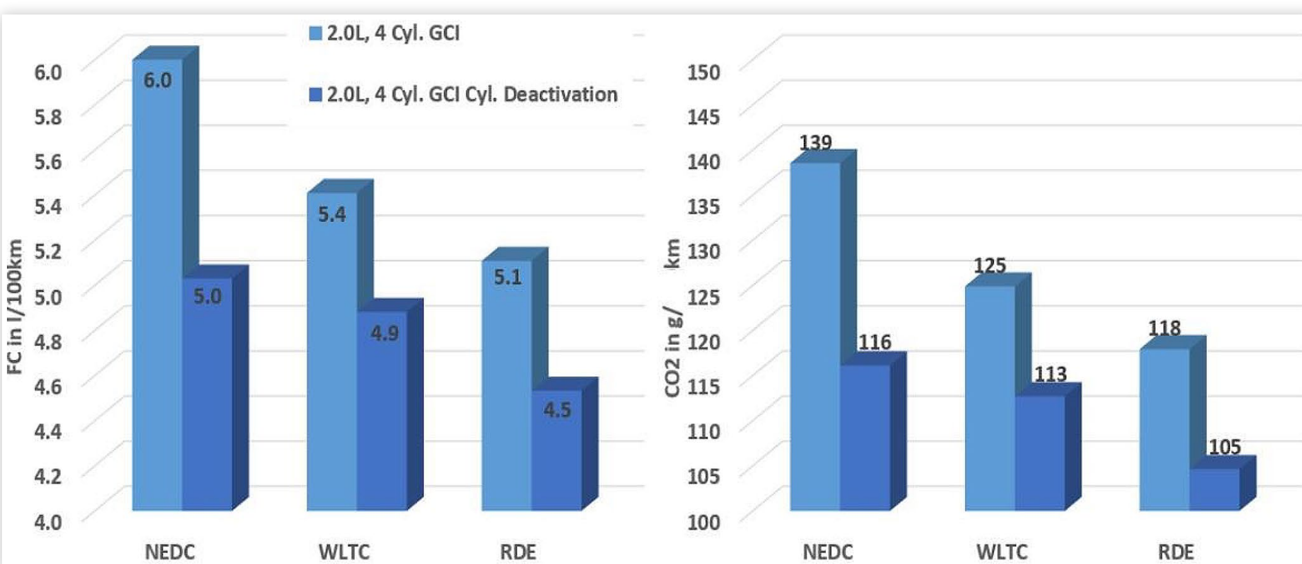
A benchmarking of a possible GCI vehicle configuration by IAV shows that GCI with cylinder de-activation tends to have a lower fuel consumption than SI vehicles running on gasoline and is comparable to CI diesel vehicles.

Conclusions

A practical Gasoline Compression Ignition (GCI) concept has been developed that works on a diverse range of market gasolines (including standard European ULG 95) over a wide speed load range, including low load. This represents a significant achievement.

The advantage of having combustion assistance at low load is that it removes the necessity to have a complex cam

FIGURE 10 Total cycle fuel consumption and specific CO_2 emissions of the investigated engine concepts.



profile to control the valve train mechanism and generally makes the combustion system more robust against differences between gasoline fuels that are likely to be encountered in the marketplace. The combustion phasing is controlled by the injection timing, and the spark only serves to assist the combustion process. In this respect it is different to Mazda's SPCCI concept in which the combustion phasing is controlled by the spark and the spark assist operates over a much wider load range.

The engine was calibrated on a single gasoline fuel, but, based on the findings of ref [21] that calibration allows operation on a range of gasoline fuels, representing RON levels from 92 to 98, as well as summer and winter grade volatility levels and differing ethanol content.

The approach is shown to be able to achieve exceptionally low fuel consumption over a wide area of the speed load map, with ISFC values of 200g/kWh or lower (i.e. equivalent to diesel efficiency). However, the efficiency deteriorates at low load where the combustion assistance operates.

In this work, a detailed, dynamic longitudinal simulation model was created that allows accurate CO_2 emission and fuel consumption predictions for a C-segment vehicle in the NEDC, WLTC and RDE driving cycles in accordance with the results of a dynamometer test. Modeling and simulation were done in the GT-Suite simulation environment. The vehicle data and emission curves used for the model calibration have been taken from the IAV knowledge database and engine test bench and chassis dynamometer measurements. Inputting the experimental ISFC map from the single cylinder bench engine into the model, allows an assessment to be made of the cycle-based fuel consumption and emissions of a vehicle with a GCI technology engine.

Because of the relatively poor efficiency at lower loads, it is not surprising that the nature of the cycle has a significant impact on the fuel consumption: Because the NEDC features a relatively high proportion of low load driving, the fuel consumption is much higher than the WLTC, which in turn gives a much higher fuel consumption than a simulated RDE cycle. De-activation of 2 cylinders makes a significant improvement to the fuel economy by reducing low load operation. A benchmarking of a possible GCI vehicle configuration shows that GCI with cylinder de-activation tends to have a lower fuel consumption than SI vehicles running on gasoline and is comparable to CI diesel vehicles. Hybridization would be expected to further improve efficiency, because low load operation could be further reduced. A next step in this work would be a full vehicle demonstration, where issues such as oxidation catalyst light-off can be investigated. further

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Acknowledgments

This project has received funding under the European Union’s Horizon 2020 research and innovation programme under specific agreement No SA/CEN/RESEARCH/EFTN000/2014-13. The authors would also like to thank Mr Tim Kirchhoff for his contribution to the IAV modelling activity.

Glossary

CI - Compression Ignition

BMEP - Brake Mean Effective Pressure

BSFC - Brake specific fuel consumption

GCI - Gasoline Compression Ignition

GDCI - Gasoline Direct-injection Compression Ignition

EGR - Exhaust Gas Recirculation

ISFC - Indicated Specific Fuel Consumption

IMEP - Indicated Mean Effective Pressure

HCCI - Homogenous Charge Compression Ignition

LTC - Low Temperature Combustion

NEDC - New European Drivecycle

PPC - Partially Premixed Combustion

FEMP - Friction Mean Effective Pressure

RDE - Real Driving Emissions

RON - Research Octane Number

SACI - Spark Assisted Compression Ignition

SI - Spark Ignition

SPCCI - Spark Controlled Compression Ignition

TGDI - Turbo Gasoline Direct Injection

WLTC - Worldwide Harmonized Light-Duty Test Cycle

Appendix

TABLE A1 Calibration parameters at 12 operating points

ref point	RPM [min-11	IMEP [bar]	Pinj [bar]	soi_1 [cad]	eoi_1 [cad]	soi_2 [cad]	eoi_2 [cad]	soi_3 [cad]	eoi_3 [cad]	soi_4 [cad]	eoi_4 [cad]	spark timing [cad]	EGR rate [%]	fuel flow [kg/h]
1	800	3.5	398	−14.6	−11.0	2.6	5.2					1.0	0.4	0.4
2	1200	3.2	398	−12.2	−7.3	1.2	6.1					5.5	8.1	0.4
3	1350	2.5	398	−12.2	−7.1	1.2	5.2					4.0	21.3	0.5
4	1500	3.5	400	−12.2	−5.4	1.2	6.0					4.0	7.1	0.6
5	1500	4.4	399	−12.2	−5.6	1.2	6.4					10.0	10.4	0.7
6	1500	7.3	449	−14.8	−7.9	7.2	13.3					8.0	6.7	0.9
7	1500	11.2	498	−22.8	−16.6	−5.4	−0.7	6.3	10.7	12.4	16.7		8.1	1.3
8	1500	15.4	567	−20,4	−16.6	−5.6	−1.5	0.1	7.5	8,4	12.7		7.7	2.0
9	2000	3.6	399	−14.6	−6.4	2.4	10.0					6.5	3.9	0.9
10	2000	11.2	498	−19.4	−11.4	−1.8	3.6	9.6	14.8	16.0	21.2		9.1	1.8
11	2280	9.7	499	−21.6	−11.6	−4.0	1.9	6.5	12.4	13.8	19.8		10.1	1.9
12	2400	13.5	498	−20.4	−14.3	−8.6	−2.4	−0.9	8.8	10.2	16.6		7.3	2.9

FIGURE A1 Comparison between measured and simulated CO2 traces of the base 1.4 Litre GDI engine (cumulated as well as instantaneous) for the NEDC and WLTC cycles.

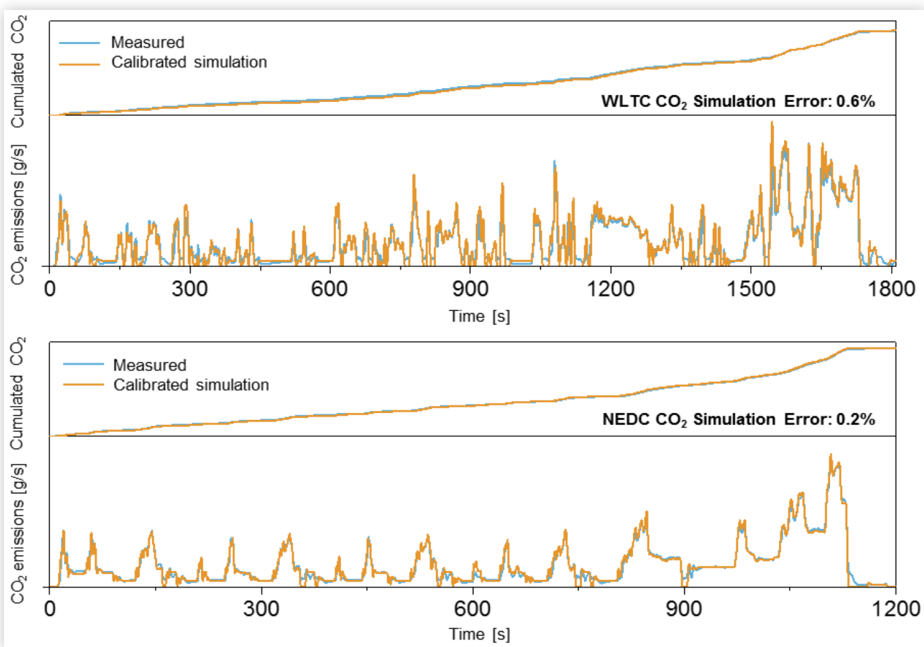


FIGURE A2 Simulated fuel and NOx mass flow rates of different engine concepts in NEDC

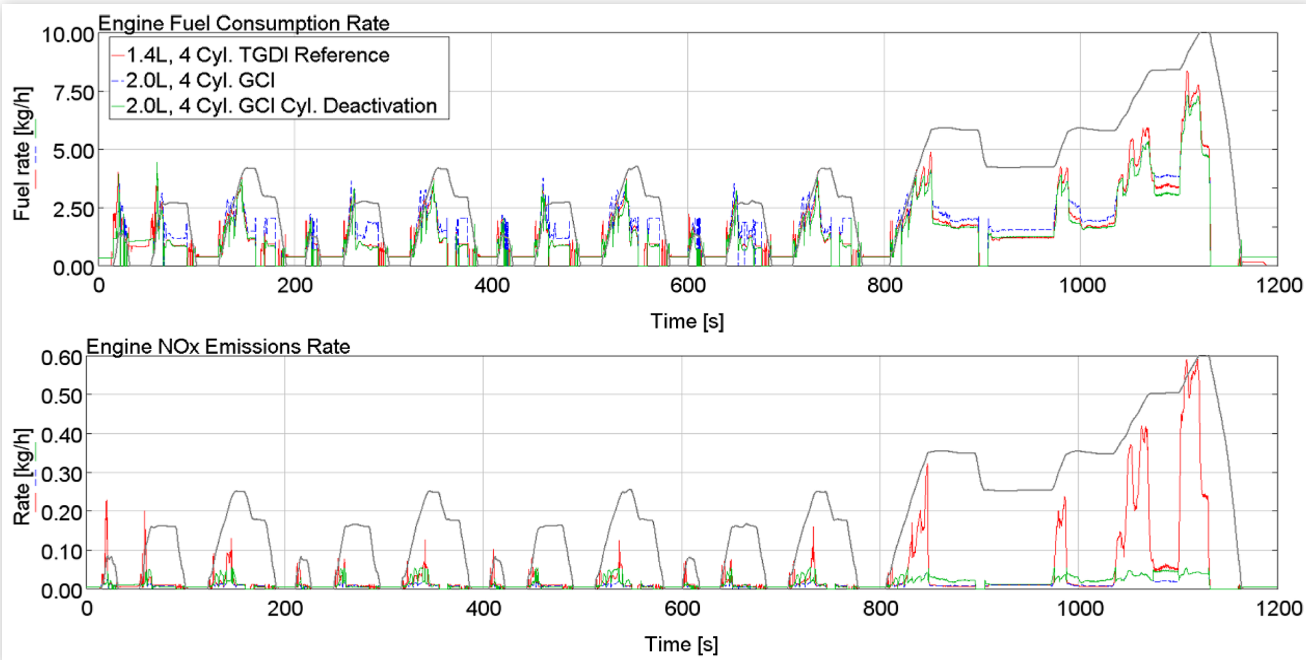


FIGURE A3 Simulated fuel and NOx mass flow rates of different engine concepts in WLTC

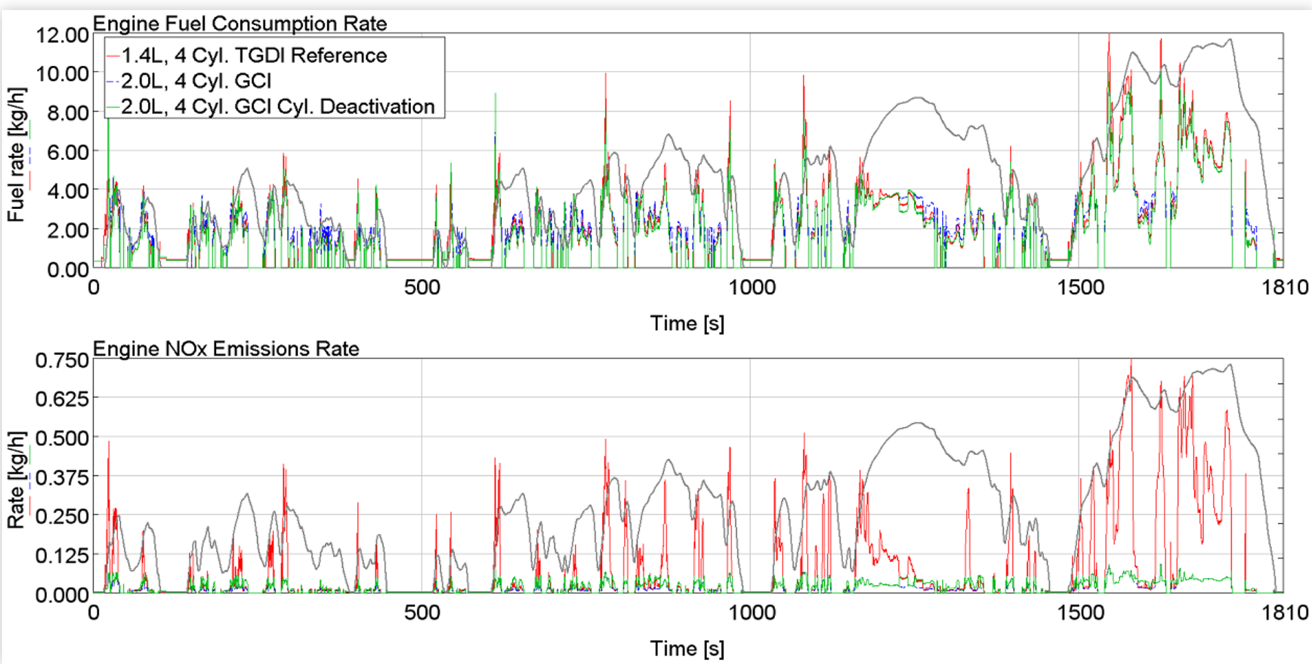
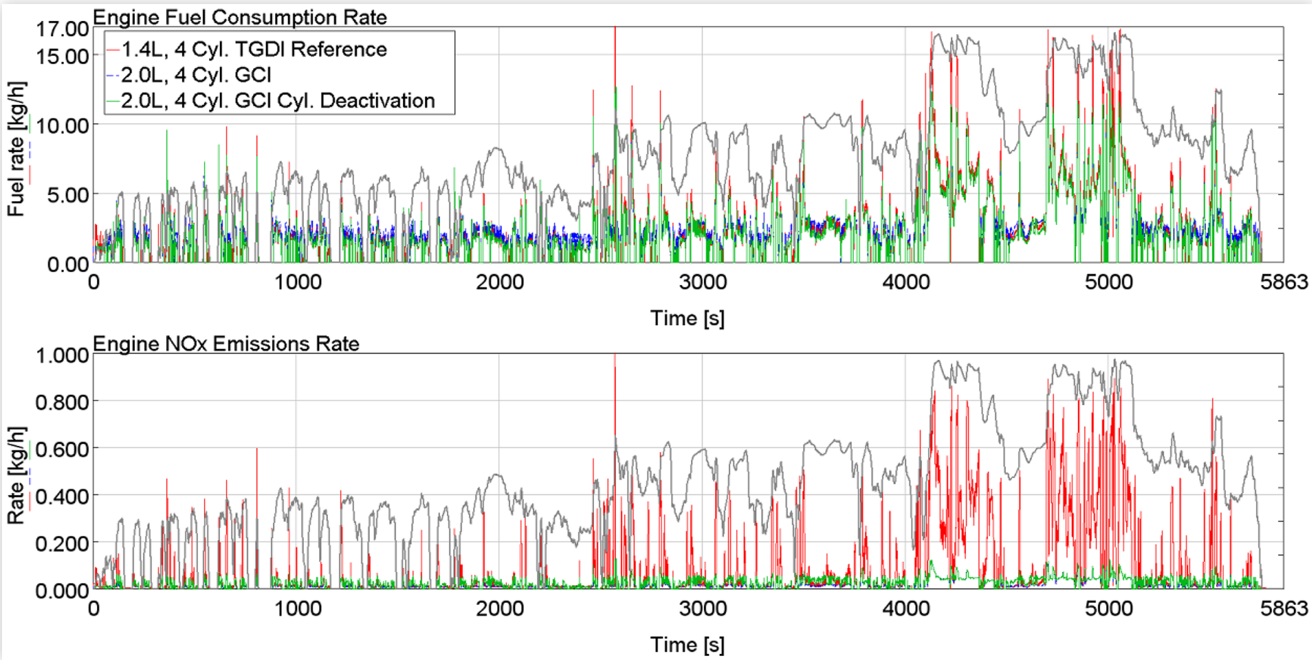


FIGURE A4 Simulated fuel and NOx mass flow rates of different engine concepts in RDE



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ISSN 0148-7191