

Analysis of the Performance of a Turbocharged S.I. Engine under Transient Operating Conditions by Means of Fast Running Models

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ABSTRACT

The aim of this work is the assessment of the predictive capabilities of fast running models, obtained through an appropriate reduction and simplification process from detailed 1D fluid-dynamic models, for a turbocharged s.i. engine under highly transient operating conditions.

Simulations results have been compared with experimental data for different types of models, ranging from fully detailed 1D fluid-dynamic models to map-based models, quantifying the degradation of the model accuracy and the reduction in the computational time for different kinds of driving cycles, from moderately transient such as the NEDC to highly dynamic such as the US06.

Although the map based approach was confirmed to be a viable means to predict fuel economy over the NEDC cycle, even for a downsized and turbocharged engine, thanks to the low accelerations involved and to the almost negligible transients effects, it showed significant discrepancies (error higher than 5%) with the experimental data when applied to highly dynamic driving cycles such as the US06.

On the other hand, the use of fast running models was demonstrated to be a suitable solution to obtain a satisfactory accuracy in the estimate of the fuel consumption also over highly dynamic driving cycles, with differences between the simulation results and the measured values which were within the repeatability range of the experimental tests. Moreover, the reduced computational effort of the fast running models allowed to reach close to real-time simulation performance.

CITATION: Millo, F., Di Lorenzo, G., Servetto, E., Capra, A. et al., "Analysis of the Performance of a Turbocharged S.I. Engine under Transient Operating Conditions by Means of Fast Running Models," *SAE Int. J. Engines* 6(2):2013, doi: 10.4271/2013-01-1115.

INTRODUCTION

The increasing complexity of automotive s.i. engines, with wide application of turbocharging and downsizing, often in combination with variable valve actuation and/or gasoline direct injection [1,2,3,4,5], along with the need to evaluate their performance over more dynamic driving cycles such as for instance US06 or WLTP [6], which have been recently introduced to get a closer representation of real-world driving conditions, is making the traditional performance map-based approach for the simulation of driving cycles and vehicle acceleration manoeuvres unsuitable, due to the lack of

capability in reproducing transient phenomena like turbolag, scavenging etc.

On the other hand, detailed 1D fluid-dynamic models which are widely and successfully used to address these issues, are usually unsuitable for the simulation of type approval driving cycles, because of their high computational time. Simulation speed is indeed of primary importance when running long transients such as a driving cycle, due to the need of comparing several different configurations for the engine or the vehicle.

The use of fast running models, obtained through an appropriate reduction and simplification process from the

detailed ones, can provide a viable solution to obtain, at the same time, a satisfactory accuracy in the reproduction of the phenomena under investigation and a reduced computational effort, allowing to reach close to real-time simulation performance. Moreover, by preserving a proper representation of the physical characteristics of the engine, these models are also suitable to allow an easy and quick investigation on the impact on vehicle's performance of different control and calibration strategies, such as for instance boost and spark advance management during transients.

The aim of this work is therefore the assessment of the predictive capabilities of fast running models for a turbocharged s.i. engine under highly transient operating conditions.

Simulations results were compared with experimental data for different types of models, ranging from fully detailed 1D fluid-dynamic models to map-based models, quantifying the degradation of the model accuracy and the reduction in the computational time for different kinds of manoeuvres and driving cycles, from moderately transient to highly dynamic.

TEST ENGINE AND VEHICLE CHARACTERISTICS

A European mid-size passenger car was selected as test vehicle for the investigation: the car has a mass of 1365 kg, almost perfectly identical the European average value, and is equipped with a 6-speed manual transmission.

The main design features of the test engine are listed in [Table 1](#): it is a 4 cylinder in line, s.i., gasoline direct injection turbocharged with wastegate, which is equipped with the Fiat Multiair™ Variable Valve Actuation system [7]. The Torque and Power curves at Full-Load are depicted in [Figure 1](#), showing an impressive 86 kW/l (114 HP/l) specific power.

Table 1. Main design features of the test engine.

Type	4 cylinder in line, s.i., turbocharged (wastegate)
Bore/Stroke	72/84 mm
Displacement	1368 cm ³
Compression ratio	9.8:1
Maximum power	121 kW at 5500 rpm
Maximum torque	250 Nm at 2250 rpm
Fuel Metering system	Port Fuel Injection
Air management system	VVA – Multiair

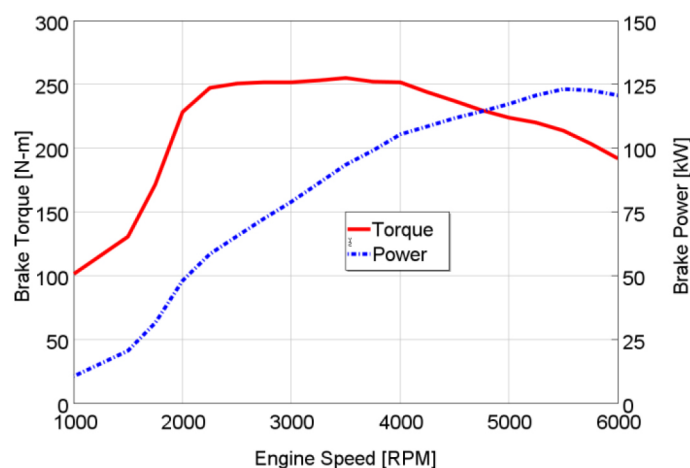


Figure 1. Brake Torque and Power at Full Load.

ENGINE MODEL CONSTRUCTION AND VALIDATION

First, a detailed engine model was built by means of GT-POWER, a one-dimensional fluid-dynamic code developed by Gamma Technologies Inc. for engine performance prediction. The main features of the code are described in detail in [8], [9], while only a brief summary of the model validation process is reported hereafter.

Model Validation at Full Load

In order to assess the engine model accuracy and its reliability, the simulation results were compared with the experimental results over the whole engine operating range, by analyzing both the global quantities related to breathing and brake measurement.

At first the detailed engine model was calibrated and validated at Full-Load. A model-based controller was introduced, which acts on the waste-gate opening diameter in order to match the experimental boost pressure. The experimental Air-to-Fuel Ratio (AFR) was imposed as well, via a simple injector model. After a thorough calibration of the gas-exchange processes, the combustion and the friction models were refined as well, on the basis of experimentally measured heat release rate profiles and of motoring torque data respectively. Eventually, the agreement between the simulated and experimental measures was considered to be satisfactory. A comparison between the experimental and simulated values of the volumetric efficiency, indicated mean effective pressure and brake mean effective pressure is shown in [Figures 2, 3, 4](#): a satisfactory agreement between the computed and measured values can be observed, with differences that are, on average, lower than 2% for the whole speed range.

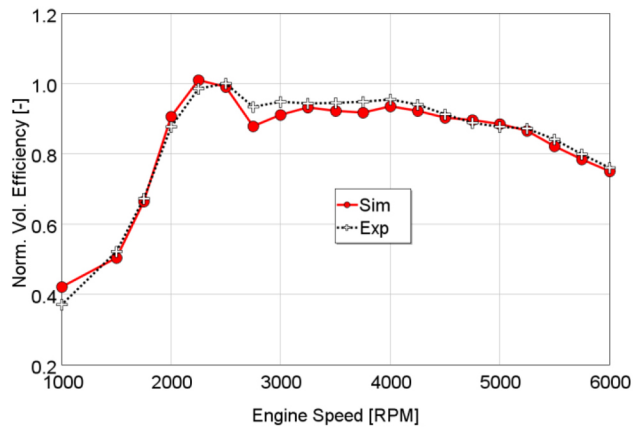


Figure 2. Normalized Volumetric Efficiency at full load, simulated (red solid line) and experimental (black dashed line).

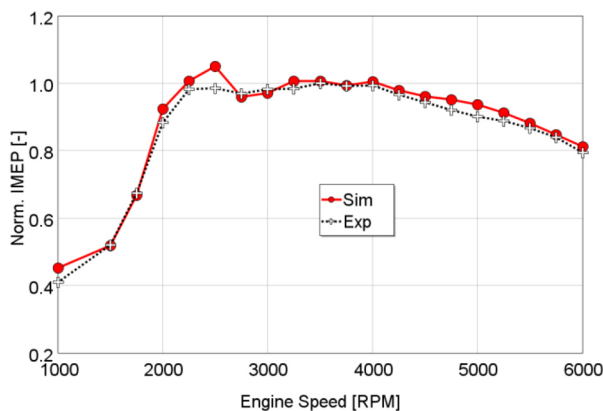


Figure 3. Normalized Indicated Mean Effective Pressure at full load, simulated (red solid line) and experimental (black dashed line).

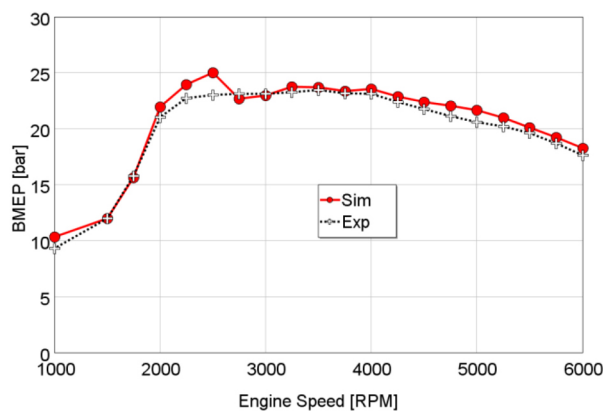


Figure 4. Brake Mean Effective Pressure at full load, simulated (red solid line) and experimental (black dashed line).

Model Validation at Part Load

The detailed engine model validated at full-load was then made part-load and transient capable by introducing speed- and load-dependent look-up tables for all the engine operating parameters (e.g. VVA lift profiles, target AFR, target boost pressure, heat release rate profiles, etc). In this way, during a transient run, the model can self-adjust the abovementioned inputs, on the basis of the actual operating point.

At steady-state, the engine load was adjusted by a model-based controller which acts on the throttle, targeting the reference BMEP. This control automatically recognizes the full-load status, and in this case it opens the throttle in its entirety.

The in-cylinder wall temperatures were calculated by a “TWallSoln” model, which is a FE-based solver that calculates in each cycle the heat flux from the gas to the walls and from the latter to the coolant fluids, water and oil.

In the following [Figures 5, 6, 7](#) the simulated volumetric efficiency, ISFC and BSFC are compared against the correspondent experimental values, over a load sweep at a constant speed of 2000 rpm. As it is possible to notice, the match between the simulation results and the measurements is quite good and can be considered as satisfactory. Similar load sweep simulations were carried out at several different engine speeds, and the resulting match between simulation and measurement is of a comparable quality on all engine operating points. Particular attention was paid in the operating region within the 20 bar BMEP and 4000 rpm boundaries, where most of the operating conditions of the more common driving cycles (i.e. NEDC and WLTP for Europe, FTP75, HWFET and US06 for US) lies, as shown in [Figure 8](#).

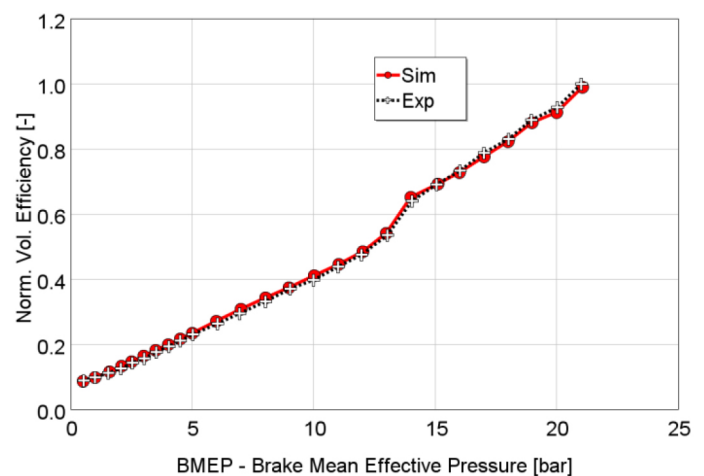


Figure 5. Normalized Volumetric Efficiency at 2000 rpm part load, simulated (red solid line) and experimental (black dashed line).

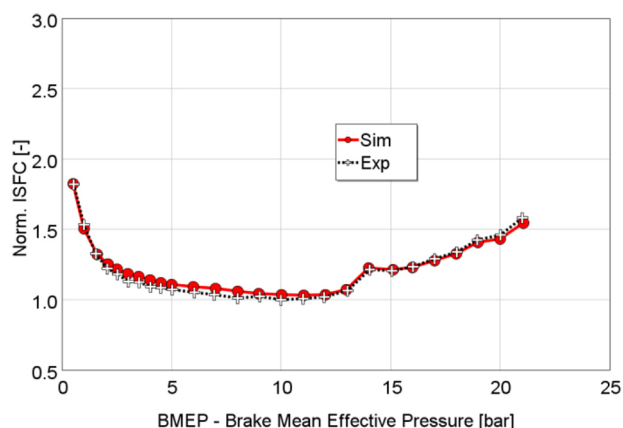


Figure 6. Normalized Indicated Specific Fuel Consumption at 2000 rpm part load, simulated (red solid line) and experimental (black dashed line).

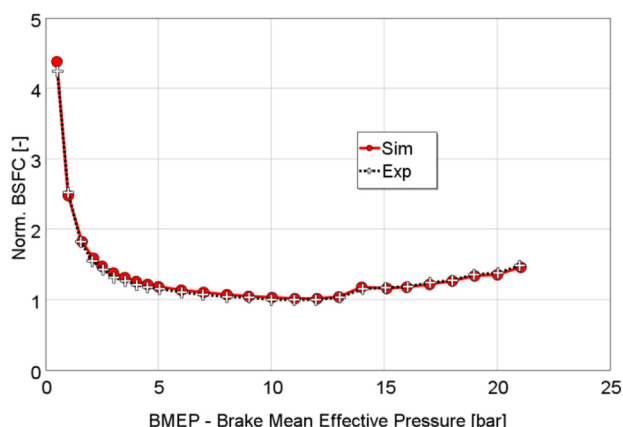


Figure 7. Normalized Brake Specific Fuel Consumption at 2000 rpm part load, simulated (red solid line) and experimental (black dashed line).

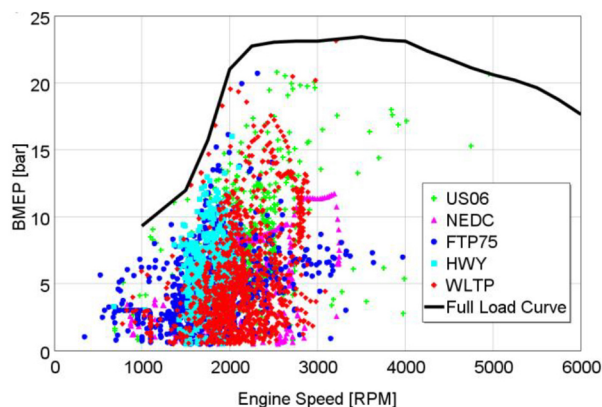


Figure 8. Engine operating points, for five different driving cycles (i.e. NEDC, WLTP, FTP75, HWFET and US06).

Figure 9 shows the relative difference between the simulated and experimentally measured BSFC over the abovementioned area of the engine operating map: differences are, on average, below 3%.

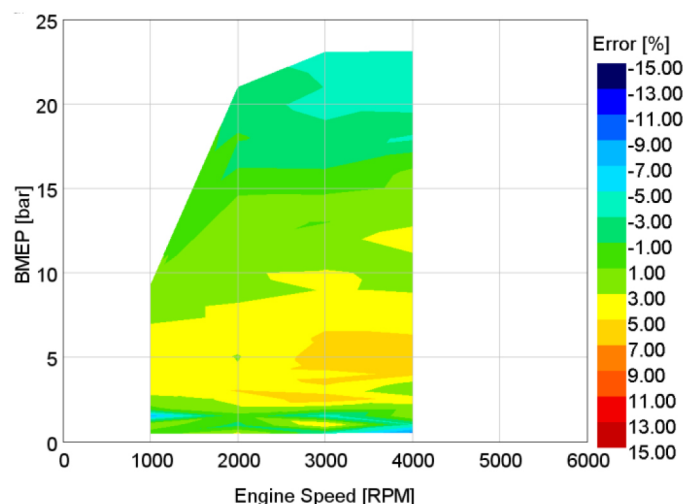


Figure 9. Percentage difference between experimental and simulated brake specific fuel consumption at steady-state.

Fast Running Model

Simulation speed is of primary importance when running long transients such as a driving cycle, with the need of comparing several different configurations for the engine or the vehicle. High level, integrated simulation models often require a fast running, yet fairly accurate, engine model to allow full system simulations to be performed in reasonable run times. Since the detailed engine model may require execution times two orders of magnitude slower than real time with current CPU capabilities, a simplified or Fast Running Model (FRM) [10] was derived from the detailed model.

In this work a Fully Physical FRM was built, which is similar to a standard detailed engine model, but with a coarser discretization of the flow path. Fully Physical FRMs are physically based and highly predictive. Furthermore, they are crank angle resolved and thus can provide instantaneous results.

The detailed engine model reduction process basically consists of lumping volumes together, thereby reducing the number of flow volumes and, consequently, increasing the time step size. In this specific case the number of flow volumes was reduced significantly, from a total of 216 sub-volumes in the detailed model to 41 only in the FRM. Of course, the total volume and the total heat transfer surface were preserved in each component. The less detailed flow path causes a certain loss in fidelity of wave dynamics. However, this drawback affects high frequency phenomena such as acoustics, while the main fluid-dynamics phenomena are still well predicted. Thanks to the larger volumes, the