

Comparison of PFI and DI Operation in a Downsized Gasoline Engine

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ABSTRACT

A 300 cc gasoline engine has been experimentally and numerically studied to compare PFI and DI operation on naturally-aspirated and turbocharged full load operating points. Experiment outlines the benefits from DI operation in terms of volumetric efficiency, fuel economy and knock propensity but also clearly indicates worse raw engine-out CO emissions. The latter is an indication of the survival of a large scale mixture heterogeneity in this downsized GDI engine even when early injection and intense induced fluid motion are combined.

For such a full load operation, the application of optical diagnostics to study mixture heterogeneity cannot be considered because pressure and temperature exceed sustainable levels for transparent materials. Therefore, 3D CFD RANS computations of the intake, injection, combustion and pollutant formation processes including detailed chemistry information are performed to complement the experimental data. The results at the end of compression stroke confirm the existence of a large scale mixture stratification. Moreover, the degree of mixture stratification can be directly related to the measured CO emissions. Combustion and pollutant formation modelling with detailed chemistry information allows quantifying the influence of locally rich mixtures on flame propagation and post-flame chemistry (CO and NO_x emissions).

Computations for various injection timings on a unique operating point explain why the experimentally-chosen injection timing is optimal. The interaction of intake-induced fluid motion with injection-induced fluid motion gives rise to a trade-off between turbulence and mixture homogeneity near TDC. Increased turbulence favours combustion speed and reduces knock propensity, which should lead to a better efficiency. But the gain in turbulence is counter-balanced by an enlarged mixture heterogeneity producing larger CO emissions and therefore impeding global efficiency.

CITATION: Knop, V. and Essayem, E., "Comparison of PFI and DI Operation in a Downsized Gasoline Engine," *SAE Int. J. Engines* 6(2):2013, doi:10.4271/2013-01-1103.

INTRODUCTION

Downsizing and Direct Injection (DI) implementation are two main trends in the gasoline engine development for the last decade [1, 2, 3, 4]. Both bring improvements in efficiency but also new challenges for engine optimisation, especially with respect to the ever more stringent emission standards. Combining both technologies is possible to extend individual gains [3, 4] but also increases the design difficulties and the need for an in-depth understanding of the underlying physical phenomena (mixture formation, flow motion, combustion initiation and spreading).

In particular, the use of downsizing requires to master the combustion at higher loads than for usual natural aspiration operation in order to maintain the engine performances. This induces a shift to a new range of local thermodynamic conditions in the cylinder during combustion spreading and therefore to new challenges for combustion modelling [5, 6].

The implementation of DI introduces the mixture formation issue [1, 7]. For full load operation, early injection is the most widespread choice since the mixture is made as homogeneous as possible and the combustion spreading is therefore the closest to what was previously known [7]. Despite early injection, CO emissions are generally larger than for PFI operation, proof that perfect homogeneity is not reached.

In the present work, a downsized engine implemented with Port Fuel Injection (PFI) and DI is studied in order to independently gain understanding on the influence of downsizing and of DI implementation in a downsized engine. Experimental measurements with PFI and DI operation show the traditional gains in efficiency and in knock resistance linked to DI implementation. On the other hand, the worse raw engine-out CO emissions prove the survival of a large scale mixture heterogeneity during combustion. 3D CFD



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Reynolds-Averaged Navier-Stokes (RANS) computations are therefore performed to bring additional information in view of better understanding the differences due to the change in operating conditions and the influence of the DI implementation design parameters.

In the following, the description of the experimental and numerical set-ups is first provided. The analysis of the experimental data then outlines the encountered problems and identifies the open questions. 3D CFD results for mixture formation process, combustion spreading and pollutant reactions during expansion stroke are eventually used to answer most questions and to identify the key points on which additional work is still required.

EXPERIMENTAL SETUP

Tests are conducted on a single-cylinder gasoline engine with a displacement of 301 cm³ and compression ratio of 9.5:1 (see full details in Table 1). Intake and exhaust valve timings are selected to avoid any scavenging and therefore any fuel short-circuiting to the exhaust when PFI operation is selected. This prevents from fully optimising DI operation on low-speed turbocharged full-load operating points but simplifies the comparison between PFI and DI operation.

Table 1. Single-cylinder engine characteristics

Displaced volume	301 cm ³
Bore	69 mm
Stroke	80.5 mm
Connecting Rod Length	139 mm
Compression ratio	9.5:1
Number of Valves	4
Inlet Valve Opening	10 CAD aTDC @ 1 mm lift
Inlet Valve Closure	20 CAD aBDC @ 1 mm lift
Exhaust Valve Opening	40CAD bBDC @ 1 mm lift
Exhaust Valve Closure	10 CAD bTDC @ 1 mm lift

The engine is equipped with a double fuel supply system and three injectors. Two Bosch TU5 injectors are located in each intake duct for PFI operation and a Bosch HDEV1.2 (multi-hole injector with 6 unevenly-located holes with a diameter of 144 µm, injection pressure of 100 bar) is implemented between intake valves and below the Siamese induction ports for DI operation. The side location of the direct injector allows to centrally implement a NGK T2511H spark plug. Since only early DI operation is considered, piston design is not complex and only includes a shallow lens shape.

Beyond static pressure and temperature measurements in the intake and exhaust lines, engine operation is monitored with the following crank-resolved pressure transducers:

- Kulite XT-123B-190-50A at inlet of each intake duct;
- Kistler 6043A60 in the cylinder;
- AVL QC43D in the exhaust pipe.

Data acquisition is performed on 300 consecutive engine cycles for all crank-resolved pressure transducers and static pressures and temperatures are averaged on 30s. In the following, crank-resolved data are presented with a mean value (ensemble-averaging on the 300 cycles) and a pressure envelope (lowest and highest measured pressure at each crank angle, which could lead to the appearance of angles in plotted curves when pressure traces from two different cycles cross each other).

Fuel metering is performed through direct weighting as well as fuel flow rate computation based on air flow rate measurement with sonic nozzles and fuel/air equivalence ratio quantification. The fuel/air equivalence ratio quantification relies on an analysis of the exhaust gas composition based on uHC, NO_x, CO, CO₂, and O₂ concentrations. Both techniques provide similar results lying within a few percents.

Operation with iso-octane only is selected so that fuel properties are perfectly known. This is important for the modelling part of the study (fuel injection and chemistry description). Experimental tests were also performed with standard gasoline to check the representativity of the measurements with iso-octane. These tests were conclusive and not further reported here.

NUMERICAL SETUP

3D CFD computations are performed with an in-house code called IFP-C3D [8]. This code is a hexahedral unstructured parallel (MPI) solver for reactive compressible gas flows with sprays dedicated to multi-physics three-dimensional simulations. It computes reactive flows in internal combustion engines, solving the RANS equations. When dealing with spark-ignition engines, detailed in-house models for spark ignition [9], flame spreading [10], auto-ignition detection [10] and pollutant formation including detailed chemical kinetics information [11] are considered. Turbulence modelling is performed with the k-ε RNG model [12] while the law-of-the-wall is computed according to the model of Kays and Crawford [13].

Spray description relies on a Lagrangian model accounting for spray atomisation (Wave model) and break-up (FIPA model) [14]. Droplet size is prescribed based on experimentally-derived SMD's and the FIPA constant is chosen to best fit the experimentally-observed spray dynamics in a test cell. The interaction of liquid fuel with walls is described with a model based on the work of O'Rourke and Amsden [15, 16] with further in-house developments to improve the description of fuel film dynamics on complex geometries [17].

Present computations include meshes with a typical cell size of 0.5 mm and describe the intake, compression and expansion strokes, including the injection and combustion phenomena. MPI parallelism provides a linear speed-up up to at least 128 processors [18] and therefore short elapsed times. With a number of processors (8-core Intel Sandy Bridge

E5-2670) ranging from 4 to 8 during the computation, results are obtained within 3 to 6 hours depending on engine speed.

Nitrogen Oxide Modelling

State-of-the-art modelling of nitrogen oxides is the only consideration of the thermal NO formation and the solving of the Zeldovitch mechanism [19], while alternative species and reaction paths exist. Since the efficiency of advanced after-treatment devices are dependent on such alternative species, their prediction becomes mandatory to allow a global system optimisation. Therefore, nowadays requirements from R&D engineers are shifting towards new numerical tools able to predict the presence of these additional species, mainly NO₂ and N₂O, but at a reasonable CPU cost.

The direct inclusion in CFD tools of new species and of the related formation pathways leads to a large increase in the number of radical species to transport and in the number of reactions to solve. To restrict the additional computational cost, a tabulated approach [11] is introduced to describe the prompt (Fenimore) route, the N₂O route and the NNH route without any additional radical species or any additional reaction. Describing the evolution of NO, NO₂ and N₂O mass fractions in the engine with a relaxation approach, this model is called Nitrogen Oxide Relaxation Approach (NORA) [11].

Based on the Intrinsic Low-Dimensional Manifold (ILDM) approach [20], the nitrogen oxide chemistry is described as idealised species evolutions [11]. Since the nitrogen oxide chemistry is slow compared to the combustion chemistry, it is not mandatory to solve the evolution of nitrogen oxides during the whole combustion event. If NO_x evolution is only computed when combustion is locally almost complete, no major discrepancy is introduced and the CPU time savings are large. The idealised species evolutions are described within the CFD computation based on the storage of a few quantities in a look-up table. The dependence of nitrogen oxide chemistry on the nature of the burnt fuel is reproduced in the model by considering a specific look-up table for each fuel or mixture of fuels. One advantage of the approach is its applicability to premixed (see below) or mixture-controlled [11] combustion modes.

The identification of dedicated nitrogen oxide ILDMs is obtained through perturbations of the equilibrium state and the computation of the subsequent chemistry return towards this equilibrium. This method is fully detailed in [11] and conclusions are only briefly summarised hereafter.

These perturbations show that NO and NO₂ are continuously interacting during combustion and expansion in an engine [11]. NO and NO₂ are therefore modelled with a common ILDM: a two-step reaction rate with a relaxation approach for each step is used in NORA following [11]:

$$\begin{cases} \dot{\omega}_{NO} = \frac{Y_{NO}^{\star} - Y_{NO}}{\tau_1} + \frac{Y_{NO}^{eq} - Y_{NO}^{\star}}{\tau_2} \\ \dot{\omega}_{NO_2} = -\frac{M_{NO_2}}{M_{NO}} \frac{Y_{NO}^{\star} - Y_{NO}}{\tau_1} + \frac{Y_{NO_2}^{eq} - Y_{NO_2}}{\tau_2} \end{cases} \quad (1)$$

where $Y_{NO}^{\star}$ is the meeting point between each steps and can be uniquely computed based on Y_{NO} and Y_{NO}^{eq} . The dynamics of each step are described by the relaxation times τ_1 and τ_2 .

Contrary to NO and NO₂, N₂O is ever close to the equilibrium (very short relaxation times) and appears to behave as an independent species. Consequently, a simple relaxation approach is enough to reproduce its reactivity:

$$\dot{\omega}_{N_2O} = \frac{Y_{N_2O}^{eq} - Y_{N_2O}}{\tau_3} \quad (2)$$

To compute these reaction rates, the quantities stored in the look-up tables are the NO, NO₂ and N₂O equilibrium mass fractions (Y_X^{eq}) and the relaxation times τ_1 , τ_2 and τ_3 . The table inputs for each quantity are pressure, enthalpy and mixture strength. The refinement along each look-up table axis is variable with a higher resolution in the large gradient areas (pressures between 1 and 10 bars as well as fuel/air equivalence ratio between 0.7 and 1.0).

Carbon Oxide Modelling

Using the same modelling approach as for nitrogen oxides, a model for CO/CO₂ chemistry behind the propagating flame is introduced. This model provides a description of the slow CO oxidation into CO₂ during the expansion stroke for full load operating points, capturing the chemistry freezing for lower temperatures.

A reaction rate description similar to Eq. (2) is considered with tabulated CO equilibrium mass fractions and relaxation times. The perturbation method for look-up table creation is identical to that for nitrogen oxides [11] except for the introduction of a carbon balance instead of the nitrogen balance.

Contrary to nitrogen oxide chemistry, CO chemistry is intimately linked to combustion progress and such an approach is not general. This approach is notably not valid within the flame brush but gives accurate results for post-flame chemistry description (in the burnt gases behind the flame). Since the CFM approach is used for the flame propagation description, this is not an issue because gases are statistically described as being either in the fresh or in the burnt gases with an infinitely thin flame brush. Consequently, the model is only applied to the burnt gases and CO evolution in the cylinder during the expansion stroke can be accurately predicted, as is demonstrated below. On the other hand, contrary to the NORA model, this approach cannot be directly used to model CO emissions from a Diesel engine.

TESTED OPERATING POINTS

PFI and DI operations are experimentally and numerically compared on stoichiometric full load operating points with conditions selected to reach as similar targets as possible.

Full Load Natural Aspiration Operating Points

First, natural aspiration (NA) full load operating points are considered to emphasize the difference in cylinder filling induced by the change in injection strategy. Fully unthrottled operating points are considered with identical intake runner pressures and therefore air mass flow rate benefits when DI operation is considered (see details in [Table 2](#)).

In addition to the cylinder filling gain, DI operation also provides a global mixture cooling and less knock propensity. Spark ignition timing for DI operation is therefore closest to the thermodynamic optimum, especially at lowest engine speeds, which should bring further torque gains.

Table 2. Natural aspiration operating points (engine load and air mass flow rate refer to DI operation)

Engine speed [RPM]	1000	3000	4000
Engine load – IMEP [MPa]	0.95	0.95	1.12
Air mass flow rate [kg/h]	9.0	25.0	37.4
Fuel/air eq. ratio [-]	1.0	1.0	1.0

On the other hand, DI is unable to provide a fully homogeneous mixture in such a small combustion chamber even with early injection timings. The stratified mixture has therefore a negative impact on emissions (mainly CO emissions) and consequently on efficiency. The global result is a limited benefit for DI operation with respect to PFI operation in terms of peak torque and slightly worst pollutant emissions.

Full Load Turbocharged Operating Points

For turbocharged (TC) full load operating points, the comparison is not performed at identical intake runner pressures anymore but at identical engine loads (see details in [Table 3](#)). The objective is not to identify a gain in peak engine torque (which would mainly result from the cooling effect) but to quantify the fuel consumption benefits.

There is no obvious enlargement of the cylinder filling gain when the fuel mass flow rate is increased since the mixture is still stoichiometric. More fuel evaporates but the mass of gases to cool proportionally increases. On the other hand, the cooling effect is far more critical for such operating points that induce a clearly larger propensity for knock (increased local pressures, with similar local temperatures). The difference in spark timing is still a benefit for DI operation, with no major changes in timing gap between DI and PFI. But combustion now begins at early expansion and any delay has far more influence.

Table 3. Turbocharged operating points

Engine speed [RPM]	1000	3000	4000
Engine load – IMEP [MPa]	1.6	1.8	2.0
Air mass flow rate [kg/h]	17	46	70
Fuel/air eq. ratio [-]	1.0	1.0	1.0

MIXTURE FORMATION

A fundamental aspect of DI is the mixture formation process. Compared to PFI operation, DI operation modifies local in-cylinder conditions on two central aspects: temperature and fuel/air equivalence ratio fields. From the experimental point of view, this is evidenced by a better cylinder filling and higher CO emissions, respectively.

Cylinder Filling and Cooling Effect

For all 6 tested operating points, the switch from PFI operation to DI operation implies a gain in cylinder filling ([Figure 1](#)). This effect originates from the heat sink term introduced by fuel droplet evaporation. For PFI operation, the evaporation takes place in the intake duct and part of the cooling effect is erased by the enhanced heat flux from pipe walls to the gas. For DI operation, the cooling effect directly occurs in the cylinder, reducing the negative effect of enhanced heat flux. Additionally, injection timing during intake stroke for DI operation is more favourable since heat is pumped out precisely when it is beneficial to cylinder filling.

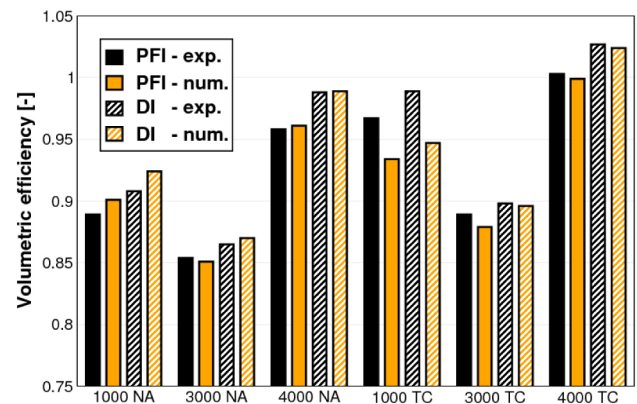


Figure 1. Effect of DI operation on cylinder filling - Comparison of experimental and numerical results

The quantification of cylinder filling with CFD is accurate both for PFI and DI operations ([Figure 1](#)) so that numerical results can be used to detail such an effect. For argument's sake, [Figure 2](#) displays the in-cylinder temperature and density during the intake stroke for PFI and DI operation at 1000 RPM NA. Both operating points have similar initial and boundary conditions (identical intake runner static pressure and static temperature) and initial conditions (similar mass of trapped burnt gases) but experiment indicates an improvement of 2.6% for cylinder filling with DI. The occurrence of fuel evaporation directly in the cylinder during