

Cylinder Balancing Based on Reconstructed Engine Torque for Vehicles Fitted with a Dual Mass Flywheel (DMF)

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ABSTRACT

The integration of a Dual Mass Flywheel (DMF) in the conventional vehicle driveline leads to various benefits, and hence today it has established its position in many passenger cars and light trucks. Transmission and driveline oscillations are reduced by mechanically decoupling the transmission from the periodic combustion events that excite the engine crankshaft, improving driving comfort and reducing transmission stresses. For systems with conventional single mass flywheel (SMF) reliable engine control systems have already been developed. However, the complexity of the driveline increases with the integration of a DMF. Hence, in the future conventional engine control systems may require adaptation, modification or even replacement, in order to guarantee the optimal control of engines equipped with advanced DMF systems. Over time, various methods have been proposed for cylinder balancing using a wide range of different and often expensive sensing techniques e.g. using acceleration transducers or by measuring the in-cylinder or exhaust manifold pressures directly. However, cost, reliability and packaging constraints mean that until now signals derived from the engine's existing crankshaft position sensor, i.e. the angular velocity or acceleration of the crankshaft, have generally been used to perform the important task of cylinder balancing. This approach works well for engines with a limited number of cylinders connected to relatively simple drivelines. Nevertheless in some circumstances torque reactions from the DMF or the rest of the vehicle driveline can be misinterpreted as cylinder imbalance; even when all cylinders are perfectly balanced. In this paper, a novel cylinder balancing method, for both diesel and gasoline engines, is introduced which uses a dynamically reconstructed continuous engine torque signal. A state space model of the DMF is used to estimate the engine torque. Cylinder imbalance can be detected and corrected far

more rapidly and reliably when the reconstructed instantaneous engine torque is used instead of commonly used signals simply derived from the crankshaft velocity or acceleration. The reconstructed engine torque can also be used to provide efficient and robust solutions for various other engine control and diagnosis purposes, e.g. OBD-misfire detection, thus eliminating the need to install multiple additional sensors in the vehicle. These aspects are particularly important for low to mid-cost vehicles.

INTRODUCTION

Significant cylinder-to-cylinder torque imbalances are introduced when the injected fuel quantity (diesel or direct injection gasoline engines) or the air-fuel mixture quantity (conventional port injection gasoline engines) differs. These differences can have various causes. Fuel injectors show variabilities due to manufacturing tolerances, aging (e.g. soot deposits build-up) or differences in the electrical activation or injector opening time. These factors as well as pressure variations in the fuel supply and the air intake systems effect the quantity of combustible fuel and air in each cylinder. Furthermore, differences in the geometry of the combustion chamber, due to manufacturing tolerances and wear (e.g. piston/cylinder height and radius or differences in the piston ring sealing efficiency) result in different effective air charge compression, combustion profiles and cylinder dynamics which also effect cylinder-to-cylinder torque variations. Cylinder imbalance has undesirable effects such as driveline oscillations that perceptibly influence the driving comfort and can in extreme cases reduce the longevity of the powertrain. Interactions with the engine idle speed controller can also occur which can further worsen the situation. Exhaust gas emissions and fuel consumption are both increased with unbalanced cylinders. To avoid the various negative effects described above, speed fluctuations resulting from

unbalanced cylinder torques have to be reliably detected and quickly corrected by the engine ECU (Electronic Control Unit).

Over the years various systems using different types of sensor have been proposed for cylinder balancing. Measuring fluctuations of the crankshaft, i.e. the engines angular velocity, is one well established method [1]. No additional sensors are required; only the existing crankshaft position sensor is used. Generally such methods are employed on engines with a lower number of cylinders i.e. 3, 4, 5 and 6-cylinder engines. On such engines, cylinder imbalance leads to significant oscillations in the rotational speed or acceleration of the crankshaft, which can often be reliably measured. Unfortunately, disturbances acting on the crankshaft (e.g. driveline torque reactions) can lead to erroneous cylinder imbalance detection and correction. This is the principal disadvantage of this method. Another method relies on relatively expensive pressure sensors installed within each combustion chamber. Here the cycle-average cylinder torque is calculated from the individual in-cylinder pressures and the crankshaft position. If all in-cylinder pressure sensors are correctly calibrated, then individual average in-cylinder torques can be compared to detect and correct imbalances. Exhaust manifold pressure sensors can also be installed to obtain a measure of imbalance by integrating the exhaust manifold pressure over one combustion event [2]. In-cylinder pressure sensors are, due to their low cross-sensitivity, expensive and predominantly used in luxury vehicles with 6 or more cylinders. Acceleration transducers have also been used to detect cylinder torque variations [3].

In order to provide an acceptable solution for low and mid-cost vehicles, cylinder balancing systems have to fulfil demanding robustness requirements at very low cost. As previously noted the crankshaft speed sensor method is very sensitive with respect to driveline torque reactions. This combined with the now widespread utilisation of the Dual Mass Flywheel (DMF), shown in Figure 1, has provoked a search for more robust and low cost methods of detecting and correcting cylinder torque imbalances.

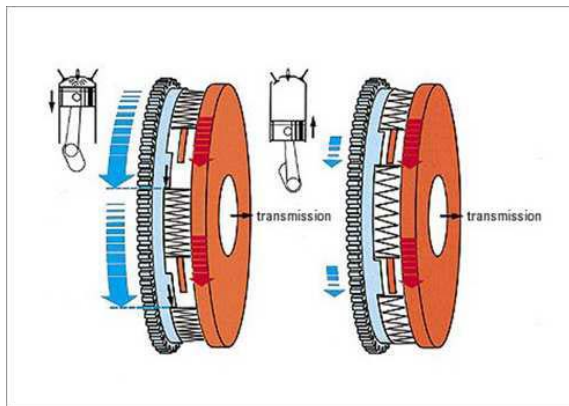


Figure 1: Functional structure of a dual mass flywheel

Basically, the DMF includes two large rotating flywheel inertias connected by long travel arc-springs. More detailed information about the DMF can be found in [4]. One of the principal advantages of the DMF is the almost complete elimination of secondary flywheel side and thus gearbox input-shaft speed oscillations. Annoying gear rattle as well as other noises and vibrations are eliminated, allowing the engine to be driven at lower speeds, improving real world fuel economy. Today the DMF is an inherent part of the vehicle driveline, due to its excellent vibration isolation and damping attributes; especially for diesel engines, with very high peak combustion torques, as well as turbo- or super-charged gasoline engines. Even in the low-cost passenger car segment has the DMF established its position.

However, despite all of these advantages, the DMF increases the driveline system complexity. Analytically, the DMF is a spring-damper element with strongly non-linear behaviour. Due to its capability of temporary energy storage, it produces a reaction torque on the combustion engine and thus affects the crankshaft speed signal. Cylinder balancing systems based on angular velocity or acceleration of the crankshaft may misinterpret these torque reactions from the DMF, which can in extreme cases effect the robustness and efficiency (i.e. time taken to eliminate torque imbalances) of conventional cylinder balancing systems. In this paper, a new cylinder balancing algorithm based on the reconstructed instantaneous engine torque signal is introduced.

RECONSTRUCTION OF INSTANTANEOUS ENGINE TORQUE

An expensive existing method of determining the instantaneous engine torque is to measure the pressure within each combustion chamber. However, in this paper, an accurate estimation of the instantaneous engine torque is achieved by using the DMF as a 'virtual torque sensor'; a reasonable, robust and cost effective alternative. Furthermore, the use of the DMF as a virtual torque sensor offers the advantage that cylinder-to-cylinder torque variations due to mechanical friction differences are also included and compensated for; this is not the case when in-cylinder pressure sensors are used.

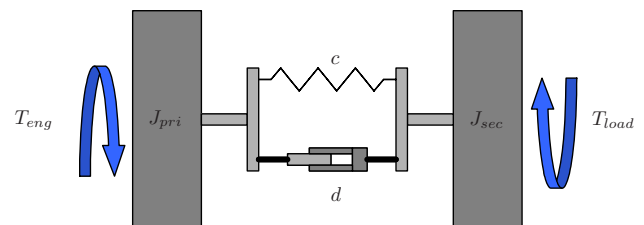


Figure 2: Simple spring-damper element DMF model

Note that in-cylinder pressure sensors also suffer from the disadvantage that they must be recalibrated over the vehicle life to ensure that they always output comparable sig-

nals. In order to estimate the engine torque, a state space model of the DMF has been developed, which is reviewed here and described in more detail in [5] and [6]. The basis for the real-time DMF model is the simple linear spring-damper element shown in Figure 2.

According to Newton's second axiom of rotational motion the following equations describe this simple linear system:

$$J_{pri} \cdot \ddot{\alpha}_{pri} = T_{eng} - c \cdot (\alpha_{pri} - \alpha_{sec}) - d \cdot (\dot{\alpha}_{pri} - \dot{\alpha}_{sec}) \quad (1)$$

$$J_{sec} \cdot \ddot{\alpha}_{sec} = -T_{load} + c \cdot (\alpha_{pri} - \alpha_{sec}) + d \cdot (\dot{\alpha}_{pri} - \dot{\alpha}_{sec}) \quad (2)$$

Here α_{pri} and α_{sec} are the absolute angles of inertias J_{pri} and J_{sec} respectively. $\dot{\alpha}_{pri} = \omega_{pri}$ and $\dot{\alpha}_{sec} = \omega_{sec}$ are the associated angular velocities and $\ddot{\alpha}_{pri}$ and $\ddot{\alpha}_{sec}$ the angular accelerations.

In order to derive a linear state space model, the states of the system have been defined to be:

$$\underline{x} = \begin{pmatrix} \alpha_{pri} - \alpha_{sec} \\ \omega_{pri} \\ \omega_{sec} \end{pmatrix} \quad (3)$$

Therefore, the state space representation is:

$$\dot{\underline{x}} = \begin{pmatrix} 0 & 1 & -1 \\ -\frac{c}{J_{pri}} & -\frac{d}{J_{pri}} & \frac{d}{J_{pri}} \\ \frac{c}{J_{sec}} & \frac{d}{J_{sec}} & -\frac{d}{J_{sec}} \end{pmatrix} \cdot \underline{x} + \begin{pmatrix} 0 & 0 \\ \frac{1}{J_{pri}} & 0 \\ 0 & -\frac{1}{J_{sec}} \end{pmatrix} \cdot \underline{u} \quad (4)$$

The input vector $\underline{u}$ includes the engine torque T_{eng} as well as the driveline load torque T_{load} . The output vector contains the primary and secondary flywheel speeds.

$$\underline{y} = \begin{pmatrix} 0 & 1 & 0 \\ 0 & 0 & 1 \end{pmatrix} \cdot \underline{x} = \begin{pmatrix} \omega_{pri} \\ \omega_{sec} \end{pmatrix} \quad (5)$$

The parameters J_{prim} , J_{sec} , c and d are determined using an identification algorithm. The model in Figure 2 is an extreme simplification of the real DMF. Therefore, the parameters in this simplified model are virtual, which means that these values cannot be directly imported from engineering drawings or data sheets.

Generally, identification describes a process of adapting system model parameters to accurately describe the real process by minimising a pre-defined cost function. The input and output values of the real plant are the same as those of the system model. In this case, the input values are the engine torque T_{eng} and the driveline torque T_{load} . The output parameters are the rotational velocities of the inertias ω_{pri} and ω_{sec} , where ω_{pri} is the instantaneous engine speed. Both, input and output values will be recorded in one dataset, called the identification dataset. The data can be recorded by measurement during a test vehicle drive cycle; note this requires accurate measurement of the torque information using additional test sensors. Alternatively, data from the real plant can be obtained from

a dynamic torsional test rig or an accurate and well validated simulation model. The identification of the system parameters is determined at once.

To compare the measured data with the model during the identification process, a cost function has to be defined. Here a quadratic cost function Q has been used to calculate the standard output error of a parametric system.

$$Q(\underline{a}) = \sum_{k=1}^N \sum_{j=1}^{q=2} \underline{e}_j^2(t_k) = \sum_{k=1}^N \sum_{j=1}^{q=2} (y_j(t_k) - \hat{y}_j(t_k))^2 \\ = \sum_{k=1}^N (\underline{y}(t_k) - \hat{\underline{y}}(t_k))^2 \rightarrow \min_{\underline{a}} \quad (6)$$

$\underline{a}$ is the desired parameter vector, which contains J_{pri} , J_{sec} , c and d . $\underline{y}$ contains the data measured on the real plant, whereas $\hat{\underline{y}}$ contains the data output by the identified model. N is the number of recorded data samples and q is the dimension of the output vector, which includes ω_{pri} and ω_{sec} . A quadratic cost function has the advantage, that strong variations of the model will be charged more than small ones.

Results from the identified model, at 850 rpm stationary (i.e. average) engine speed and 50 Nm transmission load torque, are shown in Figure 3. Here, the validation of the engine speed y_1 is shown. The validation dataset¹ is different to that used in the original identification.

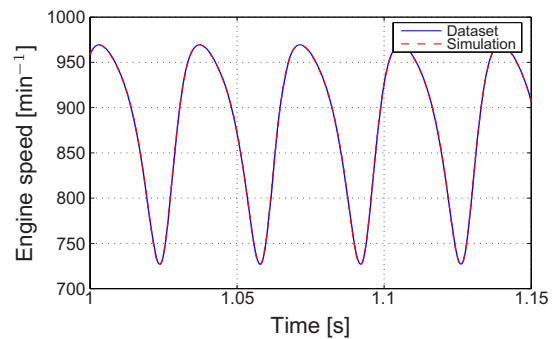


Figure 3: Validation of primary side or engine speed y_1

Results for the secondary flywheel side rotational speed (equivalent to the transmission input-shaft speed with a fully engaged and rigid clutch disc) are shown in Figure 4.

Figure 3 and 4 show, that the simple state space model used approximates the real behaviour of the complex DMF at (or near to) a constant operating point very well. The model has also been identified and successfully tested for other stationary engine speeds and loads. Note good results can be achieved for virtually all stationary engine operating points.

¹the dataset is based on a 4-cylinder 2.0 litre diesel engine