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Development of a torsiometer for on-board application

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Abstract

Modern combustion control strategies require accurate combustion control to meet the requirements for pollutant emissions reduction. Optimal combustion control can be achieved through a closed-loop control based on indicated quantities, such as engine torque and center of combustion, which can be directly calculated through a proper processing of in-cylinder pressure trace. However, on-board installation of in-cylinder pressure sensors is uncommon, mainly because it causes a significant increase in the cost of the whole engine management system.

In order to overcome the problems related to the on-board installation of cylinder pressure sensors, this work presents a remote combustion sensing methodology based on the simultaneous processing of two crankshaft speed signals. To maximize the signal-to-noise ratio, each speed measurement has been performed at opposed ends of the crankshaft, i.e. in correspondence of flywheel and distribution wheel. Since an engine speed sensor, usually faced to the flywheel, is already present on-board for other control purposes, the presented approach requires that an additional speed sensor is installed. Proper processing of the signals coming from the installed speed sensors allows extracting information about crankshaft's torsional behavior. Then, the calculated instantaneous crankshaft torsion can be used to real-time estimate both torque delivered by the engine and combustion phasing within the cycle. The presented methodology has been developed and validated using a light-duty L4 Common-Rail Diesel engine mounted in a test cell at University of Bologna. However, the discussed approach is general, and can be applied to engines with a different number of cylinders, both CI and SI.

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Keywords: "Indicated torque; Closed-loop; Combustion control; Torsiometer"

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1. Introduction

Optimal combustion control is a crucial aspect in modern internal combustion engines, due to the increasing request for pollutant emissions reduction and engine efficiency increase. In particular, control strategies based on feedbacks that provide information about combustion effectiveness allow on-board optimization of engine mapping, which is suitable to extend the engine operating range and compensate aging effects [1,2].

Closed-loop combustion control strategies usually rely on combustion indexes that can be directly calculated through a proper processing of in-cylinder pressure trace [3-8]. However, in-cylinder pressure measurement is usually unavailable on-board, mainly due to both high cost and low long-term reliability of in-cylinder pressure sensors. One of the most important quantities in a closed-loop combustion control strategy is torque delivered by cylinders. For the discussed reasons, it is interesting to investigate remote combustion sensing methodologies, i.e. based on the use of low-cost sensors applied to the engine, for the estimation of engine torque. Many approaches widely discussed in literature are based on the real-time analysis of signals coming from accelerometers applied to the engine block [9-11], engine speed sensors [12-15] or crankshaft torque sensors [16-20].

In order to achieve accurate torque estimation, this paper presents a methodology based on the real-time analysis of two speed measurements. Since one engine speed measurement is already performed, usually in correspondence of the flywheel, for engine phasing and other control purposes, this approach requires the installation of a second crankshaft speed sensor. In this work, the second speed sensor has been installed in correspondence of the distribution wheel.

The presented approach has been developed for a 4 cylinders Common-Rail 1.3L Diesel engine installed in a test cell. However, the methodology is general and it can be applied to engines with different powertrain configurations.

Nomenclature

I_m	*m*-th inertia of the engine-dyno model
J	Total inertia
t_{n_D}	Time in which the *n*-th tooth of the distribution wheel appears in front of the pick-up
t_{n_F}	Time in which the *n*-th tooth of the flywheel appears in front of the pick-up
v_n	Instantaneous engine speed, corresponding to the *n*-th tooth
$\Delta\theta_n$	Angular distance between the corresponding *n*-th teeth of flywheel and distribution wheel
$\Delta\Theta$	Average crankshaft angular torsion (over the engine cycle)
Ω	Average engine rotational speed (over the engine cycle)
T_i	Average indicated torque (over the engine cycle)
ω_k	*k*-th eigen-frequency of the engine-dyno system
ORD	Engine order
imep	indicated mean effective pressure

2. Experimental setup

The engine used to develop and validate the presented approach is a 1.3L Common-Rail Diesel engine mounted in a test cell at the University of Bologna. Table 1 summarizes the technical characteristics of the engine.

Table 1. Engine characteristics.

Displaced volume	1248 cm^3
Compression ratio	16.8
Maximum torque	200 Nm @ 1500 rpm
Maximum power	70 kW @ 3800 rpm
Number of valves	4 per cylinder

Injection system	Common Rail Multi-Jet

The engine under investigation is equipped with one standard 60-2 phonic wheel used for engine speed measurement and other control purposes. Since the presented approach is based on the simultaneous analysis of two instantaneous crankshaft speed measurements, an additional phonic wheel has been coupled to the distribution. In addition, since standard phonic wheels allow obtaining a relatively low angular resolution (approximately 6 degrees in this case), while the goal of the activity is to analyze the crankshaft speed signals in detail, two additional optical encoders faced to 180-2 toothed wheels have also been fixed in correspondence of flywheel and distribution wheel. With regard to the optical sensor, two Sensor Instruments GmbH FIA F-0.6 have been chosen. Even though cost and reliability of these sensors is not compatible with the requirements for on-board application, the high accuracy allows analyzing in detail the torsional vibrations of the engine-driveline system. Figure 1 reports the optical sensors installation. The installed optical sensors return a digital output (0-12 V) with a negligible delay. In order to accurately measure the time intervals between encoder teeth, tooth times have been measured using a specifically designed data acquisition system based on a National Instruments cRIO and a Digital Input (DI) 9401 board. Such system allows sampling time intervals between tooth transitions at 20 MHz; the measured intervals can be easily used to calculate the instantaneous rotational speed.

Once the data acquisition system has been set up, several experimental tests have been carried out running the engine over its whole operating range both in steady-state and transient conditions.

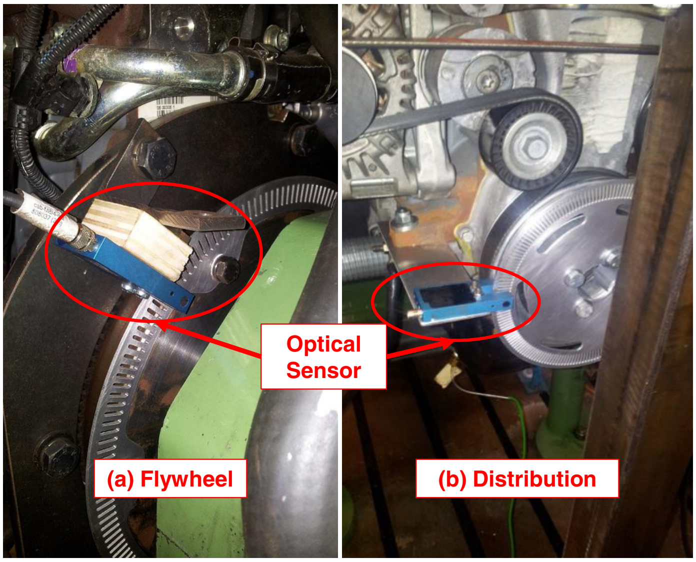

Fig. 1. (a) Installation of the optical encoder coupled to the flywheel; (b) installation of the optical encoder coupled to the distribution wheel.

The main idea behind this work is to set up a methodology for real-time torque estimation, based on the real time analysis of crankshaft torsion. Since crankshaft torsion is monitored using two speed sensors, the location of the additional measurement (i.e. the distribution wheel) has been chosen to maximize the signal-to-noise ratio.

Figure 3 reports the instantaneous speed signals, measured respectively in correspondence of flywheel and distribution, for a steady-state test run at 2000 rpm and imep = 19 bar. As it can be observed, both signals are affected by a high frequency noise, mainly due to imperfections that regular crank target wheels usually present.

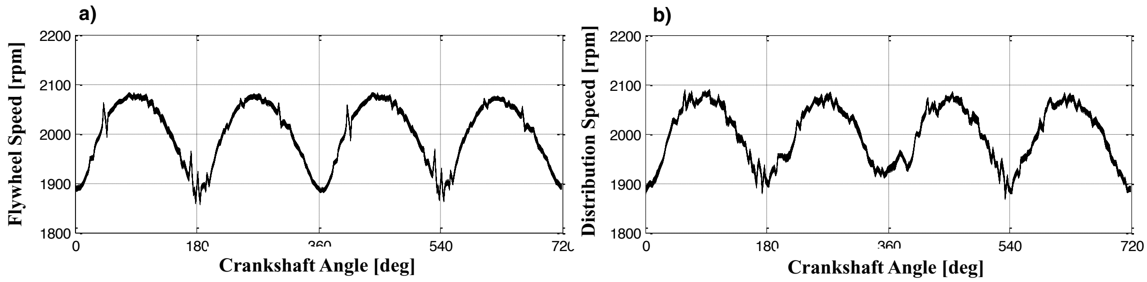

Fig. 2. Engine speed signal over the engine cycle calculated using the speed sesor installed in correspondence of flywheel (a) and distribution wheel (b).

In addition, from the observation of Figure 2 arises that the speed signal measured in correspondence of the distributions wheel seems to be "noisier" with respect to the one calculated in correspondence of the flywheel. In order to investigate the differences between speed measurements, the frequency spectra of the two signals have been compared. Figure 3 shows, for both signals, the engine speed frequency content up to order 10. Although the spectra appear similar and mainly characterized by a high frequency content in correspondence of the engine order 2 (the engine performs 4 evenly-spaced combustions per cycle), the spectrum of the speed signal measured in correspondence of the distribution wheel shows a significant frequency content approximately at 337 Hz.

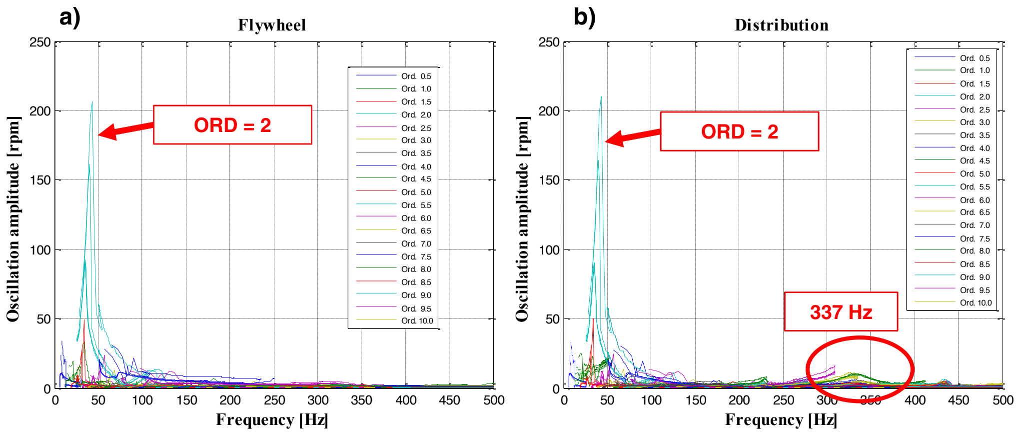

Fig. 3. Comparison between frequency spectra of the speed signals measured in correspondence of flywheel (a) and distribution (b).

The frequency content at 337 Hz is completely absent in the flywheel signal. Consequently, understanding the source of this energy contribution is not trivial. In particular, it is important to understand whether this energy is due to the torsional characteristics of the engine-dyno system under investigation or it is a "real noise", i.e. a resonance of the connection between crankshaft and additional speed sensor (distribution side). To do so, a lumped model of the engine-dyno has been developed. In particular, the system under investigation has been schematized using 7 inertias connected with stiffness and damping. Figure 4 reports a scheme of the developed model.

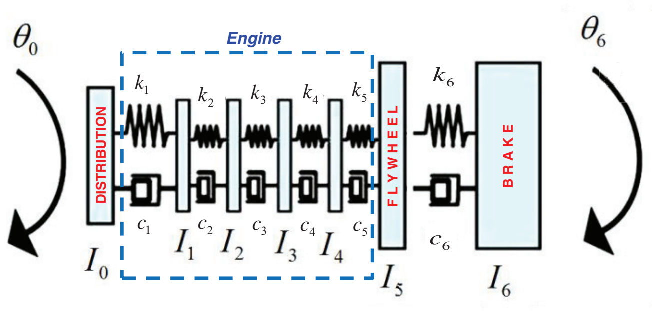

Fig. 4. Scheme of the engine-dyno lumped model.

Here, the reported 7 inertias represent:

- I_0 : distribution;
- I_1 : cylinder 1;
- I_2 : cylinder 2;
- I_3 : cylinder 3;
- I_4 : cylinder 4;
- I_5 : flywheel;
- I_6 : brake.

The torsional characteristics of the model, i.e. the values of inertias, stiffness and damping, have been determined starting from a CAD representation of the crankshaft and the manual of the brake. Once the discussed linear model has been established, the eigen-frequencies of the system can be calculated. In particular, the evaluation of the first 3 natural frequencies demonstrates that the frequency content at 337 Hz is due to the second natural frequency of the system (ω_2):

- $\omega_1 = 32\,[Hz]$
- $\omega_2 = 337\,[Hz]$
- $\omega_3 = 1200\,[Hz]$

In addition, the calculation of the first 3 eigen-values, reported in Figure 5, demonstrates that the oscillation at 337 Hz is due to a natural frequency of the crankshaft. As it can be observed in Figure 5, the effect of the 337 Hz oscillation is very high in correspondence of the first inertia (distribution wheel), while it is approximately equal to zero in correspondence of the sixth inertia (that represents the flywheel). Consequently, the energy content at 337 Hz significantly affects the engine speed measurement in correspondence of the distribution wheel, while it is absent in correspondence of the flywheel.

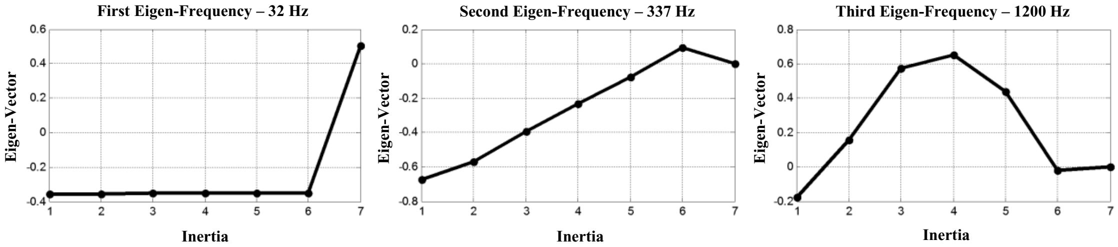

Fig. 5. First 3 eigen-vectors of the engine-dyno system.

Due to previous considerations, engine speed measurement performed in correspondence of the distribution is affected by a "noise" which is directly caused by the torsional characteristics of the engine-dyno system. It is reasonable to expect that such noise will affect the correlations between the two instantaneous speed measurements and torque delivered by the engine.

3. Correlation between double speed measurement and indicated torque

Bearing in mind the problems related to speed measurement in correspondence of the distribution, the double speed measurement has been analyzed to extract information about the correlations between crankshaft torsion and torque delivered by the engine.

First, the instantaneous angular torsion has to be calculated from the acquired speed signals. To do so, each tooth in the flywheel has been associated to a corresponding tooth in the distribution wheel. Since both instantaneous engine speed (v_n, corresponding to the n-th tooth) and the time in which the n-th tooth of each wheel appears in front of the pick-up (t_n) can be determined, the angular distance between the corresponding n-th teeth of flywheel and distribution wheel can be easily calculated through Equation (1).

$$\Delta\theta_n = v_n \cdot \left(t_{n\_F} - t_{n\_D}\right) \tag{1}$$

The angular distance calculated through Eq.(1) is affected by tooth spacing errors. However, the mean $\Delta\theta_n$, evaluated over a complete engine cycle, automatically compensates tooth spacing errors and returns the mean angular distance between flywheel and distribution wheel. It is reasonable to expect that a linear correlation between indicated torque average value and average angular torsion (both calculated over an engine cycle) can be set up. Bearing in mind that the effect of engine speed variation is not negligible, such correlation can be written as reported in Equation (2).

$$T_i = K \cdot \Delta\Theta - J\frac{d\Omega}{d\theta} \tag{2}$$

Here, T_i represents indicated torque delivered by the engine, $\Delta\Theta$ is the mean angular torsion and Ω is engine rotational speed. The linear correlation expressed in Eq.(2) has been identified through a set of specifically designed tests, performed running the engine at different variable speed and load, both in steady-state and transient conditions. The obtained results are reported in Figure 6. As it can be observed, even if mean angular torsion is very small (usually lower than 0.1 deg), the obtained correlations are very strong and suitable for real-time torque estimation.

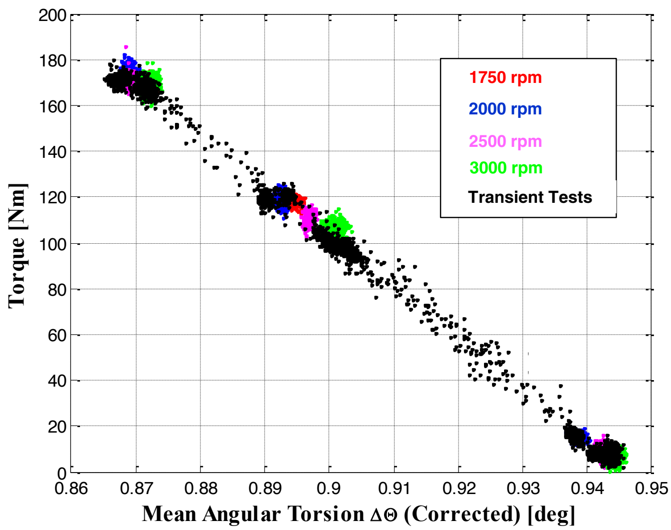

Fig. 6. Measured Indicated Torque (mean value) vs measured angular torsion.

Once the correlation shown in Figure 6 has been determined, it can be used to set up an algorithm for real-time indicated torque estimation. The developed methodology has been applied to the engine under investivation and proved to be suitable for real-time torque estimation, both in steady-state and transient conditions. The accuracy of the obtained results seems to be compatible with the requirements for closed-loop combustion control. Figure 7 reports the result obtained applied the estimation methodology to a test run at 2000 rpm and variable engine load.

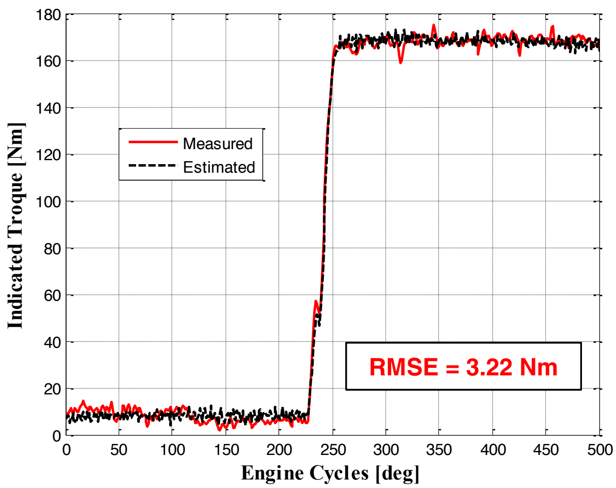

Fig. 7. Comparison between measured and estimated torque.

4. Conclusions

The increasing request for pollutant emissions reduction has generated a great amount of research in the field of combustion control optimization. Optimal combustion control strategies allow improving engine performance based on the real-time estimation of quantities that provide information about combustion effectiveness, such as indicated torque delivered by the engine.

Indicated torque can be directly calculated starting from in-cylinder pressure trace, that is usually unavailable on-board, mainly due to high cost and low reliability of in-cylinder pressure sensors. Based on double speed measurement performed using two speed sensors, each one installed at opposed ends of the crankshaft, this work presents a remote combustion sensing methodology for indicated torque estimation.

The obtained results demonstrate that the performed double speed measurement allows obtaining an accurate evaluation of the instantaneous crankshaft angular torsion, that is strongly correlated to indicated torque delivered by the engine. The identified correlation can be used to set up an algorithm for real-time indicated torque estimation.

The algorithm has been applied to a 1.3L Common-Rail Diesel engine mounted in a test cell. The methodology proved to be suitable for real-time torque evaluation, while the accuracy of the obtained results seems to be compatible with the requirements for closed-loop combustion control.

Further investigations are being performed now to implement the developed algorithm in real-time and use it on-board a vehicle properly equipped with 2 standard engine rotational speed sensors for on-board application.

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