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1D MODELLING OF ENGINE CRANKSHAFT: IMPACT OF IGNITION TIMING ON CRANKSHAFT TORSIONAL VIBRATIONS

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Internal combustion engines by nature do not generate a constant torque. The torque varies within an engine cycle as a consequence of the non-uniform combustion and inertia forces, leading to a varying engine rotational speed, named torsional vibrations. Torsional vibration can negatively impact the engine performance, and potentially leading to fatigue failure of the crankshaft, or to important NVH problems.
The demand for more fuel efficient powertrains and reduction of CO_2 emissions has been leading to a general engine downsizing trend. Technologies like charged direct injection combined with advanced ECU strategies allow reducing the number of cylinders without compromising vehicle performance. However all these technologies result in significantly increased engine torsional excitations from the firing order, making the driveline NVH integration more challenging. A thorough understanding of the impact of the engine control parameters on the NVH performance represents an important step in the engine development phase.
This paper presents an example of how different ECU (Engine Control Unit) parameters influence crankshaft torsional vibrations by means of 1D simulation. Different ignition timing settings have been simulated. Finally the simulated torsional vibration results have been compared with experimental data acquired on a Ford 1.6L Ecoboost engine. The good correlation between the simulation and measured data proves the reliability of 1D modelling for this application case, and highlights the importance of combine simulation models and advanced testing methodologies during the engine development phase.
Keywords: engine crankshaft modelling, engine control unit, torsional vibrations

1. Introduction

The more stringent regulations in terms of CO_2 reduction have created new challenges to the automotive OEMs. Regulations are transforming the powertrain development by pushing the industry to innovate and find the most cost effective solutions to meet future global emissions and fuel economy standards. As an example, the European Commission has set mandatory emission reduction targets for new cars. By 2021, the fleet average emission is 95 grams of CO_2 per kilometre. This translates to an averaged fuel consumption of 4.1 l/100km for petrol cars and 3.6 l/100km for diesel cars [1].

The key-technologies commonly adopted by the car manufacturers to meet those regulations consist mainly of downsizing the engine, adding technology devices such as turbochargers, reducing vehicle mass and improving the vehicle aerodynamics. These technologies proved to be efficient [2]. However, one of the most important consequences of the ongoing powertrain downsizing trend is the increased torsional excitation of the vehicle driveline. The reduced number of cylinders and the presence of turbochargers further increase the overall torque level and torque ripple responsible for torsional vibrations. In addition new powertrain technologies such as start-stop systems, cylinder deactivation, advanced torque lock-up strategies, direct injection and variable ignition timing strongly affect driveline torsional vibrations, efficiency and comfort.

Torsional vibrations, also known as speed fluctuations, represent an important aspect in the NVH development of new vehicles. Torsional vibrations can excite resonances in the engine crankshaft, valve train, transmission and driveline, and lead to specific noise and vibration problems such as gear rattle and booming noise, and to fatigue failure.

Torsional vibrations can be affected by the physical geometry of the mechanical engine components, or by the combustion process, which generates the force acting on the pistons head. This paper presents the impact of different ignition timing on the crankshaft torsional vibrations. The paper is organized in the following sections: firstly the 1D simulation model used to simulate crankshaft torsional vibration is presented; afterward the experimental test campaign performed on a real engine test-bench is described. The model has been validated by means of an experimental test campaign performed on an engine test-bench. During the experimental tests, the ignition timing was adjusted, within the knocking limits, by means of a fully programmable ECU (Engine Control Unit). Finally, the correlation between the simulation and the experimental results will be presented.

2. 1D simulation model

LMS Imagine.Lab Amesim is a 1D lumped parameter time domain simulation platform that enables to model, simulate and analyze multi-domain systems and offers plant modelling capabilities to connect to controls design helping you to assess and validate control strategies.

The simulation model can be used in numerous ways, from target settings to system optimization. More specifically, by using a powertrain simulation model, one can get an in-depth understanding of the NVH performance of the powertrain system and optimize mechanical parts as well as the overall architecture.

In order to simulate the crankshaft torsional vibrations, a 1D Amesim model of crankshaft has been used, starting from the components existing in the Powertrain library [3].

The crankshaft model is based on the assumption that the absolute pressure applied on the piston is the source of the torque. Thus, the pressure resulting torque is calculated as a function of the crankshaft angular position and the geometrical characteristics of the system which includes piston, connection rod length and crank throw radius. A schematic representation of the engine kinematic model is presented in Fig. 1.

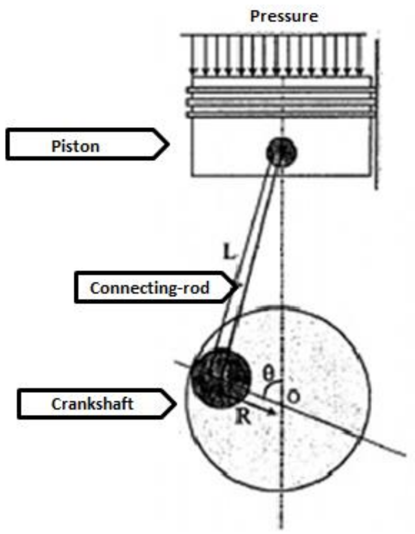

Figure 1: Schematic representation of the engine kinematic

The pressure force acting on the cylinder is obtained by Eq. (1):

$$F_p = A \cdot (P - P_{atm}) \tag{1}$$

Where A represents the piston area. The resulting torque is then:

$$T_p = G \cdot F_p \tag{2}$$

Where:

$$G = R \cdot sin(\theta) + \frac{\frac{R^2}{L} \cdot \sin(\theta) \cdot \cos(\theta)}{\sqrt{1 - \frac{R^2}{L^2} \cdot \sin(\theta)}} \tag{3}$$

In which, L is the connecting rod length, R is half of the stroke and θ is the crank angle.

By connecting several models to each other by mean of intermediate stiffnesses, the dynamic behaviour of a complete engine is modelled. The crankshaft element model includes also the dynamics of the rotating masses, taking into account the effect of the piston mass, the connecting-rod mass and inertia, and the crankshaft inertia.

The final model for a 4 cylinders in line engine used to simulate the crankshaft torsional vibrations is presented in Fig. 2.

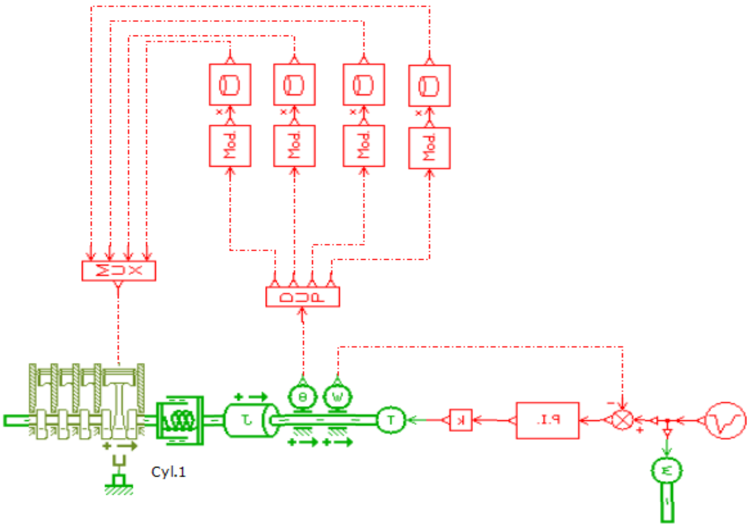

Figure 2: 1D Amesim model used to simulate crankshaft torsional vibrations

Components mass was physically measured by weighing the engine components. The inertia of the rotating components (i.e. crankshaft and connecting rod) was derived from 3D models generated in LMS Virtual.Lab.

In-cylinders pressure evolution in function of the crank angle, for one complete combustion cycle, represents the input for the Amesim model. In this application case we assumed that all the cylinders have the same pressure evolution, and in addition the pressure in the cylinder does not vary from one cycle to the other. The in-cylinder pressure evolution can be either obtained from a model or experimentally measured. In the results analysis presented at the end of this paper, the in-cylinders pressure was experimentally acquired on a real engine with LMS Scadas Mobile and LMS Test.Lab Signature Testing.

The output of the model is the time evolution of the crankshaft speed, which includes the torsional fluctuations.

The advantage of having a lumped parameter model allows to perform sensitivity analysis on engine parameters assessing their effect on speed fluctuations of the crankshaft. In addition to NVH, the analysis could be extended to evaluate CO_2 emissions, fuel consumption and other engine attributes.

In this way, in a first approximation, the simulation allows to assess and verify the engine behaviour without the need of a full-scale engine prototype. Moreover, the use of a model allows to perform multi-attribute analysis, such as the investigation of the best trade-off among different engine control parameter settings which lead to improved NVH performance while keeping fuel consumption and emissions low.

3. Experimental test campaign

An experimental test campaign on a Ford 1.6L Ecoboost has been performed in the test facility from the Department of Flow, Heat and Combustion Mechanics of Gent Universiteit. The engine was mounted on a test-bench. The objective of the test campaign was to validate the simulation model.

As the name Ecoboost suggests, the engine has been designed from Ford to lower the fuel consumption and emissions. The engine is equipped with some technologies aiming to reduce consumption and increase the efficiency, such as: turbocharger, direct injection and independent variable valve timing for further optimization. Engine details are presented in Table 1.

Table 1: Specifications of the Ford Ecoboost 1.6L engine

Displacement	1596 cc
Bore	79.0 mm
Stroke	81.4 mm
Compression ratio	10.0:1

The engine was instrumented with: a piezoresistive pressure sensor measuring the in-cylinder pressure evolution; an incremental encoder with 360ppr instrumented on the crankshaft [4]; and a torque sensor on the brake. In addition the intake manifold was instrumented with a pressure sensor; and thermocouples were instrumented at the intake and exhaust manifold.

The engine was equipped with a fully controllable ECU from Motec which allowed to freely adjust the engine control parameters and to read them through CAN bus.

The NVH signals (e.g. in-cylinder pressure, crankshaft speed, mount vibrations,…), together with the data streamed from the ECU, were synchronously acquired using the LMS SCADAS Mobile. In addition, fuel consumption was measured using a dedicated gravimetric fuel consumption system.

In this test campaign the impact of engine ignition timing on torsional vibrations of the crankshaft was evaluated. Tests were performed using four different ignition timings at stationary speed of 2000rpm and constant throttle of 50%. The ignition timing values used during the tests were: 0º, 5º, 8º, 12º. These values represent the degrees of anticipation of the ignition before the top dead centre (TDC). It is also commonly referred as spark advance (SA) and is defined in degrees.

When setting the ignition timing values, great care was taken avoiding the knocking limit, as this could seriously damage the engine. In case the ignition timing is too much anticipated, the combustion process starts while the piston is moving towards the TDC, so the compression work (negative work) increases. On the other hand, if the spark timing is too much delayed, the combustion process is delayed and the peak of pressure occurs much later in the expansion stroke [5],[6].

As can be noticed from Fig. 3, the different ignition timing settings have a clear impact on the measured in-cylinder pressure.

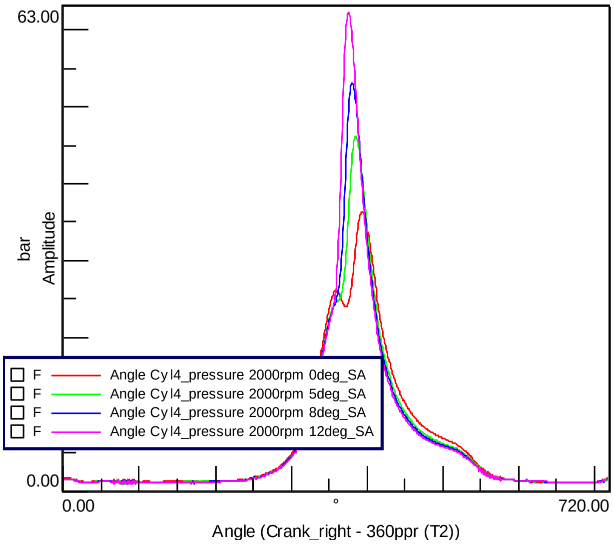

Figure 3: Measured in-cylinder pressure in function of the crank angle for different ignition timing

The peak of the measured in-cylinder pressure increases while increasing the spark advance. In addition, by anticipating the ignition timing, the engine provided a reduction in brake specific fuel consumption (BSFC) from 256.71 g/kWh to 228.08 g/kWh, and, at the same time, a power increase from 32 kW to 32.62 kW, respectively for the tests with 0º and 12º of ignition timing.

4. Impact on crankshaft torsional vibrations

Crankshaft speed fluctuations results obtained from the 1D Amesim model and one the ones experimentally measured on the real engine have been compared. Figure 4 shows the results for the 2nd and 4th crankshaft speed fluctuations in function of the ignition timing. By design, those are the most dominant orders for an in-line 4 cylinders engine.

The torsional vibration orders obtained from the 1D model show a good correlation with the ones experimentally measured on the real engine. However, a small consistent underestimation of the simulation results can be observed. This difference might be related to the model assumption. The in-cylinder pressure variability among the different cylinders has not been considered in this analysis. Only one cylinder pressure was measured.

Another interesting aspect that can be derived from Fig. 4 is the effect of the ignition timing on the orders. While the 2nd order seems to be not affected by the different ignition timing settings, the 4th order shows a clear increase in amplitude with the increase of advancing of the ignition. Even though the level of the 4th order is lower compared to the 2nd order, the results of the analysis proved that the ignition timing is an important variable to be considered in the preliminary engine NVH optimization phase.

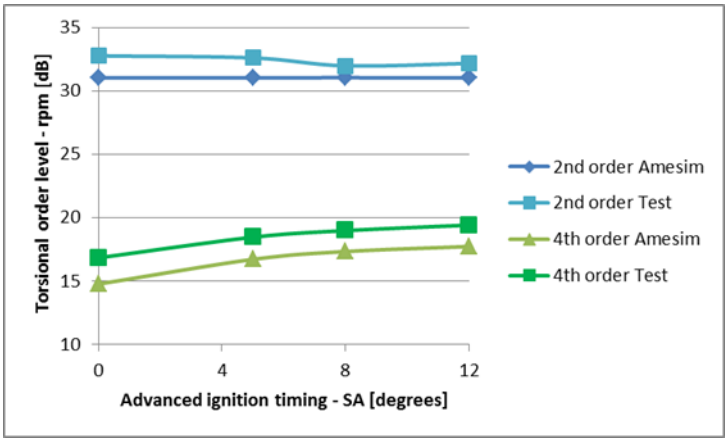

Figure 4: Comparison of simulated and measured results for the 2nd and 4th speed fluctuations order of the crankshaft in function of different ignition timings

5. Conclusions

Internal combustion engines by nature do not generate a constant torque. The torque varies within an engine cycle as a consequence of the combustion process and the inertia forces, leading to a varying engine rotational speed, named torsional vibrations.

The paper presents an application case in which crankshaft torsional vibrations are affected by different ECU settings. Specifically the impact of different ignition timings has been investigated in this study.

A 1D simulation model in Amesim has been used to simulate the crankshaft speed fluctuations. In addition experimental tests on a Ford Ecoboost 1.6L have been carried out in order to validate the model.

The results obtained in this study demonstrated that the ignition timing is an important design variable to be tuned for effective engine power assessment and NVH performance. Optimum spark timing must be obtained for which the maximum brake torque is obtained for a minimum fuel consumption and minimum NVH.

On the other hands, the good correlation between the simulation and measured data proved the possibility of using 1D simulation models, during preliminary engine development phase, for crankshaft NVH prediction, without the need of a full-scale engine prototype. However, the user should be aware of the frequency limitation of a 1D model.

Further analysis will be performed on the simulation model including in-cylinder pressure variability and extending the study to mount vibrations and other engine attributes.

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