

Instantaneous Engine Speed Measurement and Processing for MFB50 Evaluation

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ABSTRACT

Evaluation of MFB50 is very useful for combustion control, since it gives an evaluation of the combustion process effectiveness. Real-time monitoring its value enables to detect for example the kind of combustion that is taking place (useful for example for HCCI applications), or could provide important information to improve real-time combustion control. While it is possible to determine the position where the 50% of mass burned inside the cylinder is reached using an in-cylinder pressure sensor, this work proposes to obtain this information from the engine speed fluctuation measurement. In-cylinder pressure sensors in fact are still not so common for on-board applications, since their cost will constitute an important portion of the whole engine control system cost. Engine speed measurement is instead already performed in modern engine control systems and therefore being capable of obtaining MFB50 related information from this signal means obtaining it at approximately no additional costs.

The MFB50 estimation procedure presented in this paper is based on the measurement of the engine speed fluctuations and it mainly consists in two separated steps. As a first step, a torsional behavior model of the powertrain configuration is developed. The engine-driveline torsional model enables to estimate the indicated torque frequency components, from the corresponding components of the instantaneous engine speed fluctuation. This estimation can be performed cycle by cycle and cylinder by cylinder. As a second step, the analysis of the relationship between MFB50 and the phase of the frequency components over an engine cycle allowed defining the final estimation algorithm that reconstructs the MFB50 starting from the instantaneous engine speed fluctuation analysis. The developed approach has been applied with success to a diesel engine mounted on-board a vehicle.

INTRODUCTION

Modern engine control strategies require the estimation of an increasing number of quantities, in order to respect all the management requirements concerning, for example, drivability and pollutant limitation [1-2]. MFB50 is one of the most interesting parameters to evaluate on board, because an accurate estimation of its value gives a useful information about combustion. As already mentioned, the position where the 50% of mass burned inside the cylinder is reached, can be directly determined using an in-cylinder pressure sensor, which cannot still be used for on-board applications because of its unreliability and cost.

The MFB50 estimation approach used in this paper does not imply any additional cost, because it only requires the instantaneous engine speed measurement, that can be performed using the same magnetic pick-up and the same 60-2 teeth wheel already present on the vehicle for other control aims. This method showed to be robust also to the small variations in the tooth spacing that usual crank target wheels normally present, as it will be mentioned in the following.

The first step of the procedure consists in developing a torsional behavior model of the powertrain, in order to determine the relationship between the instantaneous engine speed fluctuations components and the corresponding indicated torque harmonic components. Many studies [3-9], some of them by the same authors of this paper [10-12], demonstrate the feasibility of torque estimation (frequency components and mean value) starting from engine speed measurement, because of the strong relationship between speed and torque fluctuations. In particular fluctuation measurement is needed at an order characteristic of the engine (mainly related to its number of cylinders and its firing order); at this order the speed waveform over an engine cycle shows fluctuation of the highest amplitude. Measurement of the speed fluctuation at the engine order of interest allows for estimating the corresponding

indicated torque fluctuations at the same engine order. This can be done if an accurate torsional behavior model of the engine-driveline-load system has been developed. The relationship between instantaneous engine speed harmonic components and torque harmonic components can be in this way described using a transfer function, that only depends on the structure of the engine-driveline system.

The second step of the procedure consists in determining a relationship between the phase of the torque frequency component of interest over an engine cycle and MFB50. Knowledge of this relationship enables to estimate MFB50 starting from engine speed fluctuations measurement.

The main steps of the whole estimation algorithm are schematically presented in Figure 1.

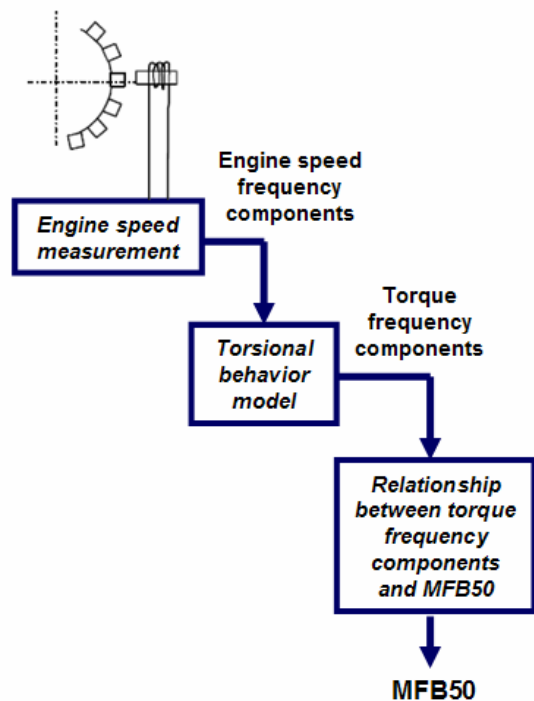


Figure 1: Scheme of the MFB50 estimation procedure described in this paper.

The first part of the algorithm, i.e. the determination of a transfer function (only depending on the engine-driveline configuration) that describes the relationship between the instantaneous engine speed frequency components and the corresponding components of the indicated torque, will be briefly described in next paragraph. For a deeper knowledge about the development of a torsional behavior model of the engine-driveline system, see references [10-12].

The following paragraphs will then describe the representation of the combustion process that has been

used in order to investigate the relationship between the phase of the torque frequency component and MFB50.

Finally some results have been presented, that have been obtained applying the proposed approach to an L4 multi-jet diesel engine.

TORQUE FREQUENCY COMPONENTS ESTIMATION

As already mentioned, the first step of the procedure consists in estimating the indicated torque harmonic component of interest, starting from the engine speed fluctuations. In order to estimate torque, only the engine speed fluctuations at an engine order characteristic of the engine under study, have to be investigated. The engine order to be considered depends on the number of cylinders and on the firing order, and is the one at which the engine speed fluctuations show the higher amplitude. For a Z cylinders engine with evenly-spaced combustions, for example, the order to be taken into account is $Z/2$, corresponding to the firing frequency. If the combustions are not evenly spaced the choice of the order could be more difficult, but still it is possible to determine the one at which the response in terms of engine speed fluctuations is expected to be higher. Since the interest is focused on the speed fluctuations over an engine cycle rather than to the instantaneous values, this methodology will be robust with respect to the tooth spacing variations that usually the crank target wheels present.

Once the engine order of interest has been selected, it is possible to determine the frequency range in which the torsional model has to correctly describe the dynamic behavior of the system. For a 4 cylinder engine with evenly spaced combustions (order 2 is selected), running from 800 to 4500 rpm, the frequency range to be investigated goes from 27 to 150 Hz. The evaluation of the interesting frequency range is important for the correct definition of the dynamic model, since all the natural frequencies of the system out of this range can be neglected. Following the example of the 4 cylinder engine, if the lowest natural frequency on the crankshaft is higher than 150 Hz, it can be neglected.

Either the engine is mounted on-board a vehicle or it is mounted on a test bench, the lowest resonance frequencies associated to the driveline are usually within the frequency range of interest. These conditions (crankshaft eigen-frequencies out of the frequency range of interest, and one or more driveline natural frequencies within this range) have been verified for all the powertrain configurations investigated and is the main hypothesis of this study. If this condition is not verified restrictions in the application of the proposed approach have to be considered.

Based on the previous considerations, only the natural frequencies on the driveline will be taken into account. The system torsional behavior can be described by

means of an equivalent model with 2 or more inertias, one of them representing the engine, the number of inertias to be employed depending on the number of natural frequencies of the driveline, up to the frequency of interest. Figure 2 shows the simplest case with 2 inertias.

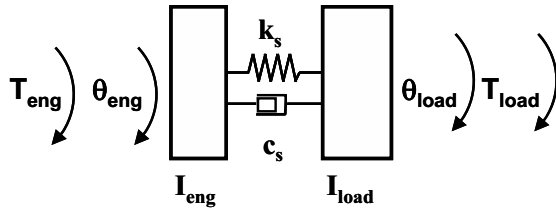


Figure 2: Scheme of the inertia-stiffness equivalent representation to describe the system dynamic behavior

Considering for the sake of simplicity the model with two inertias, dynamic equations of the system can be obtained as:

$$\begin{cases} I_{eng} \ddot{\theta}_{eng} = T_{eng} - k_s (\theta_{eng} - \theta_{load}) - c_s (\dot{\theta}_{eng} - \dot{\theta}_{load}) \\ I_{load} \ddot{\theta}_{load} = T_{load} - k_s (\theta_{load} - \theta_{eng}) - c_s (\dot{\theta}_{load} - \dot{\theta}_{eng}) \end{cases} \quad (1)$$

Applying Fourier expansion to the signals in Equation (1) and neglecting the harmonic content of the load torque T_{load} (always negligible because of its slow dynamics with respect to an engine cycle duration), and rearranging all the terms, it is possible to obtain the following expression, where the relationship between torque and speed has been considered:

$$\dot{\theta}_{engi} = \frac{1}{\omega} \frac{I_{load} \omega^2 - j\omega c_s - k_s}{j(I_{eng} I_{load} \omega^2 - (I_{eng} + I_{load})(j\omega c_s + k_s))} T_{engi} \quad (2)$$

Equation (2) has been obtained for the simplest 2 inertias model case. Similar equations can be obtained if more complex torsional models have to be designed to correctly describe the system. The obtained equation will present in the case of a more complex system a higher number of parameters (inertias, stiffness and dampings). In any case it will be possible to describe the relationship between the corresponding Fourier expansion coefficients of torque and speed using a Frequency Response Function (FRF), reported in the following as $F(j\omega)$.

$$\dot{\theta}_{engi} = \frac{1}{\omega F(j\omega)} T_{engi} \quad (3)$$

The Frequency Response Function $F(j\omega)$ represents the dynamic behavior of the system, and depends on the inertia-stiffness configuration of the system itself. Inertias and stiffness values can be determined starting from a CAD representation of the system, even if this approach is quite time consuming and requires full access to

component CAD representation [10-11]. In addition, once FRF has been obtained, usually it has to be verified and validated through specific experimental tests.

As an alternative, the FRF can be experimentally determined using properly designed engine-driveline tests, through a rearrangement of Equation (3).

$$F(j\omega) = \frac{T_{engi}}{\omega \dot{\theta}_{engi}} \quad (4)$$

Using Equation (4) it is possible to obtain from engine measurement an estimate of the FRF to be used on-board. Amplitude and phase of the engine speed oscillation can be in fact directly measured and, for the engine order of interest, it results:

$$\dot{\theta}_{engi} = \Omega_{ORD} e^{j\phi_{ORD}} \quad (5)$$

Engine torque T_{eng} is the result of all the torques applied to the cylinders, i.e. the difference between indicated and reciprocating torques. The same will apply also to the i -th harmonic component of the indicated and reciprocating torques, that can be seen as the sum of the i -th harmonic components for all the cylinders. Considering indicated and reciprocating torque waveforms similar for all the cylinders, the Fourier representation of the torque harmonic components will differ only for the phase, whose value can be determined knowing the number of cylinders and the firing order (Equation (6) reports as an example what can be obtained for a Z cylinders engine with evenly spaced combustions).

$$\begin{aligned} T_{indmi} &= T_{ind1i} \cdot e^{jim2\pi/Z} \\ T_{rmi} &= T_{r1i} \cdot e^{jim2\pi/Z} \end{aligned} \quad (6)$$

T_{ind1i} is the i -th indicated torque harmonic component for cylinder 1; Indicated torque can be evaluated from the in-cylinder pressure measurement for the same cylinder, through Equation (7) [10]:

$$T_{ind1}(\theta_1) = A_p r p_1(\theta_1) f(\theta_1) \quad (7)$$

Reciprocating torque can be expressed using the well known geometric expression [13]:

$$\begin{aligned} T_{rm}(\theta_m) &= M_{eq} r^2 f(\theta_m) \left[f(\theta_m) \ddot{\theta}_m + \frac{df(\theta_m)}{d\theta_m} \dot{\theta}_m^2 \right] = \\ &= M_{eq} \hat{T}_m(\theta_m) \end{aligned} \quad (8)$$

where M_{eq} is the value of the reciprocating masses, and $\hat{T}_m(\theta_m)$ represents the normalized reciprocating torque with respect to the reciprocating masses.