Downsizing of Gasoline Engine: an Efficient Way to Reduce CO2 Emissions

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Downsizing of Gasoline Engine: an Efficient Way to Reduce CO_2 Emissions

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Résumé — Écosuralimentation des moteurs à essence : une voie efficace pour réduire les émissions de CO_2 — En vue d'atteindre les engagements pris pour 2008 en matière de réduction des émissions de CO_2 de l'ensemble du parc de véhicules, la R&D moteur explore différentes solutions. Du point de vue des émissions de CO_2, les moteurs à essence souffrent d'un handicap en comparaison avec des motorisations Diesel. La réduction de la taille des moteurs à essence *(downsizing)* est une voie prometteuse pour améliorer le rendement des moteurs et fait donc l'objet de recherches intensives. À long terme, l'objectif pourrait être une réduction de moitié de la cylindrée des moteurs.

Des résultats issus de calculs de simulation véhicule montrent que, même un *downsizing* aussi poussé que celui-ci ne sera pas suffisant pour amener l'ensemble du parc de véhicules à essence aux niveaux de CO_2 visés. Cela sera tout juste suffisant pour atteindre les objectifs de 2008 pour un véhicule du segment moyen, motorisé par un petit moteur de 0,8 l en remplacement d'un moteur à essence 1,6 l d'aujourd'hui. La réduction des émissions de CO_2 dans un tel cas est de 18 % dans des conditions de fonctionnement moteur chaud. Des améliorations supplémentaires doivent également être obtenues en ce qui concerne les consommations spécifiques des moteurs à essence, notamment pour les véhicules plus gros.

L'*IFP* a développé une approche innovante en couplant deux technologies, bien placées en termes d'économie d'énergie, que sont l'injection directe d'essence et le turbocompresseur avec une redéfinition des lois de distribution des soupapes. Ceci a été testé sur un moteur de 1,8 l. Un balayage de la chambre de combustion, non conventionnel sur des moteurs turbocompressés, devient alors possible. La résistance au cliquetis du moteur et son rendement volumétrique sont améliorés. Les interactions fines entre la combustion, le turbocompresseur et la distribution ont été analysées et optimisées. On obtient une augmentation significative des performances spécifiques du moteur en termes de couple et de puissance, tout en conservant de bons niveaux de consommation spécifique sur l'ensemble de la plage de fonctionnement moteur.

Le couple spécifique, atteint à 1250 tr/min, a été progressivement accru de 50 % et est aujourd'hui proche de 1,7 MPa de PME, alors que la puissance maximale est de 83 kW/l avec une consommation spécifique d'environ 300 g/kWh. Ce type de moteur 1,8 l se présente comme un compétiteur intéressant pour remplacer les actuels gros moteurs à essence de 2,5 à 3 l de cylindrée.

***Abstract* — *Downsizing of Gasoline Engine: an Efficient Way to Reduce CO_2 Emissions* —** *In order to meet commitments in terms of vehicle CO_2 emission reduction for the whole fleet of cars for the year 2008, engine research and development is today exploring several fields. From CO_2 point of view, gasoline engines suffer from an handicap in comparison to Diesel engines. Reduction of size of gasoline* *engine* (downsizing) *appears to be a promising way to improve engine efficiency and is subject to extensive research. Having a look to the long term, the aim should be to reduce by half the engine displacement volume.*

Calculation results from a vehicle simulation illustrate that even a so extensive downsizing will not be enough to bring the entire gasoline fleet to the requested CO_2 levels. It would just be sufficient to reach the targeted levels for year 2008 for a mid-class vehicle powered by a downsized 0.8 l engine instead of a current 1.6 l gasoline engine. Reduction of CO_2 emission is in that case about 18% in warm engine conditions. Then, further improvements have to be achieved in terms of gasoline engine specific fuel consumption, especially for bigger cars.

IFP *has developed an innovative approach combining two energy saving technologies that are gasoline direct injection and turbocharger with renewed definition of valve timing. This has been applied on a 1.8 l engine. An unconventional combustion chamber scavenging process becomes then feasible on a turbocharged engine. Knock resistance and volumetric efficiency are improved. The close interactions between combustion, turbocharger and valve timing have been analysed and optimised. Result is a significant increase in specific engine output in terms of torque and power, while keeping low specific fuel consumption level over the whole range of engine running conditions.*

Specific torque obtained at 1250 rpm has been progressively increased by 50% and is today close to 1.7 MPa BMEP while maximum power is now 83 kW/l with a specific fuel consumption of about 300 g/kWh. This type of 1.8 l engine would be an interesting competitor to downsize current big gasoline engines with displacement from 2.5 to 3.0 l.

INTRODUCTION

Because it is a major cause of global warming, the concentration of carbon dioxide (CO_2) in the air is today of great concern. Transport represents 20 to 25% of the CO_2 release in the atmosphere and this share tends to increase [1]. A part of the automotive industry has taken into account the absolute necessity to reduce the CO_2 emission of the vehicles.

The *European Car Manufacturer Association (ACEA)* has for example entered into an highly ambitious undertaking: the commitment is that CO_2 emission of the future vehicles —averaged on the whole production of the signatories—will reach: 140 g/km of CO_2 in year 2008, and perhaps 120 g/km of CO_2 in year 2012.

Fuel consumption and CO_2 emission of a vehicle are two indissociable parameters. These two targeted levels correspond to very low fuel consumption in comparison to current vehicles *(Table 1)*. In the year 2000, *ACEA* average new vehicle CO_2 emission was 169 g/km, with respectively 177 g/km for the gasoline fleet and 157 g/km for the Diesel vehicles [2]. Measurements are made on the normalised New European Driving Cycle (NEDC), which includes a cold start. The 120 g/km level is for example the CO_2 emission of today's very small car called Smart, produced by *MCC Company*. It is a 2-seats, total mass of 720 kg vehicle, powered by a 3-cylinder turbocharged engine. The possible target is then to reach this drastic low fuel consumption value for the whole fleet perhaps in year 2012.

Reduction of CO_2 emission of the vehicle will essentially be achieved thanks to an increase in efficiency of the engine and of the gear. Of course, other features of the vehicle may be improved such as aerodynamic drag, mass, resistance of the tires… but to a lesser extend.

TABLE 1

CO_2 emission and fuel consumption
Current levels and targets for the future

Current vehicle or target for the future	**CO_2 emission (g/km)**	**Gasoline fuel consumption (l/100km, NEDC)**
ACEA average level for the cars produced during the year 2000	169	6.9
MCC Smart 0.6 l turbo gasoline engine	120	4.9
Target for year 2008*	140	5.7
Possible target for year 2012*	120	4.9

* averaged on the whole fleet of passenger cars.

Several ways are today explored by the engine researchers for the reduction of the fuel consumption of the engines. As far as the gasoline engines are concerned, the tested technologies are for example:
- stratified combustion thanks to the development of in-cylinder direct injection technology;
- variable valve process, from simple variable timing camshaft up to fully electronic control of the valves (*camless* engine);
- variable compression ratio;

– reduction of engine size, called downsizing,
– hybridation of the thermal engine with an electric one.

Downsizing is today considered as a promising way to increase fuel economy with a good cost to benefit ratio. The challenge is here to reduce the engine displacement volume while keeping the same performance in terms of torque and power than the initial larger engine, and simultaneously to ensure an improvement in engine efficiency.

Downsizing of gasoline engine is already an industrial reality. During last years, several car makers have presented 1.8 l to 2.0 l turbocharged engines. The performances of these engines are typically the ones of naturally aspirated engines with 2.5 l displacement. The reduction of fuel consumption is typically about 10%. The second generation of downsized engines is today the object of extensive research. Target is to reduce by half the displacement of the engines and also to consider the downsizing of smaller engines than the upper class engines with 2.5 l displacement or more.

Thanks to its know-how either in the field of gasoline direct injection (GDI) and also in the engine air charging area, *IFP* has developed an innovative approach combining both technologies that allows very high specific performances and increased efficiency.

After having presented the reasons why the downsizing of gasoline engine reduces the CO_2 emission of the vehicle, the paper will provide results of vehicle simulation that illustrates the absolute necessity to improve the specific fuel consumption level of the engine. Simple downsizing of today's engine is not sufficient keeping in mind the drastic targets of the next years. Progress still have to be made in terms of in-cylinder combustion process and efficiency. In a third stage, results from engine test realised on an *IFP* prototype engine will be presented showing the gradual and constant improvement achieved on this engine demonstrator during the last months in terms of specific performances and specific fuel consumption.

1 DOWNSIZING PRINCIPLE

Most of the time, and especially when the vehicle is driven at a constant speed, the engine is run under low load conditions. This leads to a poor engine efficiency especially for conventional existing gasoline engine for which load is controlled by a throttle. Throttling generates pumping losses and reduces efficiency. For example, typical power required to drive a mid-range car at a constant speed of 70 km/h is only about 7 kW. Considering an engine with a displacement of 2 l for example, these 7 kW represent only a very low load of 0.21 MPa BMEP[1] if the engine is run at 2000 rpm. Figure 1 shows a representative specific fuel consumption (SFC) map of current conventional gasoline engine. An engine functioning at BMEP = 0.21 MPa/2000 rpm is

(1) BMEP: brake mean effective pressure.

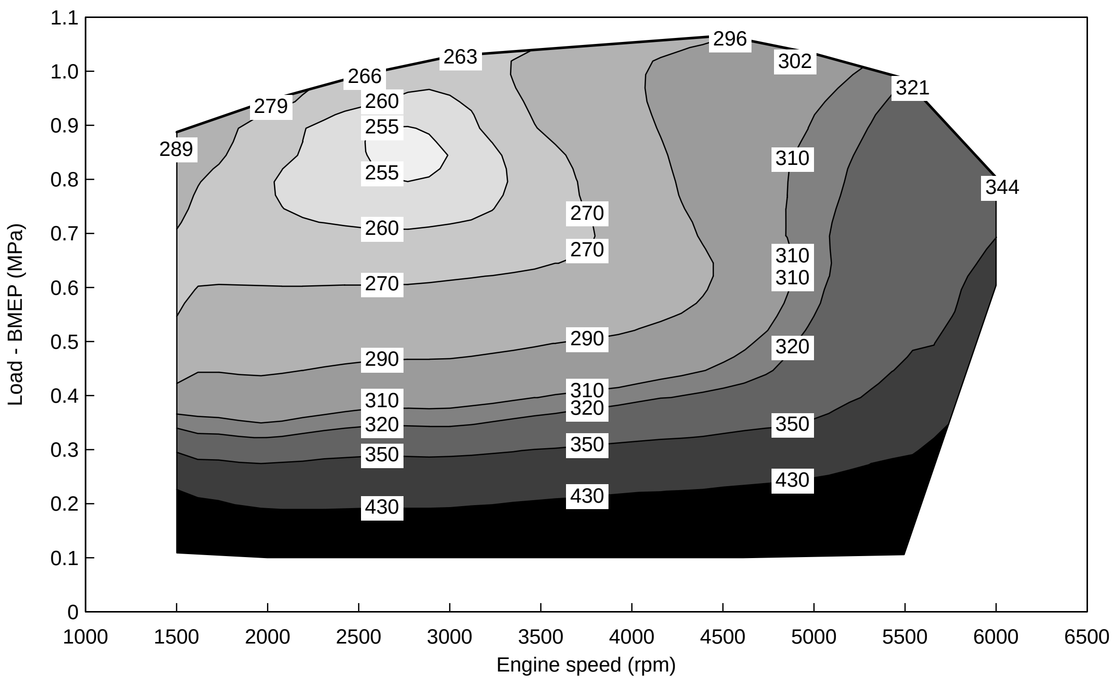


Figure 1

Typical specific fuel consumption map of a conventional gasoline engine (g/kWh).

typically close to a SFC of 400 g/kWh. If the engine is a 1 l total displacement, these 7 kW are produced with a load of 0.42 MPa and a SFC of about 300 g/kWh. For the same vehicle at the same 70 km/h constant speed, the use of the small engine represents a reduction in CO_2 emissions of 25%.

As a matter of fact, high loads engine running are required especially in transient operation, when accelerating the vehicle. This is to overcome inertia of the vehicle, either vehicle mass but also inertia of all rotating parts. Rotating inertia can represent up to an equivalence of 25% increase in mass of the vehicle when it is operated in first gear. High torque capability ensures better vehicle driveability. In order to be accepted by the customers, reduction of engine size must be invisible from driver's point of view. The specific output performance of the small engine must then be increased by a ratio equal to the reduction of engine displacement. Considering a typical maximum BMEP of naturally aspirated gasoline engine of 1.2 MPa, a reduced by half of the displacement—which represents today the ultimate downsizing that could be achieved at long term—requires BMEP up to 2.4 MPa for the small engine. That kind of very high specific performance should only be reached with the help of an increase in air and fuel content of the combustion chambers. Therefore, the use of air boosting is unavoidable.

Engine supercharging should be realised using different techniques. In fact, only two of them are really used in the Automotive Industry due to efficiency and production cost reasons. These are:
– the turbocharger;
– the (belt- or gear-) driven supercharger.

The compression of intake air to force the feeding of the engine is an energy consuming process. Equation (1) indicates the power consumption of the air charger for a given air flow consumption of the engine.

$$P_{comp} = \frac{1}{\eta_{comp}} \cdot \frac{dm_{air}}{dt} . C_p T \left[\left(\frac{P_2}{P_1} \right)^{(\gamma-1)/\gamma} - 1 \right] \quad (1)$$

where:

P_{comp} represents the power consumption of the compressor
η_{comp} the thermal efficiency of the compressor
dm_{air}/dt the air mass flow rate
C_p the calorific capacity of air
T air temperature at the inlet of compressor
P_1 the air pressure at the inlet of compressor
P_2 the air pressure at the outlet of compressor
and γ the air polytropic coefficient.

For example, to feed a 1 l displacement engine with a maximum power output of 75 kW the compression of the intake air flow requires about 9 kW itself[2]. To reduce energy consumption of the air charger, it is important to minimize air requirement of the engine and boost pressure P_2.

Because the turbocharger picks up energy from enthalpy of exhaust gases, it is a more energy efficient technology than the driven supercharger. On the other side, the turbocharger suffers from a lack of boost pressure capability at low engine speed and also from turbo lag, that induces a response time before it provides all its capacity. The driven supercharger does not present this second drawback.

The increase in specific performance and the use of high BMEP come with an occurrence of engine knock phenomenon. The higher the load, the higher the risk of self ignition of the air/fuel mixture before the last fractions of the combustion chamber have been reached by the flame front propagating from the spark plug. One solution to avoid the trouble is to lower the compression ratio of the engine. But this would results in a reduction of combustion thermal efficiency, thus partly destroying the pursued target. One challenge of the downsizing is to find innovative approach to increase maximum BMEP while containing knock propensity of the engine.

As shown before, downsizing of gasoline engines represents a promising way to reduce CO_2 emissions thanks to a better use of the engine, with running conditions closer to the best efficiency area. The downsizing of the engine could also have an induced positive impact onto the engine itself but also on the whole vehicle. As far as the engine is concerned, the smaller the engine, the less the friction losses. A smaller engine will also be lighter, thus reducing total vehicle inertia. Taking a more long-term view, the reduction of the external dimensions of the engine will allow more flexibility for the car body definition with perhaps better aerodynamic profile for the hood. These two last points have to be confirmed in the future after having taken into consideration the supplementary mass and space required for the whole air charging system, including the air cooler, and aerodynamic drag of the latter.

2 SIMULATION OF A DOWNSIZED ENGINE APPLICATION

2.1 Background

The calculation results presented here below illustrate the potential of the downsizing applied to an existing mass produced European car, referred hereafter has "reference car", representative of the mid-range class, powered by a 1.6 l gasoline engine *(Table 2)*.

The simulation tool used for the calculation is *IFP-SIMCYC* code, based on equations of longitudinal dynamics behaviour of vehicles. In a first step, the code parameters has been tuned in order to fit on the existing data of fuel

(2) Hypotheses: $P_2/P_1 = 2$, air mass flow = 350 kg/h, $\eta_{comp} = 0.7$.

consumption measured onto the real vehicle. These fittings have been made considering several vehicle running conditions (constant speeds and driving cycles), all for warm conditions.

TABLE 2

Main features of reference existing car for the simulation

	Reference Car
Fuel consumption (l/100 km)	
NEDC	6.8
NEDC (warm)	6.6
Mass (kg)	1095
SCx (m^2)	0.67
Engine features:	
type	In-line 4 cyl., spark ignition, 16 valves
displacement (l)	1.600
compression ratio	10.0:1
max. power	79 kW (49.4 kW/l)
max. torque	150 N·m (93.9 N·m/l)
Gear box ratios:	Vehicle speed (km/h) at 1000 rpm:
1st gear	8.1
2nd gear	14.7
3rd gear	20.7
4th gear	26.5
5th gear	33.3

2.2 Hypotheses

The simulation consists in substituting the 1.6 l engine by a smaller one of 0.8 l displacement while keeping all other parameters of the whole vehicle constant (maximum power, torque, vehicle mass, rotating parts inertia, SCx, gear box ratio, etc.). At this step, no assumptions is made with regard to the type of engine architecture (3 or 4 cylinders, turbocharger or driven air compressor, etc.). The main hypothesis concerns the SFC map of the small engine. This map has to be extrapolated from existing maps up to extremely high loads not encountered on today's engines.

For the low- to mid-load running points (0 to 1.0 MPa BMEP), the finally retained map is based on real SFC levels of modern, small displacement naturally aspirated engines. For higher loads, corresponding to supercharged running conditions, the SFC levels have been chosen with respect to *IFP* data base considering a conventional approach of turbocharged engines. The final assumption in terms of SFC of the small simulated engine is presented in Figure 2.

In terms of driving conditions, simulations are made considering:

– Official NEDC cycle, based on a urban phase (ECE) and extra-urban subcycle (EUDC). Average vehicle speed on ECE part is 19 km/h, with 3 vehicle stops per kilometer; on EUDC part, it reaches 63 km/h.

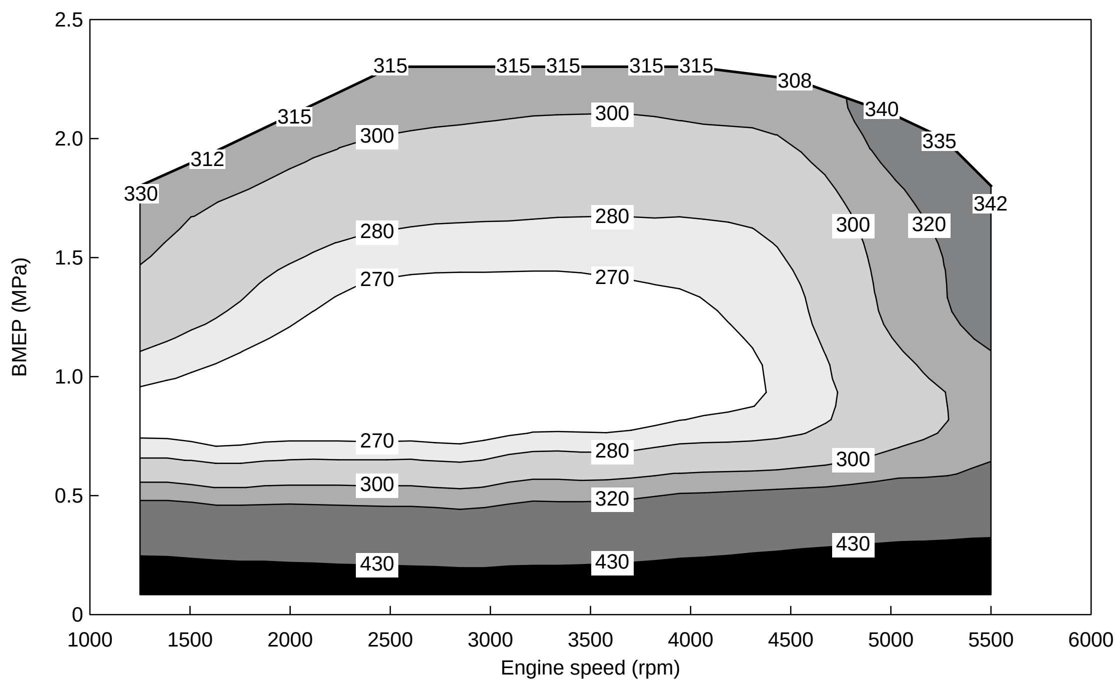


Figure 2

SFC map assumption for the small 0.8 l engine (g/kWh).

– Three other driving cycles, more representative of real life vehicle use, with a lot of transient operations and, as far as the urban cycle is concerned, 5 stop and go phases per kilometer, hereafter referred as “urban”, “road”, “motorway” *(See Appendix)*. Average speed of these three cycles are respectively 18, 58 and 100 km/h.
– An extreme vehicle running condition, corresponding to a constant vehicle speed of 150 km/h.

N.B.: All calculations have been made considering warm cycle and not cold engine starting.

2.3 Results

Results of the calculations are presented on Figure 3. Use of small 0.8 l instead of genuine 1.6 l engine for powering the reference car, all other parameters being constant, leads to a reduction in vehicle fuel consumption: the less the average speed of the vehicle, the higher the fuel consumption benefit. On standard warm European driving cycle, the average fuel consumption for the entire cycle is about 5.4 l/100 km, corresponding to a 18% decrease with regard to the reference car.

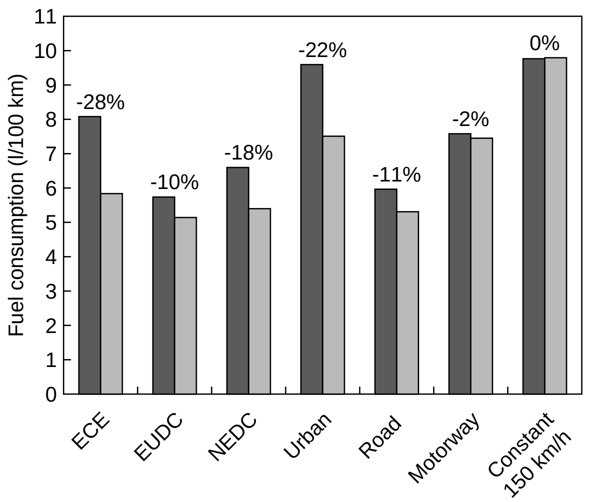


Figure 3

Calculation results: vehicle fuel consumption for different warm driving cycles (l/100 km) - Reference 1.6 l car (dark bar chart) in comparison to downsized 0.8 l engine in same car (pale).

Going into the details, the fuel benefit reaches up to 28% during standard urban cycle. Considering the more realistic urban cycle, the real fuel consumption improvement should be closer to 22% than the 28% obtained on the official cycle.

At higher vehicle speed, benefit is reduced: 10% improvement on extra-urban official cycle, to be compared to 11% for real life road conditions. When the engine is forced to run at high speed, high load, the downsizing effect disappears. Only a small 2% fuel consumption reduction should be reached during motorway use and even a slight overconsumption may be encountered at a constant 150 km/h speed.

2.4 Synthesis

Calculations presented here before show that the extreme downsizing—*i.e.* engine displacement being reduced by half—of a typical today’s mid-range 1.6 l gasoline car will lead to a fuel consumption level in the order of the targeted values for year 2008. This result is obtained considering favourable conditions and especially warm start cycle conditions, and high low end specific torque capability of the small engine.

The commitment made for year 2008 is applicable on the entire fleet. In this fleet, large vehicles with big engines will be major contributors. It is clear that the fuel consumption levels reached during the simulations are not sufficient enough and that the benefit must be higher in order to be compatible with the future targets.

As already explained, other contributors to vehicle fuel consumption will be improved during next years. But the most efficient way to improve vehicle fuel economy is to increase engine and gear efficiency.

As far as the engine is concerned, this means improvement of SFC map. For this, *IFP* has developed an innovative approach based on its experience in the fields of gasoline direct injection and turbocharging of gasoline engine.

3 OPTIMIZATION OF SPECIFIC ENGINE PERFORMANCES: EXPERIMENTAL RESULTS

3.1 Objective

The objective of the work presented hereafter, is to show the capabilities of the coupling of two techniques which are gasoline direct injection and turbocharger with innovative approach in terms of combustion process to reach extremely high specific performances while improving specific fuel consumption of the engine.

The target is to achieve respectively specific output and torque greater than 80 kW/l and 175 N·m/l. This last value correspond to a 2.2 MPa BMEP. In comparison, today’s naturally aspirated gasoline engine present 40 to 50 kW/l and 100 N·m/l performances.

3.2 Engine and Test Description

The considered engine for the turbocharging application presented here has been developed by *IFP* on the basis of *Renault* IDE 2L mass produced engine [3]. It has been adapted to turbocharging, for homogeneous operations. The full load test conditions are as follows:

– relative air/fuel ratio is fixed to 1.0 while upstream turbine temperature in the two scrolls is below 980°C;
– exhaust back pressure is fixed at 40 kPa at maximum power;
– intake air temperature is regulated at 25°C +/–1°C and hygrometry at 38% +/–12%;
– air temperature in engine intake plenum is regulated at 50°C +/–1°C by means of a liquid-cooled intercooler;
– injectors and ignition system are those of the manufactured engine;
– fuel research octane number (RON) is 95;
– ignition timing is set at 2 CAD before knock limit spark advance at full load.

The main specifications of the engine are listed in Table 3.

TABLE 3

Prototype engine features

Bore × stroke	82.7 × 83 mm
Number of cylinders	4
Displacement	1.783 l
Compression ratio	10.0:1
Injection device	*Siemens Deka* DI XL injector
Turbocharger	Twin scroll turbine housing
Valve timing	Variable

3.3 Principle and First Optimization

The innovative approach combines GDI and turbocharger plus unconventional valve timing. As already seen, the higher the load, the higher the risk of having engine knock. In the case of turbocharged engines, knock trouble has to be taken into great consideration. Otherwise, the occurrence of knock will lead the engineer to lower the compression ratio, resulting in a lack of engine efficiency.

Two main reasons may conduct to knock occurrence:

– a high temperature of the fresh air-fuel mixture during the combustion process before the considered fractions have been reached by the front of the flame;
– presence of residual burned gas fractions from previous combustion, that have a thermal and chemical impact on the mixture thus increasing knock sensitivity.

IFP innovative approach fights knock on these two aspects: for the first point, GDI offers a “cooling effect” as explained hereafter. On the other side, new optimization of turbocharger adaptation, in relationship with renewed valve timing definition offer the opportunity to scavenge the combustion chamber before the new fresh air-fuel mixture is introduced thus pushing out the residual burned gases.

3.3.1 Gasoline Direct Injection

Gasoline direct injection often means stratified operations. This kind of engine operation allows fuel consumption gains at part load due to pumping and thermal loss reduction. Nevertheless, after-treatment of NOx emissions in an oxidising environment leads to a fuel penalty. It is also difficult to carry out this after-treatment especially because of the very low sulfur level required in fuel for NOx traps. Consequently, gasoline direct injection engines do not fully benefit from their high efficiency in running at stratified conditions and consumption gains on vehicles are limited to 10% or 12% [4].

Homogeneous stoichiometric conditions present lots of advantages. After-treatment can be easily achieved without too expensive systems and applications of this combustion mode on current naturally aspirated engines shows high volumetric efficiency and compression ratio in comparison with intake port injection. Gasoline direct injection engine has a lower knocking sensitivity, which is a main advantage in supercharging applications.

3.3.2 Turbocharger Matching and Valve Timing

Scavenging process is only feasible if exhaust valves and intake valves present an overlap period during which both of them will be slightly opened *(Fig. 4)*. This could occur at engine top dead center (TDC), during end of closure of exhaust valve and begining of intake valve opening. Moreover, in order to perform a scavenging air flow from intake side of the engine to the exhaust side, it is mandatory that during valve overlap, instantaneous intake pressure is larger than exhaust pressure.

Considering conventional turbocharged engine approach using port fuel injection, scavenging is unfeasible. A valve overlap will result in a risk of an air and fuel scavenging, with direct appearance of fuel in the exhaust line of the engine, resulting in overconsumption and pollutant emission. Valve overlap is thus strictly avoided on existing turbocharged engines. Interest of GDI is here to be able to scavenge only with fresh air, and wait the total closure of the exhaust valves before starting fuel injection.

Considering instantaneous pressure situation at TDC, the use of new turbocharger technology called “twin scroll turbo” is an advantage *(Fig. 4)*. One of the main problems of the four cylinder supercharged engine is cylinder interactions in the exhaust manifold. The pressure wave emitted by one cylinder in the early stage of the exhaust stroke increases the back pressure of another cylinder in the later stage of the exhaust stroke. The residual gas fraction in the cylinder is increased with a negative effect on engine knock resistance. 1D calculations performed with Wave code show a residual gas fraction in the cylinder of about 10% in mass at 1000 rpm and 6% at 2000 rpm with a conventional turbocharger. In the case of a twin scroll

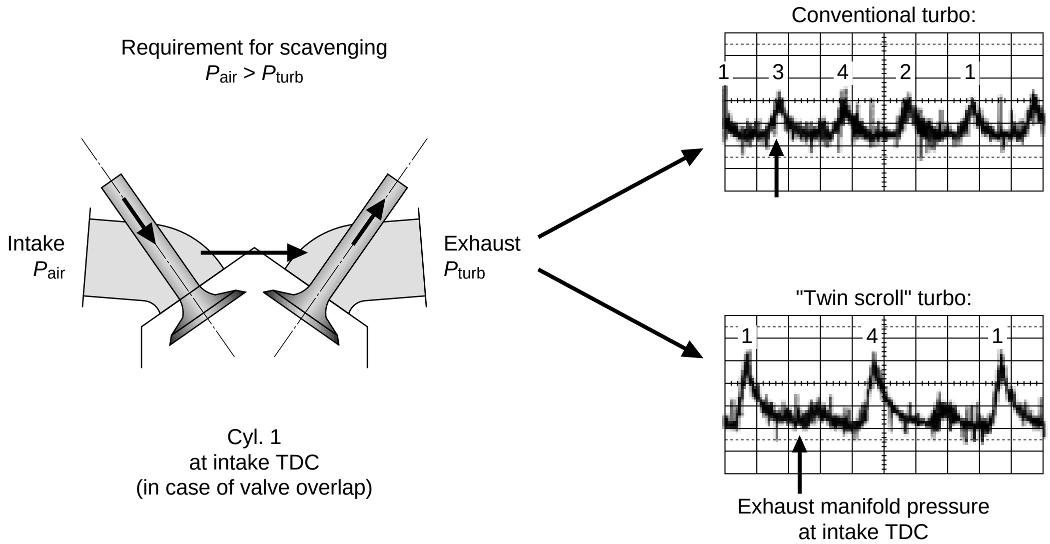


Figure 4

Principle of scavenging process and advantage of twin scroll turbine technology.

turbocharger, the cylinders whose combustions are separated by one engine rotation are connected to the same scroll of the turbine housing. Interaction between cylinders are thus drastically reduced because there is no communication between the two exhaust-gas flows 1-4 and 2-3. Cylinder emptying is then improved.

Previous *IFP* work on turbocharging applications has confirmed that in the case of very low exhaust-gas flow rate, *i.e.* at low engine speed, turbine efficiency is improved when burned gases are pulsed. On a four cylinder engine, the 1-4 and 2-3 exhaust-gas flow disjunction has the further advantage of preserving gas pulsations up to the turbine, which allows a higher air boost pressure ratio. Twin scroll housing allows an increase of the turbocharger compression ratio as it is shown in Figure 5 where the comparison with a conventional housing was performed at 1500 and 1750 rpm on the same engine.

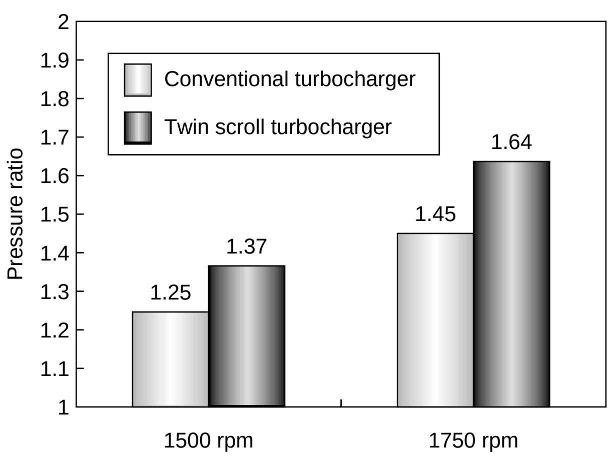


Figure 5

Comparison of compressor pressure ratio with conventional and twin scroll turbocharger - 1500, 1750 rpm, full load.

3.3.3 Valve Timing

Figure 6 shows result from a previous work [5], with same *IFP* turbocharged engine but bigger displacement (2.0 l instead of 1.8 l), comparing output torque level with two intake valve timings at low engine speeds and full load. The first intake valve timing (no. 1) is close to the original naturally aspirated manufactured engine valve timing. The intake valve opening is timed close to TDC and the valve closure for the same lift is timed after bottom dead center (BDC). With the second intake valve timing (no. 2) the intake valve opening is timed before exhaust valve closure, and closure is timed early close to BDC.

Engine performances are higher in the case of valve timing no. 2, which is due to several reasons.

First reason is the early intake valve closure which increases engine volumetric efficiency at low engine speeds. Figure 7 shows a high volumetric efficiency level with valve timing no. 2. Volumetric efficiency takes into account the amount of scavenged fresh air. Turbocharger compression ratios are then lower for a fixed output torque target with such an intake valve timing. These low compression ratio

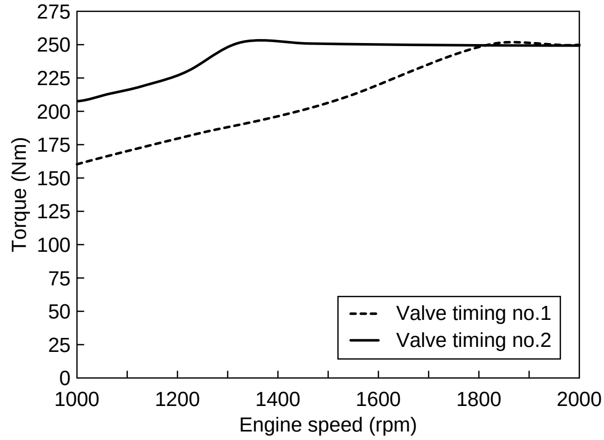


Figure 6

Torque *versus* engine speed at full load for two valve timings.

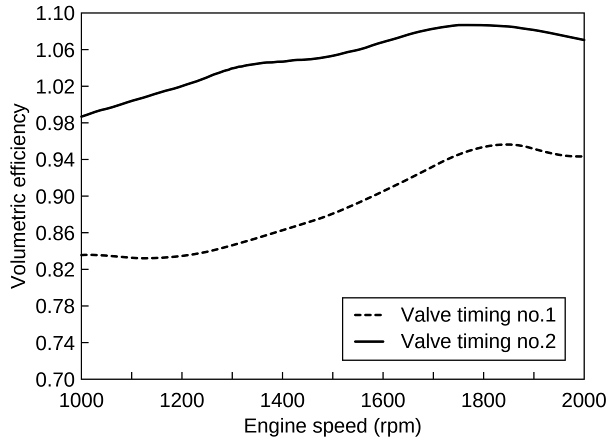


Figure 7

Engine volumetric efficiency *versus* engine speed at full load for the two valve timings.

levels required to reach the output target allow the reduction of the supercharging response time and reduced power consumption of compressor stage.

Moreover, scavenging of a part of the inlet air in the exhaust pipe leads to a fuel enrichment of the in-cylinder charge with a positive effect on engine knock resistance and engine output performances. This is obtained while keeping a stoichiometric close loop control, from exhaust side point of view.

3.4 Further Improvements

The increase in engine specific performances and fuel consumption has been reached in several steps [5, 6]. All these improvements have been made while keeping very low SFC on the entire engine map as shown on Figure 8.

All along this progressive evolution, results always confirmed an intricate influence between the in-cylinder

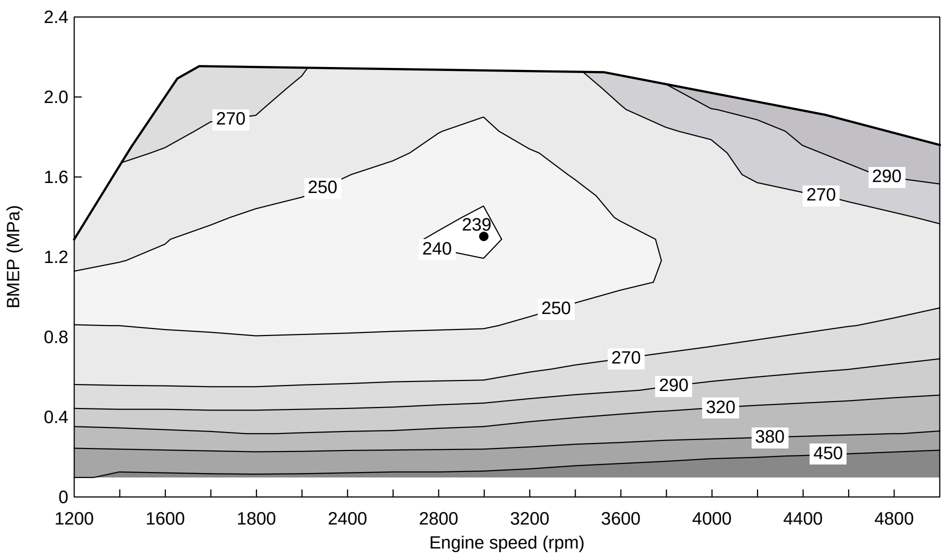


Figure 8

Specific fuel consumption variation within the engine map - Stoichiometric running (g/kWh).

combustion process and the functioning of the turbocharger with close interaction.

The most recent results combine all accumulated knowledge with progressive improvements of low end torque thanks to:

– optimization of valve overlap by simple use of intake valves (improved knock resistance, increased volumetric efficiency, in-cylinder enrichment);
– then having reconsidered the trade off between low end torque and maximum power (see after), reduction of turbine stage size to reduce turbocharger inertia;
– improvement of combustion chamber design thanks to a modification of piston crown leading to a faster and stable combustion;
– increase in valve overlap duration by making use of both intake and exhaust valves. Enlarging valve overlap thanks to the shift of exhaust camshaft appeared to have also a significant impact, with same effect as allready explained for intake camshaft;
– modification of valve event duration to further improve volumetric efficiency.

Low end torque has then been progressively increased by more than 50%. These improvements are summarised in Figure 9.

Because torque at low engine speed and power at high revolution are closely linked by a trade off concerning turbine stage, great care has been taken to ensure that low end torque improvements would not impact maximum power features.

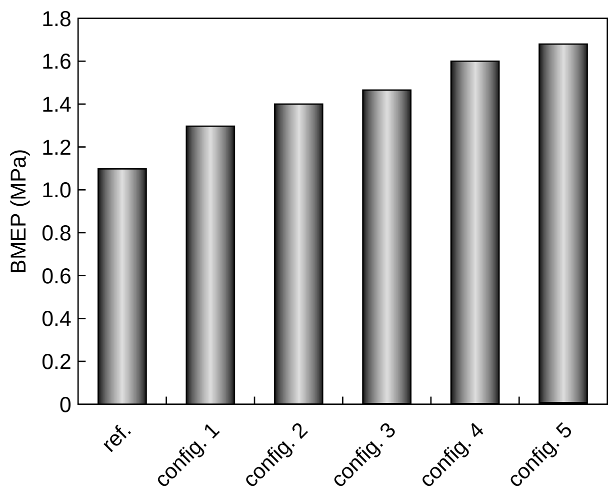
Table 4 illustrates the improvement achieved during last two years both in terms of specific engine power and associated specific fuel consumption. In a first step, while keeping a same performance level of 73 kW/l, the SFC at maximum power running condition has been reduced by 12%, moving from 345 g/kWh down to 305 g/kWh.

TABLE 4

Progressive improvement of maximum power running features

Engine configuration	**Specific output at 5500 rpm (kW/l)**	**Fuel consumption at maximum power (g/kWh)**
Twin VVT	73	345
Twin VVT + short intake	73	305
Twin VVT + short intake + reduced valve event duration	83	301

This was obtained using a new unconventional design of engine air intake system, with a small volume of intake plenum and very short intake ducts. At high engine speed, the induced modification in terms of acoustic significantly reduces engine volumetric efficiency *(Fig. 10)*. The turbocharger is used to compensate this phenomenon by an increase in air pressure. Normally, it is not desirable as can be seen in Equation (1). But, in that case, this shift in turbocharger running condition puts latter in a best efficiency area. In terms of turbocharger functioning, and especially for the high air mass flow required at maximum engine power, this is linked to an important improvement in compressor efficiency (η_c in Equation (1)). Moreover, balance between air pressure delivered by the turbocharger against back pressure generated at turbine stage entrance is improved. This leads to less in-cylinder residual burned



config.. no.	Cumulative engine evolution
reference	Fixed valve timing
config.. 1	Intake variable valve timing
config.. 2	Smaller turbine
config.. 3	Improved combustion (piston)
config.. 4	Exhaust variable valve timing
config.. 5	Re-optimized valve duration

Figure 9

Low end torque progressive improvement (1250 rpm).

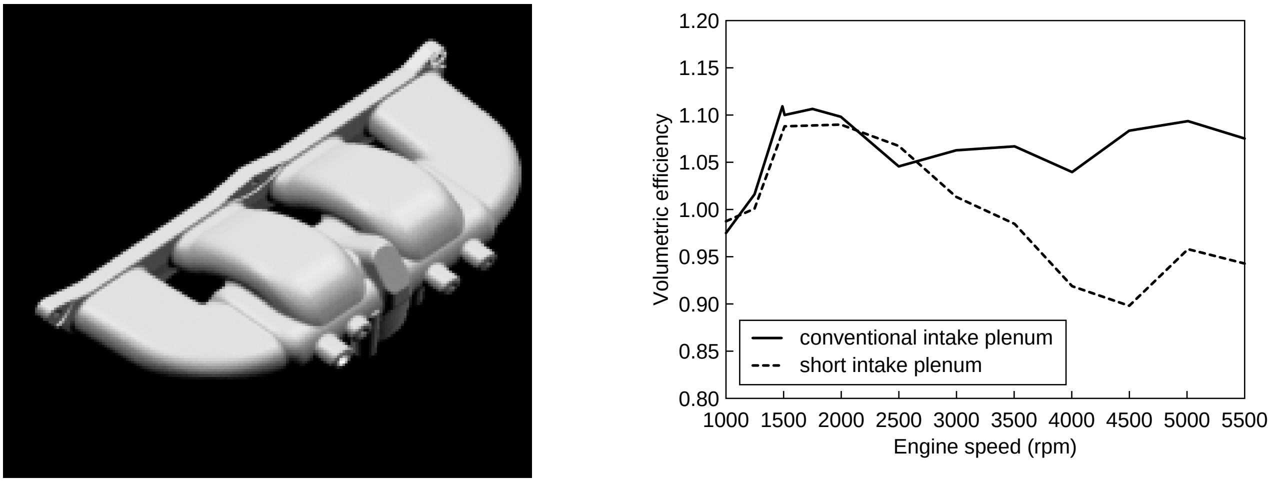


Figure 10

Optimized air intake plenum and effect on volumetric efficiency.

gases, thus a better combustion process. Fuel enrichment required to cool down turbine stage is lowered leading to the 12% improvement in SFC.

Recently, same principle was used, engine volumetric efficiency of the engine being modified by a shorter intake valve opening duration. Turbocharger thermal efficiency has been again improved. Then, the specific performance has been increased by 14%—from 73 to 83 kW/l—while keeping same SFC level *(Table 4)*.

CONCLUSION

IFP has developed an innovative approach combining two energy saving technologies which are gasoline direct injection and turbocharger with renewed definition of valve timing. At low engine speed, an unconventional combustion chamber scavenging process becomes then feasible on a turbocharged engine. Knock resistance and low speed volumetric efficiency are improved. The close interactions between combustion, turbocharger and valve timing have been analysed and optimized.

Recent results get on the *IFP* demonstrator 1.8 l engine, fitted with twin variable valve timing system, twin scroll well-matched turbocharger, improved combustion chamber and in-cylinder air flow design generates combination of extremely high specific output in terms of low end torque and power while presenting very low specific fuel consumption level on the entire range of running conditions. Torque reached at 1250 rpm is close to 240 N·m corresponding to an about 1.7 MPa BMEP while maximum power is 147 kW (83 kW/l) with a SFC of 301 g/kWh. That kind of engine is an interesting candidate to compete with current big gasoline engines with displacement from 2.5 to 3.0 l.

Next step is to apply the same approach on smaller engines and to find again so promising results. Then downsizing will be confirmed as a very promising way to reached the highly ambitious commitments taken concerning CO_2 emission reduction of the whole vehicle fleet.

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APPENDIX

Hereafter are presented, in Figures A1 to A3, the 3 driving cycles (from ARTEMIS European Research Program) representative of real life vehicle use, which have been used for calculations. They present a lot of transient operations and, as far as the urban cycle is concerned, 5 stop and go phases per kilometer. Average speed of these three cycles are respectively 18, 58 and 100 km/h.

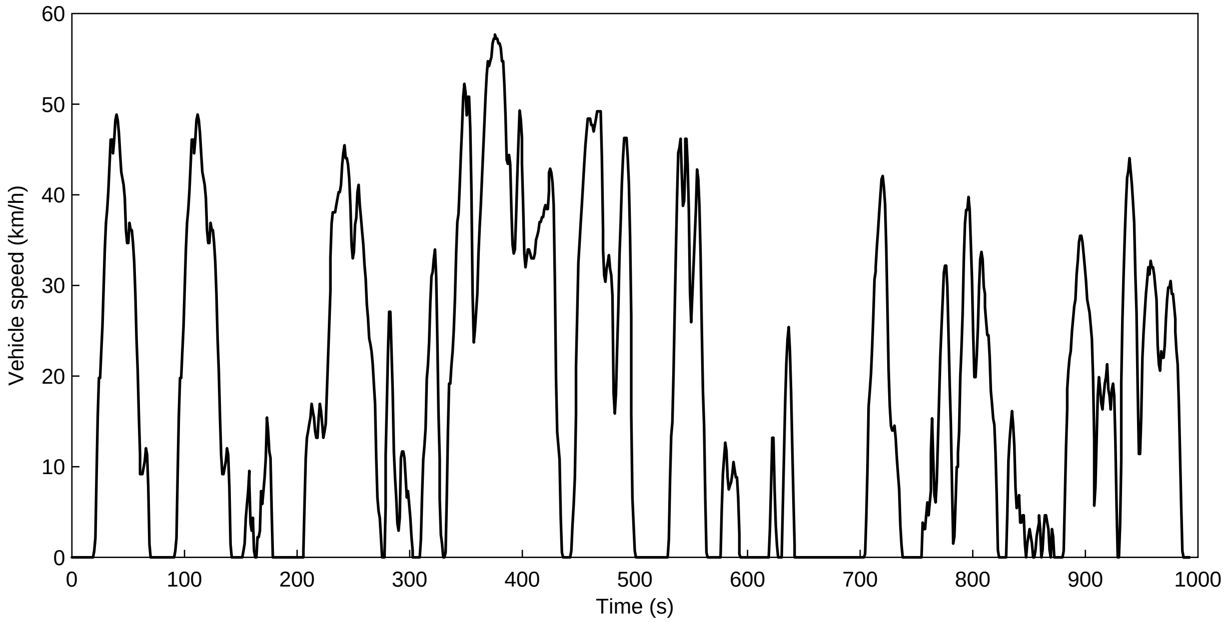

Figure A1

Urban driving cycle.

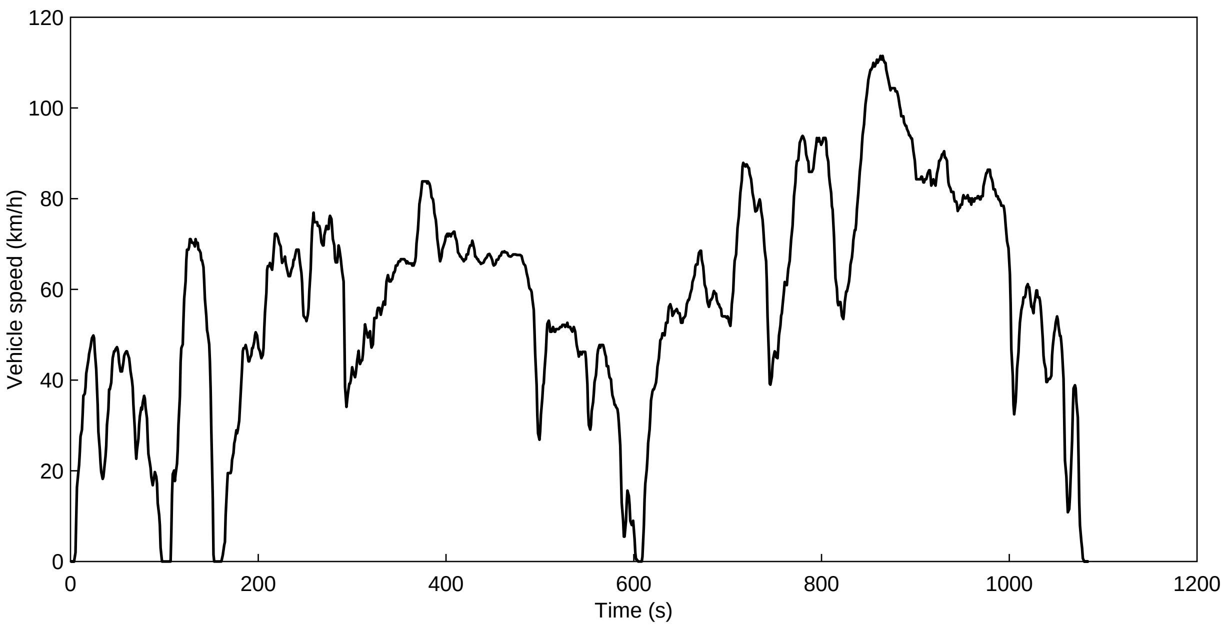

Figure A2

Road driving cycle.

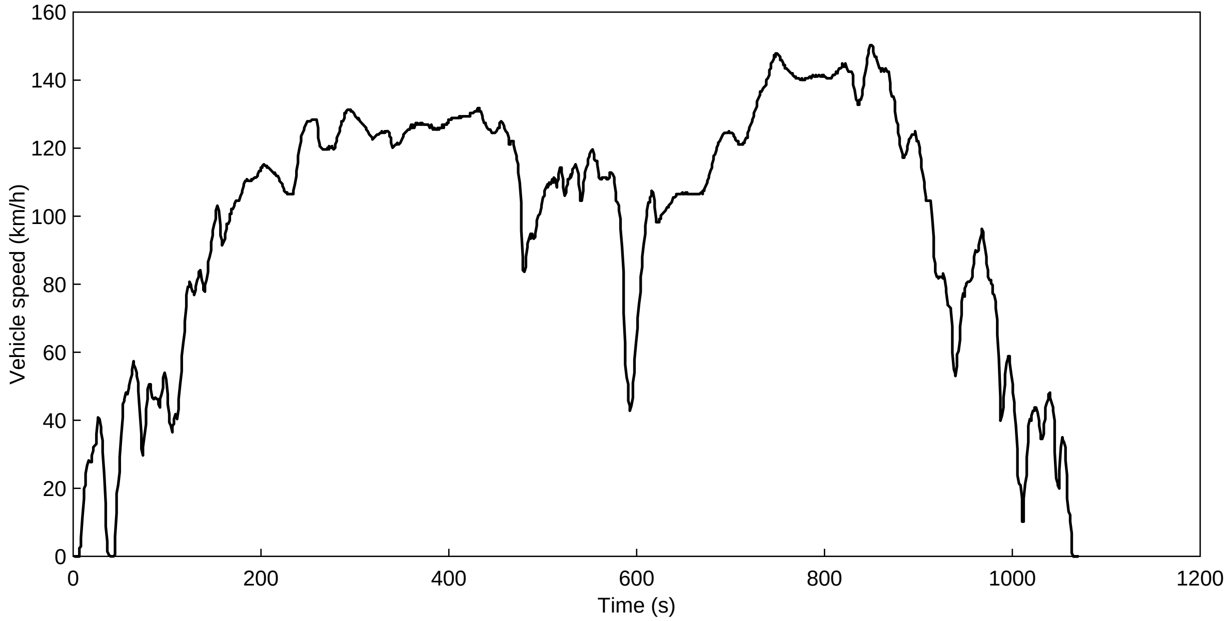

Figure A3

Motorway driving cycle.