

# Fuel Economy Improvement and Knock Tendency Reduction of a Downsized Turbocharged Engine at Full Load Operations through a Low-Pressure EGR System

Luigi Teodosio and Vincenzo De Bellis  
Univ Naples Federico II

Fabio Bozza  
Univ Naples Federico II-Ist. Motori CNR

## ABSTRACT

It is well known that the downsizing philosophy allows the improvement of Brake Specific Fuel Consumption (BSFC) at part load operation for spark ignition engines. On the other hand, the BSFC is penalized at high/full load operation because of the knock occurrence and of further limitations on the Turbine Inlet Temperature (TIT). Knock control forces the adoption of a late combustion phasing, causing a deterioration of the thermodynamic efficiency, while TIT control requires enrichment of the Air-to-Fuel (A/F) ratio, with additional BSFC drawbacks.

In this work, a promising technique, consisting of the introduction of a low-pressure cooled exhaust gas recirculation (EGR) system, is analyzed by means of a 1D numerical approach with reference to a downsized turbocharged SI engine. Proper “in-house developed” sub-models are used to describe the combustion process, turbulence phenomenon and the knock occurrence.

The effects of cooled gas recirculation are studied for various EGR and A/F levels. The presented results highlight that the proposed solution involves significant BSFC improvements. In particular, it is able to reduce the engine knock tendency, resulting in the possibility to advance the combustion phasing. Moreover, it involves a TIT reduction that allows to limit the over-fuelling.

The developed numerical procedure is able to take into account the complex interactions among different driving parameters affecting the fuel consumption. It can be hence very useful to define a numerical engine pre-calibration and to realistically predict the EGR-related BSFC advantages which can be gained at different speed and load conditions.

**CITATION:** Teodosio, L., De Bellis, V., and Bozza, F., "Fuel Economy Improvement and Knock Tendency Reduction of a Downsized Turbocharged Engine at Full Load Operations through a Low-Pressure EGR System," *SAE Int. J. Engines* 8(4):2015, doi:10.4271/2015-01-1244.

## INTRODUCTION

Recent automotive internal combustion engines (ICEs) are characterized by more and more complex architectures in order to minimize the Brake Specific Fuel Consumption (BSFC), without penalizing power and torque performance, and to comply with the European law concerning the pollutants and CO<sub>2</sub> emissions [1]. To this aim, regarding the Spark Ignition (SI) ICEs, a common strategy consists in the reduction of the total displacement and in coupling the engine to a turbocharger (*downsizing*) [2,3].

Downsizing allows for a BSFC reduction at part load with respect to the naturally aspirated engine delivering a similar power output. For an assigned torque request, in fact, a smaller displacement determines a higher Break Mean Effective Pressure (BMEP) level. Consequently, the engine frequently works in a more efficient zone of its operating plane, where the intake throttling losses are less relevant [2,3] and the mechanical efficiency is higher, as well. Further improvements can be achieved at very low BMEP operation through the adoption of Variable Valve Timing or Actuation (VVT/VVA) systems [4, 5, 6]. However, engine downsizing does not undermine the full load power and torque performance, thanks to the possibility to increase the intake air density by means of a turbocharger.

On the other hand, this solution causes an increase in the thermo-mechanical load, that forces a more robust engine design. Furthermore, the engine efficiency may be strongly penalized when the boost level becomes too high, since pre-ignitions or heavy knocking combustions may arise [7,8]. Pre-ignition mainly occurs at low speed and is usually avoided reducing the engine load. Knock phenomena indeed can be controlled by moving the combustion phasing very late in the expansion stroke. This, while penalizing the thermodynamic efficiency [9], causes an increase in the exhaust temperature and, in particular, at the turbine inlet. This additional concern is usually compensated by an enrichment of the air-fuel mixture, with further fuel economy penalizations [10].

One of the more promising techniques to overcome the above issues is the recirculation of a large amount of inert exhaust gases into the cylinders. This can be obtained by proper valve strategies, such as an extended exhaust/intake valve overlap [11], or by an external Exhaust Gas Recirculation (EGR) circuit [12]. The latter solution, although requiring some complications in the engine architecture, allows to substantially cool the recirculated gases, with direct advantages on knock and Turbine Inlet Temperature (TIT), as well [12].

Current literature is plenty of papers experimentally demonstrating the potential of EGR in improving fuel economy [13, 14, 15, 16, 17]. Bourhis et al. [13] analyzed the effects of internal and external EGR in turbocharged SI engines at moderate load. In particular, the first solution allows an Indicated Specific Fuel Consumption (ISFC) improvement of 8% at 1500 rpm and 8 bar Indicated Mean Effective Pressure (IMEP) with an EGR increase from 5% up to 8%. Through the second strategy, a ISFC reduction of 4% is obtained at 2000 rpm and 10 bar IMEP.

In [14], indeed, high load operation was mainly addressed, showing fuel consumption improvements ranging from 6% at high speed, up to 17% at low speed. Cairns et al. [15] performed a comparison of excess fuel, excess air, and cooled external EGR charge dilution, revealing that cooled EGR is the most effective knock suppressant among the other investigated strategies. Experimental work of Grandin et al. [10] confirmed that cooled EGR is the most effective replacement of fuel enrichment, especially if a three-way catalytic converter has to be used at all operating conditions. Potteau et al. [16] measured BSFC reductions up to 14% and an exhaust gas temperature decrease up to 100°C. Similar advantages were claimed in [17], once again emerging from a decrease in knock tendency allowing for a better combustion phasing, as well as from reductions in exhaust temperatures that eliminated the necessity of over-fuelling.

The above results, as said, were demonstrated by extensive experimental analyses at the test bench, requiring a complete engine recalibration to fully exploit the EGR advantages. Of course, the possibility to recover similar outcomes through numerical 1D or 3D analyses [18, 19, 20] would be very helpful to reduce the development costs, the time-to-market and to also analyze the trade-offs among different architectures (low-pressure or high-pressure EGR, recycled gas temperature, etc.). However, the complex

interactions among different driving parameters affecting the fuel consumption - namely the mixture composition, the auto-ignition chemistry, the flame speed, the heat transfer, the boost level, etc. - imply the need of employing very accurate numerical tools. In addition, since the whole engine system has to be characterized, a 1D model, enhanced to properly furnish a satisfactory description of combustion and knock phenomena, appears to be the more suitable numerical tool.

In the present paper, the EGR potential in mitigating knock phenomena are analyzed with reference to a small twin cylinder turbocharged VVA engine through the employment of a commercial 1D code (GT-Power™). High-pressure and Low-Pressure (LP) circuits are initially considered. However, preliminary calculations for the former solution highlighted the presence of intense pressure fluctuations within the high pressure branches of the intake and exhaust circuits (mainly due to the twin-cylinder architecture), responsible for frequent flow reversals within the EGR circuit. The above problems are indeed drastically mitigated in the second solution, because of the damping effect exerted by the turbocharger. For this reason, the LP cooled EGR system is only analyzed in the following. Model predictivity is greatly improved by “in-house developed” sub-models for the description of combustion [21] and knock [22] processes. The engine model has been in fact already extensively validated at full and part load operation [23, 24, 25, 26], as well.

Preliminarily, in order to also assess the accuracy of the knock sub-model, proper simulations are carried out to identify the Knock Limited Spark Advance (KLSA) at full load for various speeds. In a second stage, a LP cooled EGR circuit is “virtually” mounted on the base engine to evaluate the BSFC improvements which can be gained at full load, where the knock and over-fuelling problems are more relevant. In particular, the effects of different EGR amounts and Air-to-Fuel (A/F) ratios are analyzed.

The numerical model is build up in such a way that the Spark Advance (SA) and the waste-gate opening are automatically adjusted to achieve an assigned knock threshold and BMEP level, respectively. In addition, prescribed maximum levels are specified for TIT, turbocharger speed and boost pressure.

## EXPERIMENTAL ACTIVITY

The considered engine is installed at the test-bench, and full load operation is analyzed for different speeds (1500-5500 rpm). The main characteristics of the tested engine are listed in [Table 1](#). Acquired reference data include air flow, fuel flow, torque, pollutants emissions, and fuel consumption. Instantaneous in-cylinder pressure measurements are also carried out in both cylinders. Main control parameters, namely the intake valve closure timing, the A/F ratio, the boost level, and the SA are selected referring to the ECU-stored values of the standard engine calibration. In particular, as shown in [Figure 1](#), due to knock limitations a delayed combustion phasing is realized (here identified by the angular position of the 50% Mass

Fraction Burned - MFB50%). Combustion phasing substantially increases at low engine speeds, where a higher knock tendency occurs. In addition, the A/F ratio (Figure 2) has to be particularly enriched at high engine speeds mainly to maintain the TIT below a prescribed threshold level (950 °C). Of course, the heat subtracted by the fuel evaporation contributes to reduce knock risks, and a more favorable MFB50% setting can be realized at high speed (Figure 1). In any case, both the above control strategies, although mandatory, strongly penalize the fuel consumption.

Table 1. Engine Main Features

| Model                 | 2 cylinders, 8 valves, VVA, Turbocharged and Intercooled |
|-----------------------|----------------------------------------------------------|
| Displacement          | 875 cm <sup>3</sup>                                      |
| Stroke / Bore         | 86 mm / 80.5 mm                                          |
| Connecting rod length | 136.85 mm                                                |
| Compression Ratio     | 9.9                                                      |
| Max Brake Power       | 64.6 kW @ 5500 rpm                                       |
| Max Brake Torque      | 146.1 Nm @ 2500 rpm                                      |

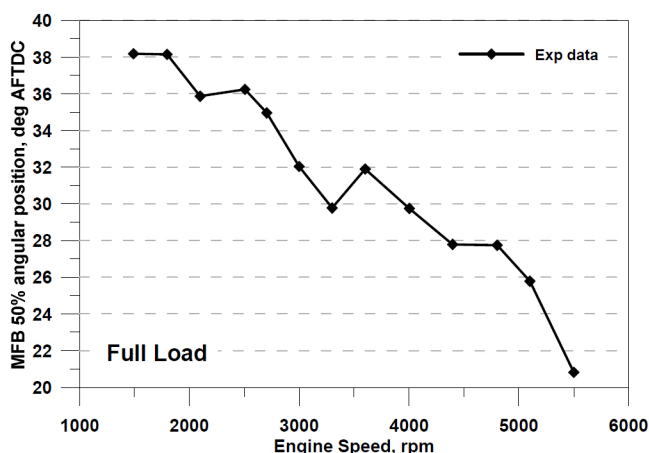


Figure 1. Measured MFB50% angular position at full load

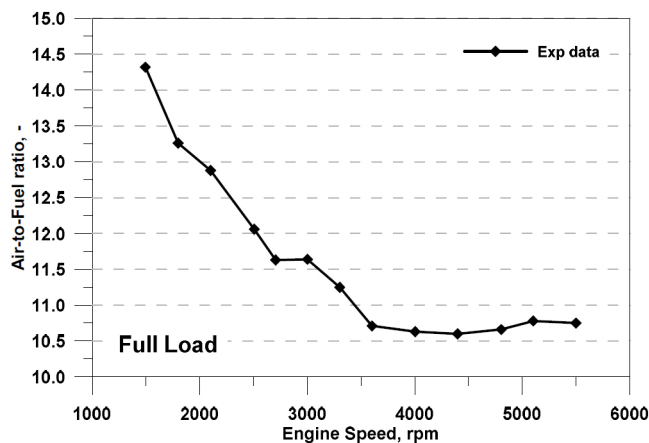


Figure 2. Experimental air-to-fuel ratio at full load

## COMBUSTION / TURBULENCE AND KNOCK MODELS

As said, in order to account for the complex interaction among several devices, great care in the simulation must be paid in modeling the combustion and knock processes. Both phenomena are properly described by sub-models introduced in GT-Power through “user routines”. The combustion process in particular is modeled through a “fractal approach” [21, 22, 23, 24], sensing both the combustion system geometry and the operating parameters, including the residual gas content. Flame wrinkling scales and the fractal dimension of the flame area depend on the in-cylinder turbulence evolution, described by a 0D sub-model, extensively reported in [23, 24]. Following the referenced hierarchical 1D-3D approach, the turbulence and combustion model tuning constants are selected in order to fit the in-cylinder 3D-derived turbulent intensity and the in-cylinder pressure cycles, respectively. A very good agreement between the numerical and experimental results is obtained [24]. The model consistency has been proved at part load, as well, in terms of fuel consumption and combustion duration/phasing [25]. It must be stressed here that a single set of tuning constants has been selected all over the engine operating plane.

With the aim of also describing knock phenomena, turbulence and combustion models are coupled to a chemical kinetic solver (CHEMKIN), able to detect the presence of auto-ignition reactions in the end-gas zone [22]. A reduced kinetic mechanism for the oxidation of iso-octane and n-heptane mixtures is specified. It was developed by Tanaka and Keck [27,28] and includes 5 elements, 32 species and 55 reactions. The mechanism handles for both low and high temperature reactions and it is particularly tuned [29] to reproduce the end-gas conditions in a SI ICE. The commercial gasoline used in the experimental tests is schematized in the simulations as a blend of iso-octane and n-heptane (97.5% and 2.5% in volume, respectively). More details about the knock model assessment are included in [30, 31, 32].

A special attention is devoted to the heat transfer modelling because of its critical influence on the knock prediction. An almost standard Woschni correlation is adopted, while the piston, cylinder liner and head temperatures are calculated by the finite element approach implemented in GT-Power. The heat transfer coefficient on the coolants side (water and oil) is derived by dedicated simulations of the coolant circuit and properly adjusted according to the engine speed, following the approach described in [33].

## MODEL VALIDATION

As stated before, the engine model has been already validated at full load operation in [23,24]. In particular, for each considered operating point, the experimentally actuated intake valve lift profile and A/F ratio are assigned as an input for the simulations. Because of the absence of information about the experimental waste-gate valve opening, the latter is automatically identified by a PID controller targeting the measured in-cylinder pressure level at the firing TDC. The experimental spark advance is imposed as a further input

parameter. The latter corresponds to the experimentally determined KLSA. In Figure 3 and Figure 4 previously obtained numerical outcomes (labelled as “Model results - Exp KLSA”) are compared to the measured data (labelled as “Exp data - Exp KLSA”) in terms of BMEP and BSFC, respectively, denoting a satisfactory accuracy.

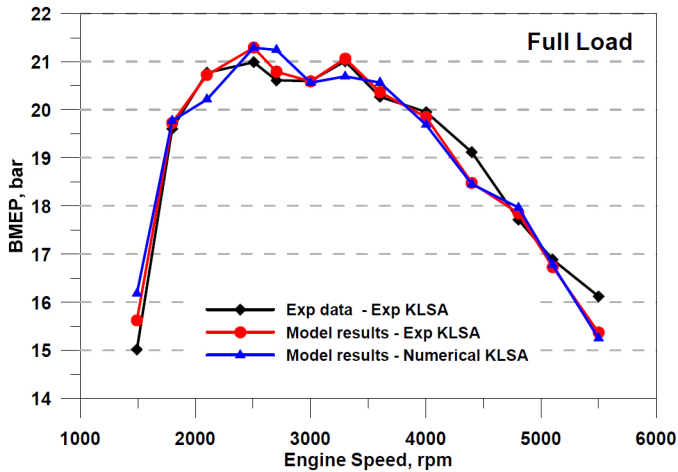


Figure 3. Numerical/experimental BMEP at full load

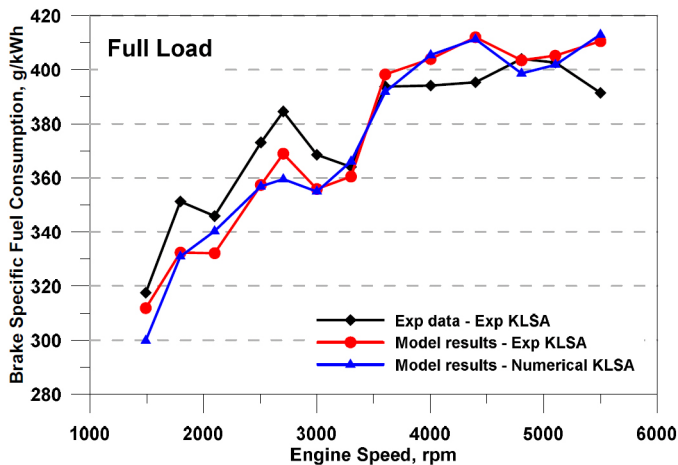


Figure 4. Numerical/experimental BSFC at full load

The Figures also report BMEP and BSFC results obtained without changing the valve lift profile, the A/F setting and the WG control strategy, but applying a numerical recalibration of the spark advance at the numerically identified knock limit (labelled as “Model results - Numerical KLSA”). Knock onset is here assumed as the condition when “high temperature” reactions are activated in the unburned zone. Actually, at borderline knock, in fact, “low temperature” reactions are always activated, inducing, however, a reduced heat release, and a negligible oscillation in the in-cylinder pressure trace. The latter is hence typically accepted by standard experimental knock detection techniques [8,34].

To better understand the numerical procedure for knock detection, representative in-cylinder pressure, temperature and MFB traces,  $p$ ,  $T_u$ ,  $T_b$  and  $x_b$  are shown in Figure 5 at borderline knock. Solid lines (labelled as “Knock”) refer to a spark timing setting inducing the activation of high-temperature reactions. Only low-temperature reactions occur for a slightly delayed spark advance (dashed lines, “No knock” label). The Figure also reports the overall heat fraction,  $x_{Qk}$ , released by the auto-ignition reactions, computed by the eq. (1).

$$x_{Qk} = \frac{\int \frac{dQ_k}{dt} dt}{\dot{m}_f LHV} \quad \frac{dQ_k}{dt} = -m_u \sum_1^{N_{spec}} e_i \frac{dx_i}{dt} \quad (1)$$

In the above equations, the auto-ignition heat ( $dQ_k/dt$ ) is estimated as a function of the composition variation of the  $i$ -th species ( $dx_i/dt$ ), provided by the solution of the considered reaction scheme at each time step.

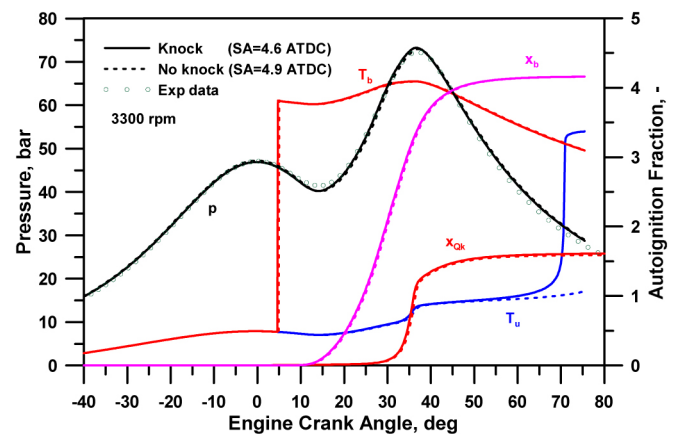


Figure 5. Typical engine cycles at borderline knock

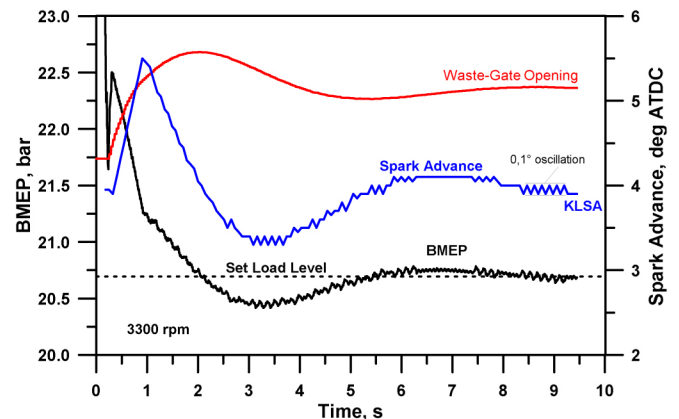


Figure 6. Time history of the engine control parameters for KLSA determination