JJMIE

Volume 6, Number 1, Feb. 2012
ISSN 1995-6665
Pages 71 - 74

Jordan Journal of Mechanical and Industrial Engineering

Identification and Analysis of Engine Speed and Noise in In-line Diesel Engine

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Abstract

For the cylinder power imbalancing/balancing methods proposed earlier in this research work by the authors, measurements of the angular speed of the crankshaft with sufficient quality is essential. In this paper different aspects of speed measurement are presented. A method is in addition is outlined for identification of the torque-noise component due to Quantization noise & Geometric errors in the speed measurement setup. As the influence of mass torque and noise due to geometric noise is constant on the reconstructed oscillating torque for small fuel injection adjustments, the resulting change in the oscillating torque can be directly attributed to the fuel injection and gas torque change. The phase-angle diagram of the engine can consequently be identified through parameter estimation.

Keywords: Geometric noise; Speed Measurement; Quantization noise

1. Introduction

Internal combustion engine noise has been drawing significant attention from automotive to power generation manufacturers. To effectively reduce the noise level, the first step is the identification of noise sources, which relies on the noise signal analysis. From the aspect of condition monitoring or manufacture/assembly quality assessment, abnormal noise signals usually indicate abnormal conditions or problems in the manufacture quality. To pick up the problem, the abnormal signal source should be found first. This also depends on the signal analysis. Internal cylinder pressure (or torque) estimation is an important engine parameter with significant implications for diagnostic and control applications in internal combustion engines.

Unfortunately, direct measurements of internal cylinder pressure or torque production are expensive and currently not feasible for field applications. Instead, methods based on direct speed measurements [1–2] or engine block vibrations [5-7] are often employed for internal cylinder pressure or torque estimation. The advantage of crankshaft speed measurement-based algorithms is the fact that they rely on the existing crank and camshaft sensors and require no additional instrumentation. Speed-based cylinder health diagnostics are generally executed in the crank-angle domain, in which the position of the crankshaft replaces time as the independent variable [1]. The main advantage of formulating and executing in the crank-angle domain is the synchronisation of speed sampling with the engine cycle. Loss of critical cylinder health information can also be prevented by avoiding the re-sampling of the speed signal in the time-domain. A great deal of cylinder health information is periodic throughout an engine cycle, thereby limiting the frequencies of interest to integer frequencies in the crank-angle domain. Many speed-based diagnostics methods employ models of the engine dynamics. In [8], the signature analysis of crankshaft speed fluctuations is utilised for detection of misfiring cylinders. In [1, 2], dynamic engine models with flexible crankshafts are used to construct high-gain non-linear observers for indicated pressure or torque. The estimated indicated pressure or torque can be directly used for cylinder health diagnostics as well as controls. In [9], frequency-response function models of the speed signal is used for cylinder diagnostics. In [11], a survey of previously existing diagnostics methods based on Fourier analysis and correlation methods is provided and is followed by three different cylinder health diagnostics methods. A diagnostics method based on the measurement and comparison of the speed fluctuations at the flywheel and at the front end of the engine over an engine cycle is presented in [12].

The vibratory signal measured at any point on the internal combustion engine structure, is composed of a very complex superposition of the contributions of different vibratory sources modified by their respective transmission paths. These sources originate from several internal phenomenons in the engine and excite the natural modes of the engine. The vibration is amplified at the natural frequencies of the engine. Therefore, the produced vibration and the noise radiated from the engine result

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from the combination of the excitations and the dynamic response of the structure. Diesel engine vibration & noise is caused by combustion, mechanical, and the combination of both sources, such as combustion forces, piston slap, inertia forces, injection forces, the distribution system forces due to impacts of the intake and exhaust valves on their seats as well as the excitation of the valve tails by the rocker arms, applied sometimes simultaneously at different phases of the engine cycle.

1.1. Diesel Engine Mechanical Noise Model:

With reference to the above discussion on the composition of the engine noise in section 1, a noise signal can be mathematically described by Eq.1 as;

$$x(t) = \sum a_i \cos(\omega t + a_i) + \sum\sum b_{ij}(t)u(t - t_j) * \cos(\omega_{ij} t + \alpha_{ij}) \qquad (1)$$

where a_i and $b_{ij}(t)$ denote the amplitude of a signal component, ω_i and ω_{ij} represent the frequency, $u(t)$ is the step function, t_j is the instant at which an event occurs, and α_i and α_{ij} are phases of signal components. That is, a noise signal component can be characterized with four quantities: amplitude, frequency content, time, and phase. The first part of this equation is major harmonic components. The signal is dominated by this harmonic components i.e., if the first term of this equation is removed, it can be seen that the signal contains many transient components which are the second term of the equation.

In this work a method for identification of the torque-noise component due to Quantization noise & Geometric errors in the engine speed measurement setup is outlined, as the measurements of the angular speed of the crankshaft with sufficient quality is essential for engine health diagnostics.

2. Experimentation and Measurement

2.1. Measurement & Calculation of Angular Speed & Noise of Multi-cylinder Diesel Engine:

Crankshaft angular speed of internal combustion engines is usually measured by means of a toothed flywheel or measurement disk and an inductive speed-pickup. As the teeth or mark passes the sensor, a step formed voltage is generated. The so-called pulse train, which is generated as the wheel rotates, can then be used for calculating the angular speed of the wheel. In Fig. 1(a, b) the speed sensor and the toothed flywheel is shown for a 6-cylinder diesel engine SL90-SL8800TA.

The estimation algorithm used to calculate the angular velocity at every new edge from the transmitter wheel signal is based on finite automaton theory. The crankshaft angle can be measured only every six degrees, i.e when a positive edge on the transmitter wheel occurs. The estimation algorithm uses the last calculated angular velocity to decide the position of the crank shaft. The angular velocity is calculated, when a positive edge appears, using this differential Eq. 2;

$$\omega = \frac{\Delta\varphi}{\Delta t} \qquad (2)$$

where $\Delta\varphi$ is the known sector angle described by the set of pulses for which the engine speed is measured and Δt is the measured time. The time is usually measured by a digital timer which is controlled by the zero-crossings of the pulse signal.

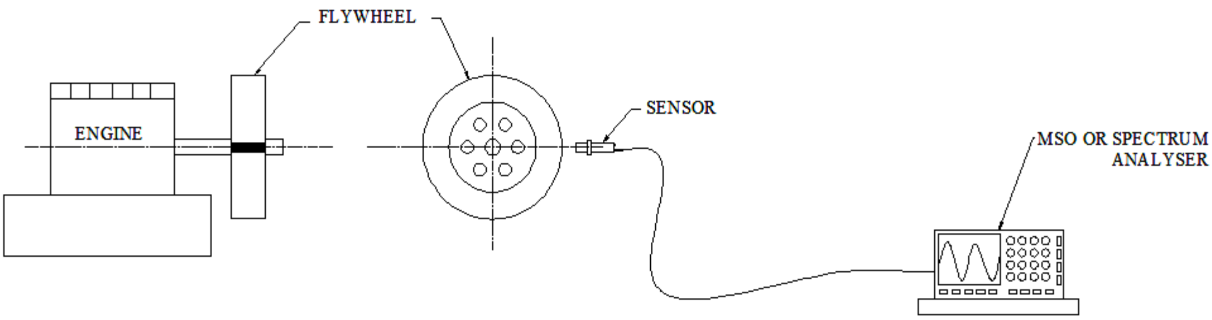

Figure 1: (a). Schematic Layout of test set up for Measurement of flywheel speed.

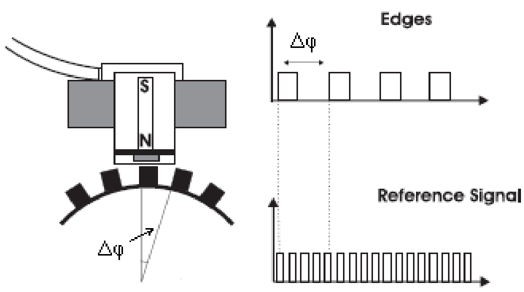

Figure 1 (b). Time measurement between two edges.

2.2. Measurement Noise:

From the experimental investigations it is found that the measured engine speed will now contain mainly two types of measurement noise:

1) Quantization noise.
2) Geometric noise.

2.2.1. Quantization Noise:

The quantization noise is due to the finite sample rate of the timer, which has a limited time resolution. In other words, the faster the shaft rotates, the higher the quantization noise *'q'* becomes. The quantization noise in rpm can be expressed as function of the timer frequency and the rotational speed of the engine as;

$$q = N_r \left[\frac{1}{1 - \frac{f_s(N_r)}{f_t}} - 1 \right] \quad (3)$$

where N_r is the engine speed in rpm, f_t is the timer frequency and f_s is the sample frequency of the engine speed. Note that f_s is a function of the rotational speed N_r. In Fig. 2 the dependency between the quantization noise and the rotational speed given by Eq. 3.

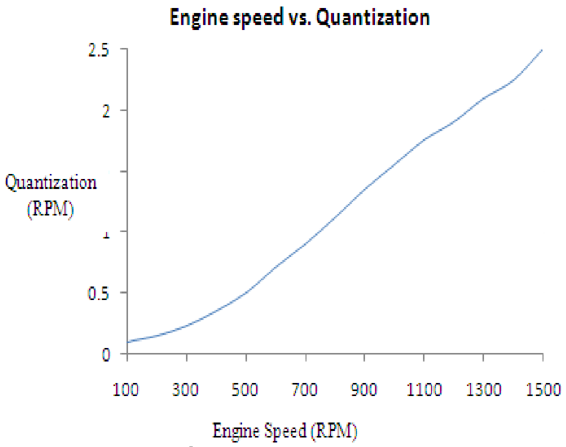

Figure 2: Engine speed vs. quantization noise.

quadratic with the engine speed. As a consequence, the information from orders with low amplitudes will be deteriorated or even lost. There are several methods to reduce the influence of quantization noise. One revisited method is to introduce dither, i.e. with noise, to the signal before performing the A/D conversion. However, considering the used measurement technique this was not directly applicable.

Quantization noise is a broadband noise which is distributed uniformly throughout the whole frequency band of the signal. As the gas and mass torques excite a known subset of frequencies, the total influence of the quantization noise can be reduced by considering only the known frequency orders which should be in the signal by using, for example, Discrete Fourier Transform analysis. Since the number of interesting frequencies are quite few there are, however, more effective approaches. For applications where only a limited set of frequencies are needed it can be more efficient to use a Goertzel filter instead for determining the Fourier series. Several Kalman filter approaches have also been suggested in order to damp the effects of quantization noise, where the dynamics of the system are taken into account, (Kiencke, 1999).

2.2.2. Geometric Errors:

The second source of noise is due to the geometric errors of the sensor disk, e.g. the flywheel. As all mechanical processing of the flywheel is done with a given accuracy, the teeth or holes drilled to the sensor disk include geometric errors. Consequently, the angle Δφ includes an unknown error, which affects the measured angular speed ω. An interesting property of the geometric noise is that it repeats itself after each revolution. This implies that that the energy due to the geometric noise is concentrated to integer multiples of the rotational frequency of the shaft.

A number of methods have been proposed to tackle the problem of sensor wheel having geometric errors. Assuming that the torque affecting the shaft is constant, the measured torsional vibration of the shaft depends only on natural frequencies of the shaft and geometric errors. Assuming that the period of the natural frequencies are not orthogonal to the rotational speed of the shaft, the geometric errors can be determined by averaging the oscillations over several rotations. This would also be possible for internal combustion engines. However, since the gas torques contain multiples of half the rotational frequency, half of the frequencies from the geometric errors will coincide with a subset of the torque orders and cannot be separated from each other. A method has been proposed where a lumped-mass model is used for producing a reference signal to the measured angular speed. The discrepancy between the reference and measured signal is then associated to the geometric errors of the sensor disk.

2.3. Identification of the Torque Noise Due to Geometric Errors:

Several methods have been proposed for identification and compensation of geometric errors of a sensor disk. None of these make use of fuel-injection adjustments and the known resulting behaviour of the vibratory orders of the crankshaft. In this paragraph, a method for identification of the geometric error for a given order is outlined.

Suppose that the gas torque of an engine is reconstructed from angular speed measurements by using, for example, a lumped-mass model. The reconstructed gas torque order T_p vector can now be expressed by Eq.4 as;

$$T_p = \dot{T}_p + e_p \quad (4)$$

where $\dot{T}_p$ is the actual complex-valued gas torque of order p and e_p is the complex-valued error due to the geometric errors of the sensor wheels. As the order-wise geometric errors are independent of changes in the fuel-injection durations, it follows that e_p can be determined by parameter estimation through adjustments of the cylinder-wise torque contributions.

To visualize this, the torque noise can be illustrated geometrically, Fig. 3. Suppose the gas torque vector T_p for a specific order p has been reconstructed from measurements of angular speeds, where, $T_1 = \hat{T}_1 + e_p$. By performing a small decrease in the overall output power of the engine, a new vector T_2 for the same order p can be calculated. Since the relative change in the torque profile of the cylinders is zero, it can be assumed that $\hat{T}_1$ and $\hat{T}_2$ have the same phase, but different amplitudes. Note that the torque noise vector e_p is unaffected by the change in gas torque. A line L_1 can now be drawn between the coordinates of T_1 and T_2 which is parallel with the true unknown gas-torque vectors $\hat{T}_1$ and $\hat{T}_2$. Consequently, it can be established that the end point of the torque-noise vector e_p is somewhere on the line L_1.

Let now the relative torque profile of the cylinders change significantly, in such way that the phase of the new torque vector T_3 is different from T_1 and T_2. By again performing a change in the overall engine output power, a new torque order vector T_4 can be established. In analogy with T_1 and T_2, a *line* L_2 can be determined on which the end point of the torque-noise vector e_p resides. As a result, the torque-noise vector is described by the vector between origin and the intersection of L_1 and L_2.

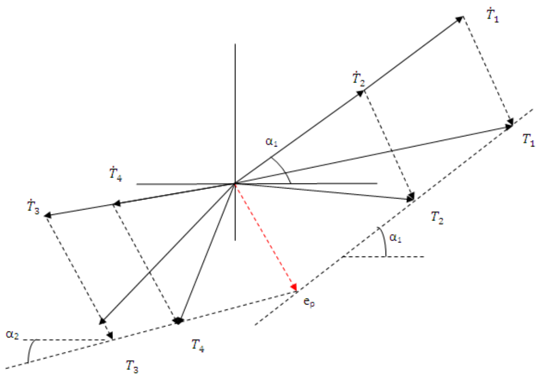

Figure 3: Graph of the relationship between the gas torque and the geometric torque noise.

3. Conclusions

The method proposed in this paper for identification and compensation of geometric errors of a transmitter wheel make use of fuel-injection adjustments and the known resulting behaviour of the vibratory orders of the engine crankshaft. From the performed analysis it is found that as the influence of mass torque and noise due to geometric noise is constant on the reconstructed oscillating torque for small fuel injection adjustments, the resulting change in the oscillating torque can be directly attributed to the fuel injection and gas torque change. As the quantization noise increases quadratic with the engine speed cause the information from orders with low amplitudes will be deteriorated or even lost.

Acknowledgement

Authors would like to thank Mr. G. S. Supekar, Sr. General Manager (R&E), and Mr. Shrikrishna Pathak, Asst. Manager, of Kirloskar Oil Engines (Corporate R&E) Pune-03 for providing necessary experimental inputs. Again the authors would like to thank Mr. Sudarshan Mudhe & Mr. Avinash Shinde (R&D) of M/s Greaves Cotton Limited Pune-19 for co-operation. Again the authors would like to thank Prof. A.J. Hake, Vice-Principal of M.E.S.C.O.E. & other college authority for providing necessary facilities. Authors thank BCUD Department of University of Pune for providing necessary financial assistance.

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