TOTAL LOSS TORQUE WAVEFORM ESTIMATION FOR A TURBOCHARGED DIESEL ENGINE.

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Abstract:

The indicated (gas) torque is a key parameter for reciprocating engine. It reports on various aspects of the engine, such as the quality of the combustion process, the fuel combustion rate and the average effective pressure. This is a good indicator of the stability of the engine at work. The form of the instantaneous indicated torque is precious information for the control and diagnosis of the engine. The study of the simulation of crankshaft dynamics illustrates the need to know the total loss torque (i.e. the addition of friction torque, auxiliary torque and load torque) to achieve the indicated torque. To date, it has proven difficult to obtain an accurate predictive expression of the instantaneous total loss torque. This expression includes adequate simulation of friction of all moving parts of the engine (piston ring and skirt, crankshaft bearings, valve gear...), auxiliaries and load. Based on experimental results conducted on a four-cylinder turbocharged diesel engine, this work offers a simple and practical solution for estimating the total torque loss. The proposed method makes it possible to successfully estimate total torque losses under a wide range of engine operating conditions. The result prediction has been confronted to experimental data.

1 Introduction

Modeling the engine performances has received much research attention. The built models help improvement of the engine design and its operating condition simulation. In addition, due to the rapid advances of computer technology, it becomes possible to use models for real time applications, such as control and fault diagnosis. However, there needs further research in modeling techniques, to improve predictive accuracy, before models will be used for estimation and robust control tasks. Knowledge of indicated torque (gas torque) generated during the combustion process is desirable for several key applications, such as idle speed control and engine diagnosis. While there exist in-cylinder pressure sensors, and there is a direct correlation between in-cylinder pressure and indicated torque, these sensors are generally deemed too expensive and impractical for implementation in a serial automotive engine production [1]. Sensors such as this are not cost-effective either in manufacture or installation or in maintenance because of the decidedly high pressures to be measured. This disadvantage is even greater in internal combustion engines with a large number of cylinders. Torque sensors, while available, generally have complicated construction, and their real-world production line performance and reliability are yet to be established. Consequently, there is much investigation into methods that will allow the indicated torque to be obtained from available measurements.

The crankshaft speed fluctuations contain information about the combustion in the cylinders. The angular speed signal has been found as a key measurement due to its convenience, its consistency and based on the physics of the engine process [2]. This measured parameter was largely exploited. Various studies have estimated indicated torque and eventually in-cylinder pressure using crankshaft angular speed measures injected in a rigid-body or elastic-body engine model built from the second Newton's law [3,4]. The indicated torque reconstruction from crankshaft speed fluctuations can be achieved with two main approaches, mathematical models and database-based methods like explained in the reference [5,6]. J. Franco et al. [7]

reported several methods for estimating the indicated torque to overcome the difficulties encountered to reach the target.
They cite:

- The Frequency Response Functions (FRFs) method used to correlate in-cylinder pressure and crankshaft angular speed;
- The Kalman-filter-based deconvolution;
- The Discrete Fourier Transforms employed to show the relationship between indicated torque and crankshaft angular speed;
- The 'stochastic method" based on least squares approach to statistically correlated variables.

Alexander Stotsky [8] used a new filtration algorithm based on Kaczmartz's projection method to reconstruct combustion quality information from noise-contaminated engine speed measurements. All these methods use advance means to reconstruct the indicated torque by possibly following the pressure in the cylinder. However, some errors persist due to noise in real operation, especially when using the FRF method to reconstruct the pressure waveform, as the FRF matrix is inherently poorly conditioned. A small amount of noise in the response data gives spectacular results, especially depending on FRF anti-resonances. In addition, there are problems with restoring the shape of the pressure signal from speed fluctuations. In fact, as R. Johnsson mentioned [6], regardless of the reconstruction method used to estimate the indicated torque, there are problems and limitations when using the crankshaft angular velocity as an input variable. First of all, the engine torque is provided by a gas torque component (pressure torque) and an inertia torque component. As the speed increases, the inertial torque caused by the acceleration and deceleration of the piston and its connection links with crankshaft becomes more and more important and thus masks the contribution of the gas torque. Secondly, when determining the fluctuations in the torque of the motor, it is necessary to consider the fact that the motor is simultaneously connected to its load by elastic elements and is itself an elastic unit.

The conclusion to be drawn from these limitations is that the reconstruction method used must be able to consider non-linearities and to use different relationships between fluctuations in crankshaft velocity and gas pressure in each cylinder. This article discusses total loss torque modeling to create a simple method to predict the torque of a gas from measuring fluctuations in crankshaft angular velocity. Experimental measurements are carried out on a turbocharged 4-cylinder diesel engine on a test bench.

2 Engine crankshaft dynamic modeling

Modeling of the crankshaft dynamics requires a good compromise between accuracy and computing time. In this work, the rigid-body model is considered due to its simplicity and low computation cost. The rigid-body model is derived from the multi-body model [9]. All the inertias of the multi-body model are lumped together in the rigid-body model providing the corresponding single degree of freedom described by the equation (1), instead of the complete equation of dynamics integrating stiffness and damping. The interest of the rigid-body model is the simplification of a multi-cylinder engine into a virtual single-cylinder engine that has multi-firing within one engine cycle.

This model is easier to inversely solve the dynamics to estimate indicated torque and in-cylinder pressure from the crankshaft instantaneous speed signal. Figure (1) shows a dynamic model of a 4-cylinder engine with 7 degrees of freedom. The moment of inertia of the crankshaft has been located and subdivided into 7 parts identified from J_0 to J_6. Each of these parts has its variables, including position and angular velocity denoted respectively $\theta_{0\ldots6}$ and $\dot{\theta}_{0\ldots6}$. These variables facilitate the analysis of crankshaft deformation.

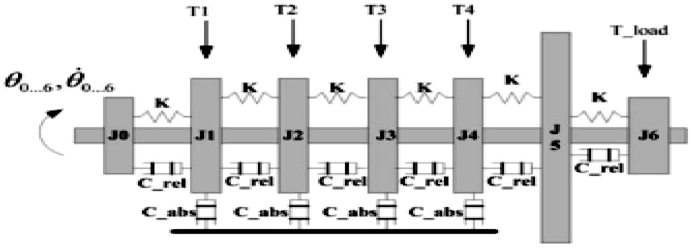

Figure 1. The 7-Degrees of Freedom lumped-mass crankshaft dynamic model of a 4 cylinders in-line engine [9]

The total inertia at the free end of the vibration damper and other auxiliary components is called J_0. J_1 to J_4 represent the inertia of the individual crank that corresponds to each cylinder. Moreover, it includes the rotating part of the connecting rod, which is the largest part. J_5 refers to the inertia of the flywheel, while J_6 refers to the inertia of the drive. The rotating stiffness between the two adjacent inertia of the crankshaft is represented by K. C_{abs} refers to the absolute friction between an inertia and a non-rotating reference point, such as a motor body, while C_{rel} refers to the relative friction between two adjacent inertias. In their turn, C_1 to C_4 represent the torque stimulations of each cylinder operating on the crankshaft. Finally, C_{load} refers to the charging torque that opposes the cylinder torques.

3 Total loss torque

The friction torque term includes piston assembly friction, bearing friction torque, valve train friction torque, pumping losses torque and pumps losses torque [10]. The sum of the friction torque and load torque (including auxiliary) is called the total loss torque and constitutes the target to be reached. The equation (1) indicates that the total loss torque evaluation amounts the estimation of the in-cylinder pressure because the total torque ($J\ddot{\theta}$) and the mass torque $T_m(\theta, \dot{\theta}, \ddot{\theta})$ are easy to determine them, where only the geometry, the moving masses and the kinematic of the engine are occurring (see the reference [11]):

$$T_{totalloss}(\theta) = T_f(\theta) + T_l(\theta) = T_i(\theta) - T_m\left(\theta, \dot{\theta}, \ddot{\theta}\right) - \mathrm{J}\ddot{\theta} \quad (1)$$

From measures of engine in-cylinder pressure and angular speed fluctuation, it will be possible to deduct the waveform of the total friction loss at each engine operating condition. In this work, a large engine operating domain is covered by the experimental tests and the obtained results will be analyzed and a modeling approach proposed.

4 Experimental setup

All experiments were performed on a 4-cylinder turbocharged diesel engine. The figure (2) shows a view of the test cell. Table (1) gives the engine specifications. An eddy current dynamometer is used for loading the engine. An AVL 365C angle encoder, which is part of the experimental device, was connected to the free end of the crankshaft. It is used to convert the angle of the crankshaft into an electrical and then digital signal, evaluated by a data processing system that can determine the position and speed of the crankshaft. The pulse pick-up is rigidly mounted to the motor housing. This high-precision optical-electronic encoder allows angular indication measurements and rotational oscillation analyzes in internal combustion engines.

The encoder sampling step is 0.1 crankshaft degree or 7200 points per cycle. Two systems are installed to manage the control and acquisition of measured signals. The first one controls the engine-dynamometer and achieve the acquisition of low-frequency measurements (torque, average engine speed, manifolds pressure and temperature, air and fuel flowrates). The second system measures high-frequency signal, which mainly concern the crankshaft speed fluctuations, the in-cylinder pressure, and fuel injection pressure.

Figure 2. View of the engine test bench

Table 1. Engine specifications

Parameters	Specifications
Engine type	K9K
Nb of cylinders	Four cylinder
Nb of strokes	Four-stroke
Max. Power	78 kW @ 4000 rev/min
Max. Torque	240 Nm @ 2000 rev/min
Bore x Stroke (mm)	76 x 80.5
Total displacement (cc)	1461
Compression ratio	15.9:1
Air measurement device	AVL Flowsonic
Fuel measurement device	Electronic balance
Model	AVL Dynoperform 80
Eddy current dynamometer	

5 Total loss torque prediction method

The first step is to choose a method based on a mathematical model of the engine or a method based on a database. Measurements under different engine operating conditions must be made, both to build a model and to validate the chosen method. Therefore, we begin to display the characteristics of the engine in a spatial coordinate determined by the simplest measured parameters α (position of the accelerator pedal) and N (average engine speed), as schematized in Figure (3). The parameter α is proportional to the amount of fuel consumed and assumes the included values between 0, the free position of the accelerator pedal and 1, the total fuel consumption by mass. For engine operating conditions determined by coordinates (α, N), the total loss torque can be derived from equation (1) calculated on the basis of cylinder pressure and angular velocity measurements.

Thus, the task is to predict the total loss torque, without relying on the pressure in the cylinder. To do this, it is necessary to build a mathematical model that compares the total loss torque with the operating conditions of the engine, determined by the coordinates (α, N).Thus, it appears from the experimental data on the total loss torque that for a given position of the accelerator pedal α the variation of the total loss torque as a function of the average mode N of the engine can be described using the following linear differential equation:

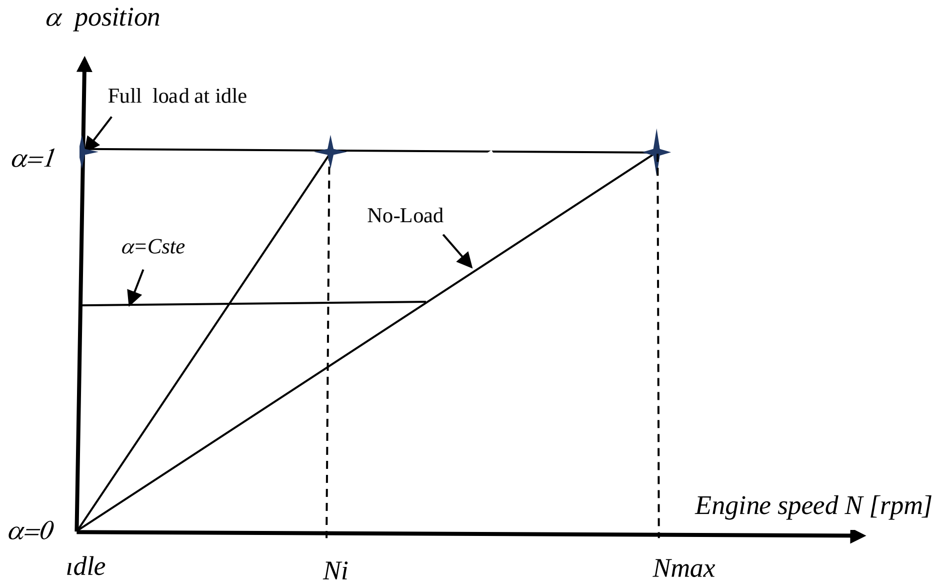

Figure 3. Engine cartography

$$\frac{dT^{\alpha}_{Totalloss}(N)}{dN} = \lambda(\alpha).T^{\alpha}_{Totalloss}(N); \lambda(\alpha) \in R^{n}; 0 \le \alpha \le 1 \tag{2}$$

Where $T^{\alpha}_{Total\ loss}(N)$ is the instantaneous total loss torque value for a given average engine speed N in the fixed position accelerator pedal. The n-dimensional scalar λ(α) is called the «constant of disintegration». The dimension of this scalar depends on the sampling frequency of the angular tone of the crankshaft. For each angular position α, the integration of equation (2) gives a solution:

$$T^{\alpha}_{total\,\mathrm{loss}}(N) = k(\alpha)\cdot T^{\alpha}_{total\,\mathrm{loss}}(N_0)e^{\lambda(\alpha)\cdot N},\ k \in R \tag{3}$$

Where $T^{\alpha}_{Total\ loss}(N_0)$ - instantaneous total loss torque value for the reference speed of the engine $N = N_0$ at the fixed position α accelerator pedal. According to figure (3), the selection of N_0 at the idle regime seems to be interesting. Therefore, from the measured P_{cyl} and θ, we identify and store a database $T^{\alpha}_{Total\ loss}(N_0)$ in the whole variation range of the gas pedal position α.Under intermediate conditions, with a good approximation, the total loss torque $T^{\alpha}_{Total\ loss}(N_0)$ can be estimated by the parameter α using equation (4), [12]:

$$\hat{T}^{\alpha}_{Total\ Loss}(N_0) = T^{\alpha\max}_{Total\ loss}(N_0)\cdot\alpha + T^{\alpha\min}_{Total\ loss}(N_0)\cdot(1-\alpha) \tag{4}$$

Where $\hat{T}^{\alpha}_{Total\ loss}(N_0)$- estimated instantaneous total loss torque at the given position of the accelerator pedal α, $T^{\alpha_{max}}_{Total\ loss}(N_0)$ and $T^{\alpha_{min}}_{Total\ loss}(N_0)$ - respectively the instantaneous total loss torque at maximum ($\alpha=1$) and minimum ($\alpha=0$) values of the gas pedal position α.

The n-dimensional scalar $k(\alpha)$ and $\lambda(\alpha)$ in the equation (3) are also identified a priori for the whole variation range of the parameter α using experimental data (P_{cyl} and θ) for $T_{Total\ loss}$ calculation. In intermediate conditions, a linear interpolation based on the Taylor-Young first order formula is used, equation (5):

$$\hat{P}_{\alpha} = \frac{P_{\alpha(i-1)} - P_{\alpha(i+1)}}{\Delta\alpha}\cdot\alpha + \frac{\alpha(i-1)\cdot P_{\alpha(i+1)} - \alpha(i+1)\cdot P_{\alpha(i-1)}}{\Delta\alpha} \tag{5}$$

Where $\hat{P}_{\alpha}$ represent the estimated parameter *k(α)* or *λ(α)* at a given position *α* of the gas pedal, $P_{\alpha(i-1)}$ and $P_{\alpha(i+1)}$ the corresponding values at the limits of the interval $\Delta\alpha$.

The Figure (4) shows a series of total loss torque obtained for the gas pedal position *α*=0,45. From this Figure we notice that the evolution of the total lost torque is proportional to the rotational speed. For this sample, the average $T_{Total\ loss}(\theta)$ curve, figure (5), matches with that of 1500 rpm corresponding to the mean engine speed value of the considered range for this sample. The same situation is systematically found for all other samples. This remark confirms the strong linear correlation between $T_{total\ loss}(\theta)$ and the average engine speed at a fixed *α* position.

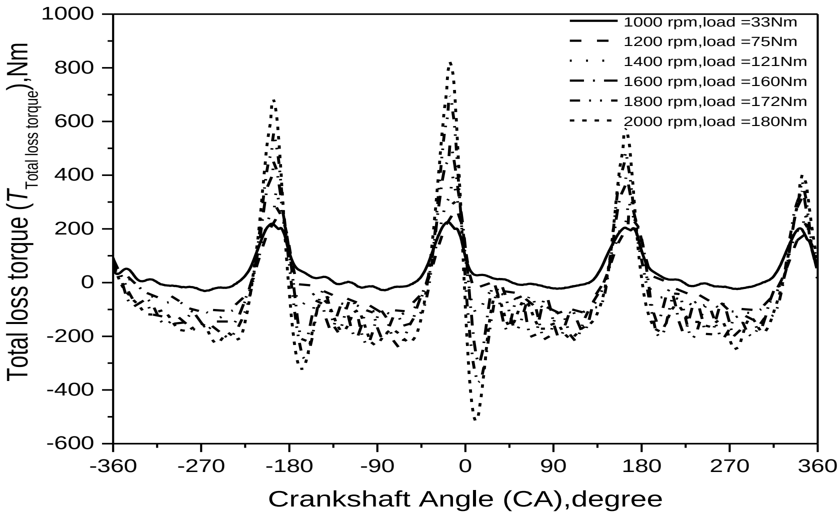

Figure 4. Set of total loss torque for one cylinder at α= 0,45

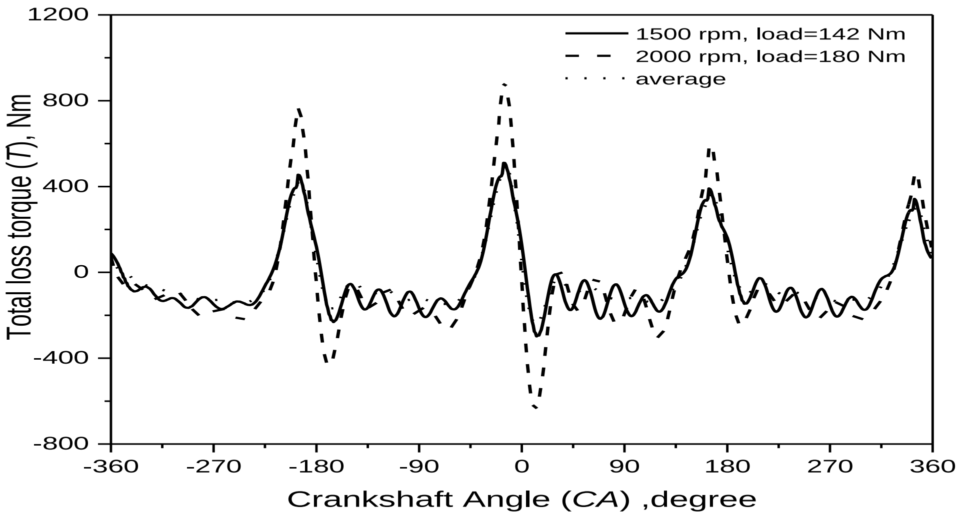

Figure 5. Average total loss torque for one cylinder

Hence, the equation (3) is resolved for each angular step position □ during the cycle to estimate the total loss torque evolution. Consequently, knowing the average engine speed and the angular speed fluctuations it is possible to compute the total loss torque leading to the indicated torque and using methods described in the references [12], the reconstruction of the in-cylinder pressure P_{cyl} will be possible. The computation algorithm is showed on the Figure (6).

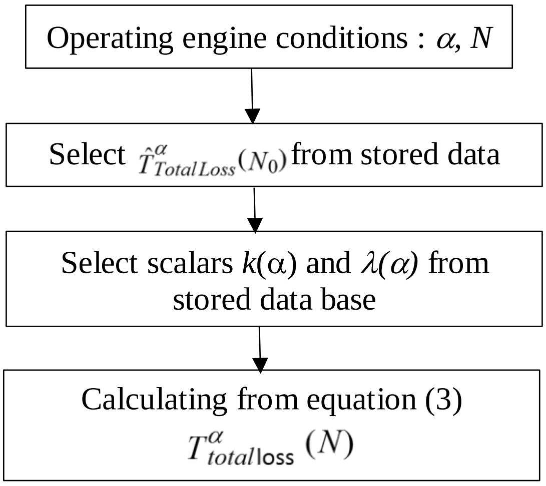

Figure 6. Computational algorithm for total loss torque prediction

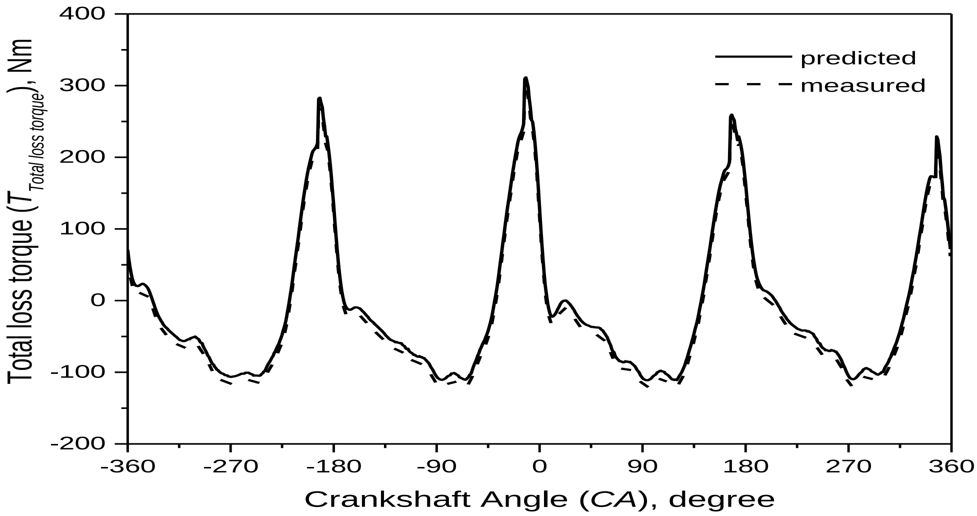

Figure 7. Predicted and measured total loss torque. Engine operating conditions N=1100 rpm; Load=67 Nm

6 Results

Figures (7) to (10) are selected arbitrarily to show some results. The figures (7) and (8) illustrate the comparison between estimated and experimental total loss torque at two different engine operating conditions. Figures (9) and (10) show the estimated indicated torque T_i (gas torque) obtained from the total loss torque prediction and that obtained via the equation (5) using the measured variables P_{cyl} and θ. There is a good agreement between the predicted and measured curves, which indicate the robustness of the proposed model for the total loss torque prediction. The quadratic deviation varies between 5% at low engine speed and 11% at high engine speed. This model can be used for online applications due to it very short computation time.

Also, the prediction of the indicated torque during transient conditions is in progress with this model. This constitutes our next publication work.

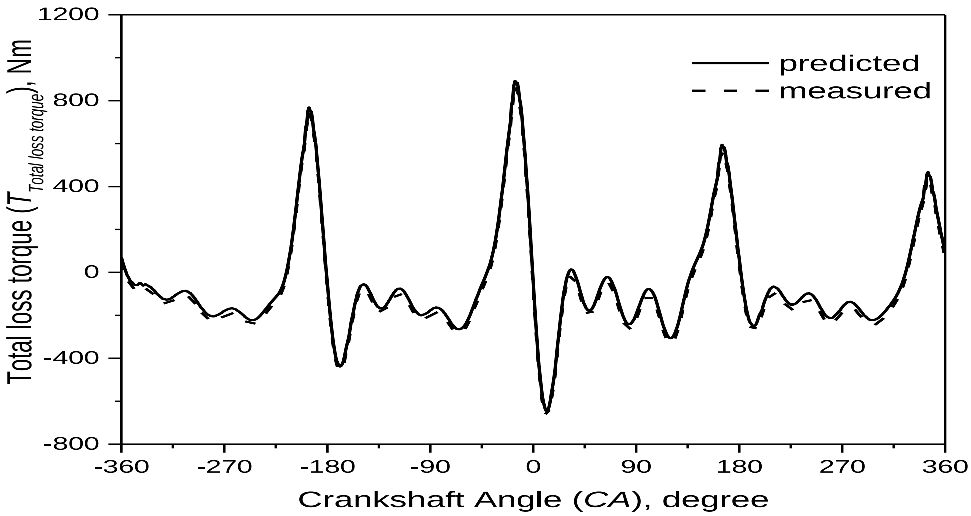

Figure 8. Predicted and measured total loss torque. Engine operating conditions N=2000 rpm; Load=180 Nm

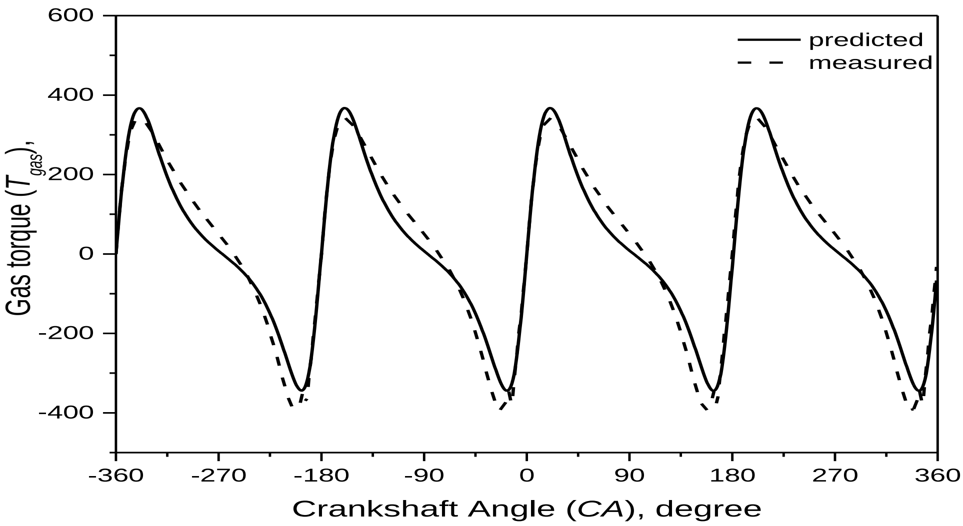

Figure 9. Comparison between estimated and measured gas torque. Engine operating conditions N=1300 rpm; Load=96 Nm

7 Conclusion

The prediction of the waveform of the total loss torque is a key step in the prediction of the gas torque. The objective of this study is to evaluate an innovative method for estimating the total loss torque. The proposed method is based on a simple and practical approach capable to be implemented online for the diagnosis or control of the internal combustion engine. The online measured parameters are the position of the gas pedal, the average engine rotational speed and the instantaneous angular speed of the crankshaft. The application was conducted on a turbocharged 4-stroke 4-cylinder inline engine on which the engine crankshaft dynamic model is based.

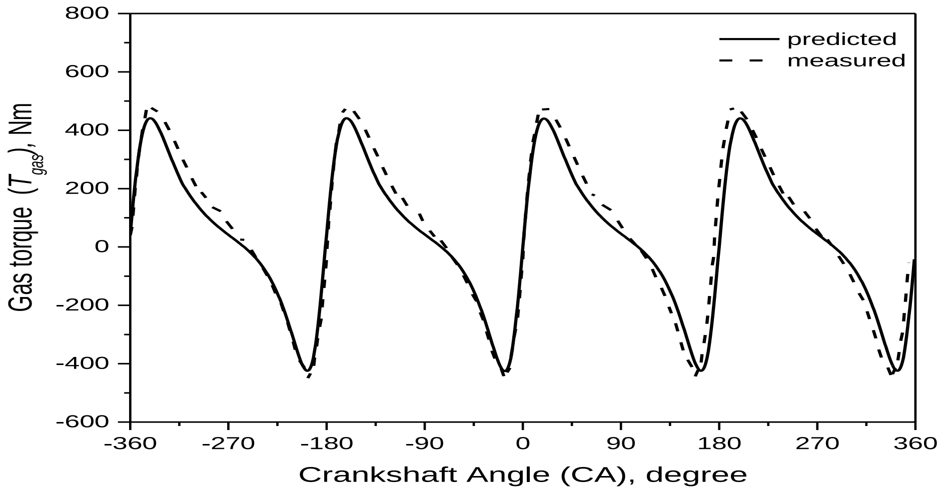

Figure 10. Comparison between estimated and measured gas torque. Engine operating conditions N=1600 rpm; Load=160 Nm

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