

Numerical Investigation of the Relationship between Engine Performance and Turbocharger Speed of a Four Stroke Diesel Engine

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ABSTRACT

A condition monitoring activity consists in the analysis of several information from the engine and the subsequent data elaboration to assess its operating condition. By means of a continuous supervision of the operating conditions the internal combustion engine performance can be maintained at design-level in the long term.

The growing use of turbocharger (TC) in automotive field suggests to use the TC speed as a possible feedback of engine operating condition. Indeed, the turbocharger behavior is influenced by the thermo and fluid-dynamic conditions in the cylinder exhaust port: this feature suggests that the TC speed could provide useful data about the engine cycle.

In this study the authors describe a theoretical and numerical analysis focused on the TC speed in a four stroke turbo-diesel engine. The purpose of this study is to highlight whether the TC speed allows one to detect the variation of the engine parameters. In addition, when the TC speed alone is not sufficient to detect the variation in the engine operating conditions, the proper set of additional, easy-to-measure, engine parameters is studied to univocally identify the causes of the variation. The analysis was carried out numerically by using a commercial 1D code calibrated with experimental data on a four cylinders diesel engine coupled with a turbocharger.

The numerical data was analyzed to estimate the optimal set of information required to identify the changed engine parameter that affects in particular the combustion process. By using the calibrated model, a sensitivity analysis of the detection parameter response, with particular attention to TC speed variations, was carried out by varying the operating conditions of the engine such as fuel mass flow rate, EGR rate and back pressure at the turbine outlet.

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INTRODUCTION

In order to ensure a high level of performance and to comply with more severe limitations in terms of fuel consumption and emissions reduction, a continuous supervision of the engine operating conditions is needed. In this context the monitoring of the engine thermodynamic cycle is the most suitable approach during both the development stage and the onboard operating conditions.

Direct analyses of the in-cylinder pressure are commonly performed at the test bench in order to optimize the engine thermodynamic cycle with particular attention to the combustion process. Indeed the in-cylinder pressure is one of the most important parameter for the combustion evaluation, but its direct measurement with the pressure sensor is intrusive and expensive.

Up to now, many studies on non-direct measurements for the evaluation of the quality of the engine thermodynamic cycle have been performed with the purpose to employ non-intrusive and low-cost sensors [1, 2, 3, 4, 5, 6, 7, 8, 9, 10, 11, 12].

Among these, the instantaneous crank angle speed measurement is a widespread methodology [1, 2, 3, 4, 5, 6, 7, 8]. In this case the quality of reconstructed thermodynamic cycle is affected by the high inertia of the engine moving components, particularly at high engine speed. In other cases the thermodynamic cycle derives from the engine vibration measurements [9, 10]. This kind of measures is affected by a low signal-to-noise ratio at low frequencies that prevents the correct reconstruction of the engine cycle.

Combining measuring techniques improves the accuracy of the cycle reconstruction but increases both the complexity of the acquisition system and the related costs. Examples of combined approaches are engine vibrations coupled with both engine speed monitoring [11] and acoustic acquisitions [12].

The growing use of turbocharger in automotive field and its direct connection with the thermo and fluid-dynamic conditions at the inlet and outlet of the engine suggest to use the average and instantaneous TC speed as a feedback of the engine operating conditions. In this regard its low inertia could entail a higher sensitivity to the engine cycle behavior.

In this paper a theoretical and numerical analysis of the turbocharger speed in a four stroke turbo-diesel engine is presented. The 1D numerical model of the engine was calibrated with experimental data. The model allows to take into account the TC characteristics both for the fluid-dynamic aspects (by using compressor and turbine working maps) and for the mechanical characteristics by introducing the TC inertia. With a sufficiently detailed numerical model it is possible to appreciate not only the average TC speed but also the angular velocity oscillations along the engine cycle. Therefore, a first approach was developed to determine whether it is possible to correlate the TC speed to the engine torque.

The aim of this study was to estimate qualitatively the TC speed sensitivity to the variation of the engine working parameters. However in some cases the TC speed alone could not be sufficient to univocally detect the causes of the variation in the engine operating conditions. In these cases the cause could be univocally identified by the determination of a proper set of additional, easy-to-measure, engine parameters.

By using the calibrated numerical model, a sensitivity analysis to detect the engine parameters response, with particular attention to TC speed, was performed by varying the operating conditions of the engine as the fuel mass flow rate, EGR rate

and back pressure at the turbine outlet. A further analysis of the numerical data allowed to estimate a proper set of information useful to identify the variation of the engine working parameters that affect in particular the combustion process.

THE TURBOCHARGER ENGINE NUMERICAL MODEL AND CALIBRATION

In order to carry out an extended sensitivity analysis on the whole operating range of the engine by keeping under control several engine parameters, a calibrated numerical model is certainly useful. Moreover, once calibrated, the numerical model follows the physics of the system without being affected by possible issues related to experimental measurements.

The tested engine is a 4S 2.1 l four cylinders direct injection turbo diesel for power generation (42 kW @2750 rpm), whose characteristics are summarized in [Table 1](#).

Table 1. Engine technical data.

N° Cyl	S. for cycle	Eng. Type	Bore (mm)	Stroke (mm)	Compr. Ratio
4	4	C. I.	86	90	19.2

The experimental campaign was performed by Yanmar. The in-cylinder pressure data was acquired with KISTLER 6125 dynamic pressure transducer, whereas for in-ducts pressure measurements KISTLER 4049 piezoresistive pressure sensor was used. The TC speed was measured using a Hall-effect commercial sensor. For confidentiality reasons, the authors cannot provide additional details.

The measure quantities and their accuracies are listed below:

- Torque and power (±1%)
- TC average speed (±0.2%)
- Air mass flow (±1%)
- Fuel mass flow (±0.12%)
- EGR rate (±1.3%)
- In-cylinder pressure (in all cylinders) (±0.5%)
- Intake and exhaust dynamic pressure (±0.1%)
- Air and exhaust gas temperatures (±0.5°C)
- Intake and exhaust wall temperatures (±0.5°C)
- Data from ECU (Engine Control Unit)

Pressure and temperature acquisition sensors were located as shown in [Figure 1](#). The large number of acquisition points guarantees a robust calibration of the numerical model.

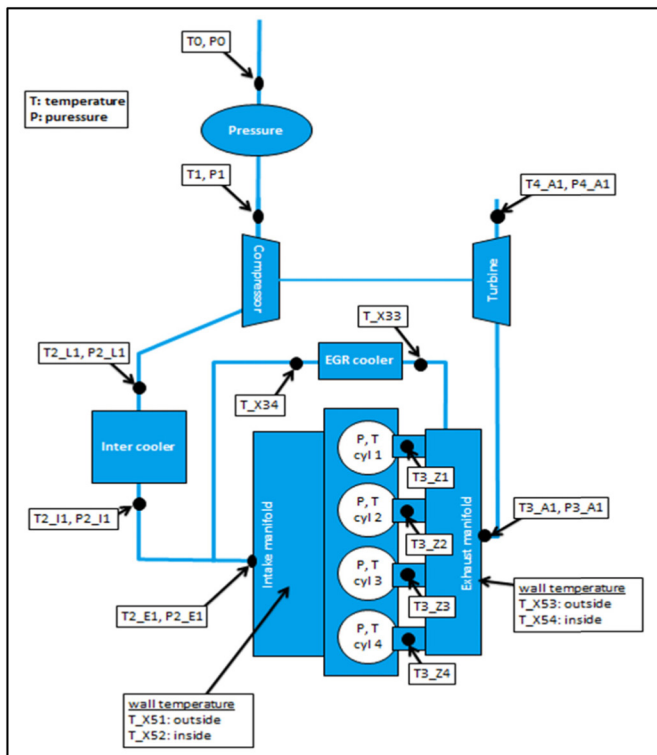


Figure 1. Engine scheme and experimental acquisition points.

The engine was tested at constant load with step of 250 rpm from 1000 rpm to the maximum rotating velocity (3000 rpm). These measurements were repeated in five load conditions: the full load and other four load levels at constant injected fuel quantity (Figure 2). Moreover, the friction mean effective pressure (FMEP) was measured in driven condition (Figure 3).

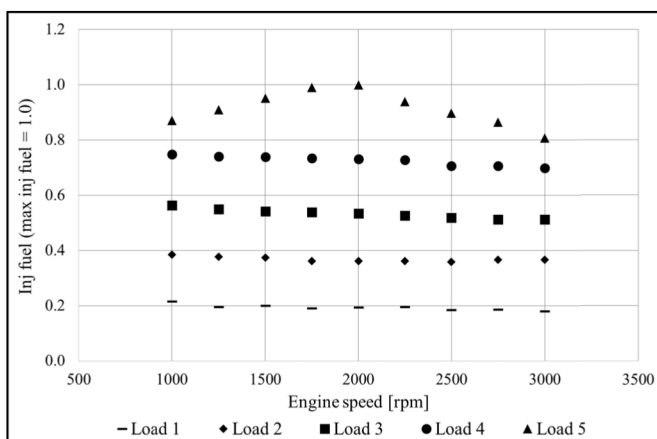


Figure 2. Injected fuel quantity at five load levels by varying the engine speeds from 1000 rpm to 3000 rpm (step of 250 rpm); the data is normalized with respect to the maximum injected fuel quantity.

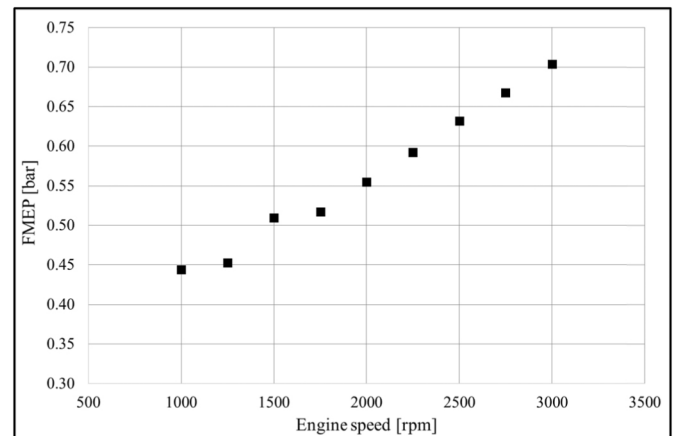


Figure 3. Experimental Friction Mean Effective Pressure measured on the driven engine.

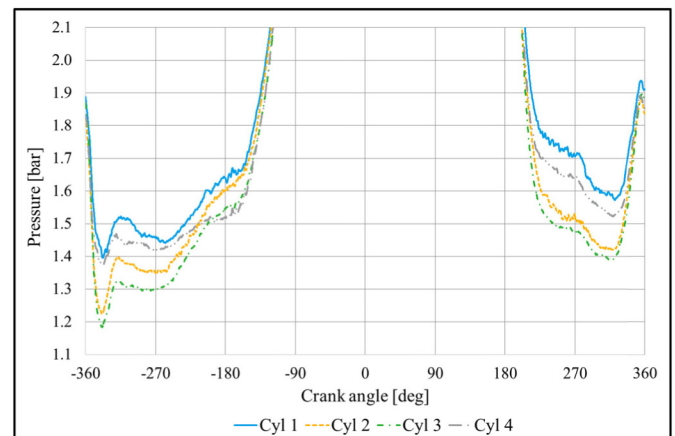


Figure 4. Experimental pressures inside the four cylinders (intake and exhaust phases) at 1750 rpm and full load (load 5).

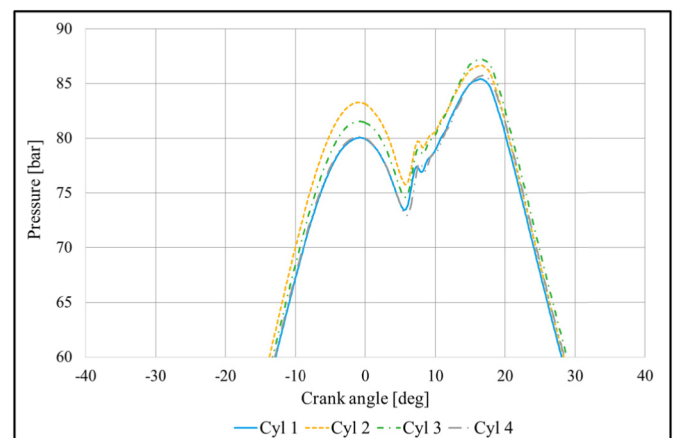


Figure 5. Experimental pressures inside the four cylinders (combustion phase) at 1750 rpm and full load (load 5).

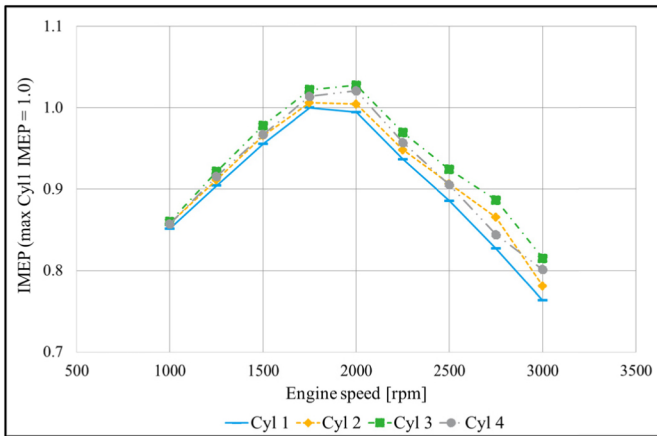


Figure 6. Experimental Indicated Mean Effective Pressures at the full load measured in each cylinder.

It is important to notice how the four cylinders do not perform exactly the same thermodynamic cycle (Figure 4 and Figure 5), due to the not perfect uniformity of the charge among cylinders, differences in charge motion inside each cylinder and uncertainty of the measurement. Therefore, for each engine operating condition there is a set of four experimental IMEP (Figure 6). Conversely, in the present study the model was set without including this kind of differences because evaluated negligible for the purposes of the sensitivity analysis. As a consequence the average cycle was considered as experimental reference.

The numerical model was built by using a multi-physics 1D code. Nowadays these codes have reached a high level of reliability and the model, once calibrated, is able to predict the engine performance and all its operating parameters. For the simulation of fluid dynamics inside ducts, the code solves the Navier-Stokes 1D equation by using a second order Lax-Wendroff scheme. The in-cylinder thermodynamic phenomena are simulated through the first principle equation, supported by dedicated equations used to calculate the combustion heat release and the heat exchanges. The numerical model calculates all the physical parameters (pressures, temperatures, fluid density, heat flows, etc.) for the whole modelled domain, and processes this data to compute all the engine performance parameters (torque, power, BMEP, volumetric efficiency, etc.). The numerical model needs an appropriate calibration in order to be reliable.

In the first phase of the numerical activity, the model was built by referring to the engine technical data including engine geometrical characteristics, valves lift and their flow coefficients. A dedicated experimental campaign for turbine and compressor characterization was carried out in order to introduce the TC maps in the numerical model. Indeed the TC model uses interpolated maps based on experimental data. In order to consider the influence of inertia on the instantaneous TC speed, the mechanical characteristics of TC were introduced in the numerical model.

In order to match the experimental volumetric efficiency the following calibration parameters were set: wall temperatures, heat and friction coefficients, waste-gate and EGR valves opening.

To simulate the behavior of the turbine waste-gate, an actuator varies the valve opening with a control feedback. This control regulates the waste-gate position in order to limit the pressure downstream the compressor below a reference experimental value.

The EGR valve opening was not experimentally measured; as a consequence a PID control varies the section of EGR valve until the experimental EGR rate is reached.

In this calibration phase, in order to properly consider the influence of the combustion on the charge exchange phase, the imposed combustion sub-model was used and the experimental heat release rate was imposed.

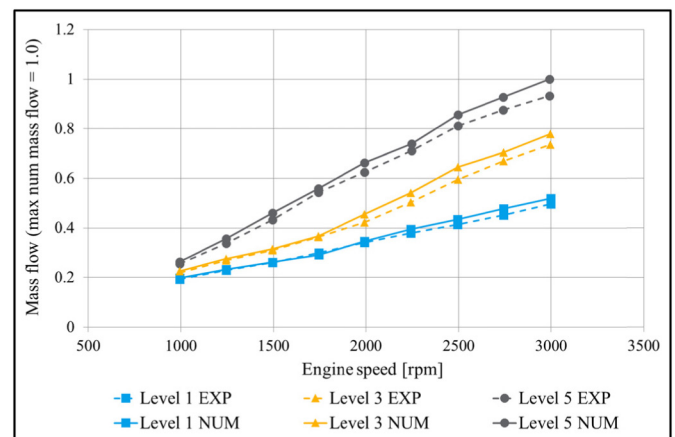


Figure 7. Intake air mass flow at three different load levels (numerical and experimental data); the data is normalized with respect to the maximum numerical air mass flow.

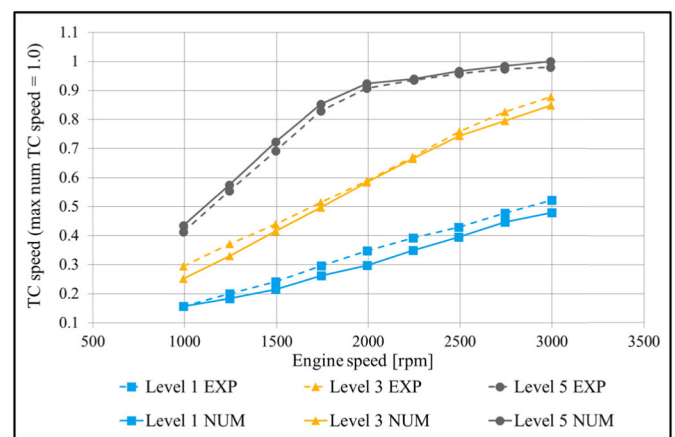


Figure 8. Average TC speed at three different load levels (numerical and experimental data); the data is normalized with respect to the maximum numerical average TC speed.

Because of their strict connection with the engine volumetric efficiency, also the intake and exhaust dynamic pressures were compared with the corresponding experimental data. For brevity of discussion the main significant comparisons between experimental and numerical data are shown in order to highlight the good predictability of the 1D model (from Figure 7, Figure 8, Figure 9, Figure 10).

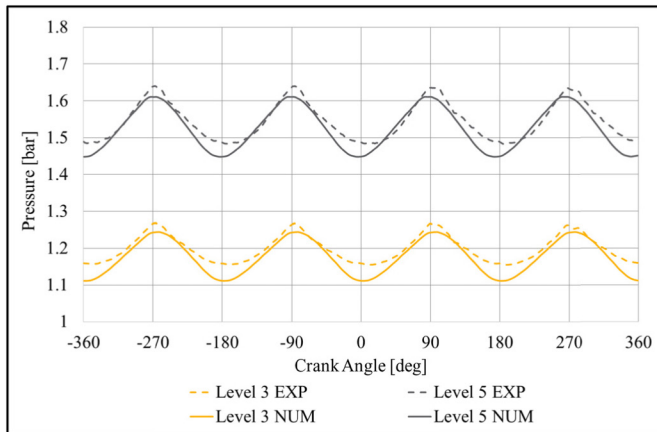


Figure 9. Dynamic pressure at the compressor outlet at two different operating conditions (load level 3 and 5 at 1750 rpm); comparison between numerical and experimental results.

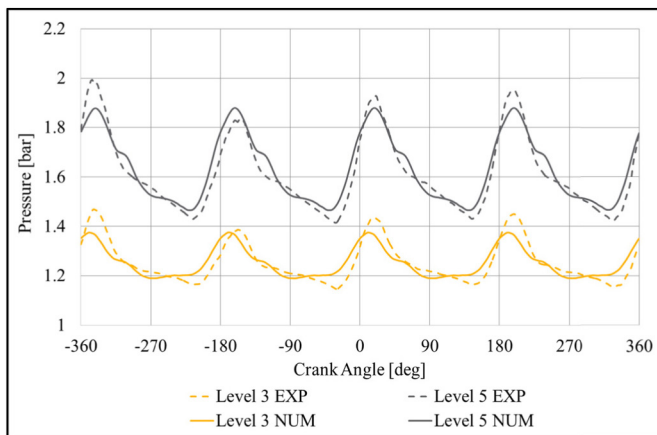


Figure 10. Dynamic pressure at the turbine inlet at two different operating conditions (load level 3 and 5 at 1750 rpm); comparison between numerical and experimental results.

Once the volumetric efficiency was calibrated, the focus of the model calibration shifted to the combustion process. The semi-predictive Diesel combustion sub-model was tuned. The main calibration parameters are the injection profile (defined by the experimental measures of the fuel mass per cycle, the start and the end of the injection), the heat transfer inside the cylinder during combustion and the coefficients that regulate the weight of premixed and diffusive combustion phases. The experimental reference data were the heat release rate, the in-cylinder pressure and finally the indicated mean effective pressure (Figure 11 and Figure 12). It is to be highlighted that during the model validation, the authors aimed to define a calibration as valid as possible over the whole operating range

and set of parameters of the engine. This issue justifies the small differences between numerical results and experimental data for some engine speeds and loads.

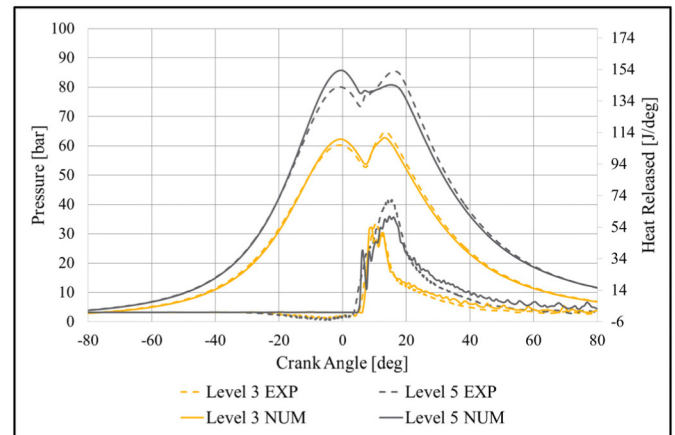


Figure 11. In-cylinder pressure and heat released at two different operating conditions (load level 3 and 5 at 1750 rpm); comparison between numerical and experimental results.

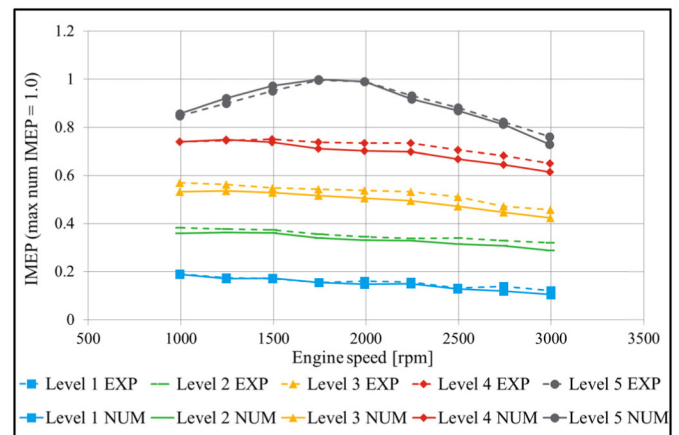


Figure 12. Indicated Mean Effective Pressure at five different load levels (numerical and experimental data); the data is normalized with respect to the maximum numerical IMEP.

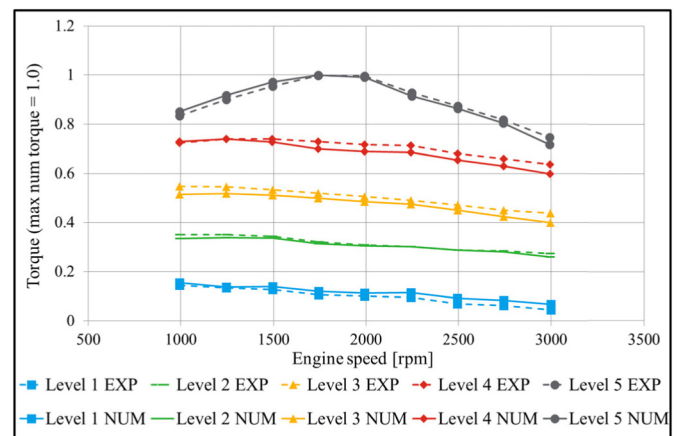


Figure 13. Engine torque at five different load levels (numerical and experimental data); the data is normalized with respect to the maximum numerical torque.