

Non-Intrusive Methodology for Estimation of Speed Fluctuations in Automotive Turbochargers under Unsteady Flow Conditions

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ABSTRACT

The optimization of turbocharging systems for automotive applications has become crucial in order to increase engine performance and meet the requirements for pollutant emissions and fuel consumption reduction. Unfortunately, performing an optimal turbocharging system control is very difficult, mainly due to the fact that the flow through compressor and turbine is highly unsteady, while only steady flow maps are usually provided by the manufacturer. For these reasons, one of the most important quantities to be used onboard for optimal turbocharger system control is the rotational speed fluctuation, since it provides information both on turbocharger operating point and on the energy of the unsteady flow in the intake and exhaust circuits.

This work presents a methodology that allows determining the instantaneous turbocharger rotational speed through a proper frequency processing of the signal coming from one accelerometer mounted on the turbocharger compressor. Consequently, the developed algorithm can be used to determine both rotational speed mean value and the amplitude of speed fluctuations that are caused by unsteady flows. From this last evaluated quantity, it is also possible to obtain an estimation of power delivered by the turbine, that might be used for control and diagnostic purposes.

The whole estimation algorithm has been developed and validated for a light duty turbocharged Diesel engine installed in a test cell at the University of Bologna. This paper reports the experimental layout used in this work and the accuracy obtained applying the speed fluctuations estimation procedure to the turbocharged Diesel engine under study.

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INTRODUCTION

Turbocharging technique is playing a fundamental role in automotive applications not only to improve engine performance, but also to obtain a significant reduction in fuel consumption and exhaust emissions both in Spark Ignition and Compression Ignition engines. As a matter of fact, in order to achieve such reduction, one of the most investigated fields of research is the combination of turbocharging technique and engine downsizing, that consists in engine displacement reduction aimed at engine global efficiency increase [1].

In this scenario, optimal turbocharging system management becomes crucial to achieve high engine efficiency and excellent drivability. However, the optimization of turbocharging system control is very difficult, especially for turbochargers working over a wide operating range and coupled to engines with a reduced number of cylinders. The first issue to be

addressed is the fact that maps provided by the turbocharger manufacturer usually do not cover the lowest rotational speeds, that become predominant in normalized driving cycles [2]. In addition, the most important limit to be taken into account is the lack of information on turbocharger unsteady flow performance, since only steady flow maps are usually available for both compressor and turbine [3].

Many works, carried out over the last years, demonstrate the feasibility of rotational speed calculation based on a proper processing of vibration [4] or acoustic emission [5] signals. The main limit of these methodologies consists in the fact that only turbocharger rotational speed mean value (evaluated over a complete revolution) can be estimated, while no information about the amplitude of speed fluctuations is provided. The knowledge of this quantity would be crucial for the investigation of phenomena due to unsteady flow conditions, such as those in interpolation/extrapolation of flow maps [6] or instabilities at

low-flow range [7], and to determine the actual power delivered by the turbine, as it will be discussed in the following. Due to the above reasons, this work has been carried out to demonstrate that the vibration signal coming from one accelerometer mounted on the compressor diffuser can be used to extract both turbocharger speed mean value and speed fluctuations due to pulsating flows.

The whole methodology has been developed for a 4-cylinder Common-Rail light-duty Diesel engine installed in a test cell at the University of Bologna. The first section of the paper describes the technical characteristics of the engine-turbocharger system under investigation and the sensors used throughout this work. Then, a methodology that allows calculating turbocharger rotational speed mean value starting from the acquired accelerometer signal is discussed in the second section. Finally, the last section of the paper demonstrates that also turbocharger speed fluctuations can be extracted through a proper processing of the vibration signal. In order to highlight the importance of speed fluctuations amplitude evaluation, some experimental results are reported to prove that this quantity provides important information about turbocharger operating condition, such as the actual power delivered by the turbine.

EXPERIMENTAL SETUP

The whole work discussed in this paper has been developed analyzing a set of specifically designed tests acquired running a light-duty Common-Rail Diesel engine mounted in a test cell. The main technical characteristics of the engine under investigation are summarized in [Table 1](#).

Table 1. Engine characteristics.

Displacement	1248 cm ³
Maximum Power	70 kW @ 3800 rpm
Maximum Torque	200 Nm @ 1500 rpm
Injection System	Common Rail Multi-Jet
Compression ratio	16.8
Architecture	L4, firing order 1-3-4-2

In order to take into account most of the engine operating range, 16 tests have been run at different speed and load levels. The operating points of interest are summarized in [Table 2](#).

Table 2. Engine operating points analyzed in this work.

		IMEP [bar]			
		3	8	14	20
Engine Speed [rpm]	1750	Test 1	Test 2	Test 3	Test 4
	2000	Test 5	Test 6	Test 7	Test 8
	2500	Test 9	Test 10	Test 11	Test 12
	3000	Test 13	Test 14	Test 15	Test 16

With regard to the engine boosting system, it is a Borg Warner's BV35 with Variable Geometry Turbine (VGT) and maximum speed approximately equal to 250000 rpm. In order to monitor turbocharger operating conditions, pressure and temperature have been measured in both compressor and turbine upstream and downstream circuits.

In addition, compressor diffuser has been modified to install an eddy-current sensor faced to the compressor rotor blades. The rotational speed sensor used in this work is a Micro-Epsilon TurboSpeed-135, that returns one analog output (voltage) proportional to rotational speed (in this case 1V corresponds to 40000 rpm) and 2 digital outputs (0-5V) characterized by a rising transition in correspondence of a compressor blade or a compressor rotation (10 blades) respectively. In order to maximize the accuracy of turbocharger speed measurement, the transitions of the digital outputs coming from the Micro-Epsilon sensor have been detected using a data acquisition board with clock frequency equal to 20 MHz.

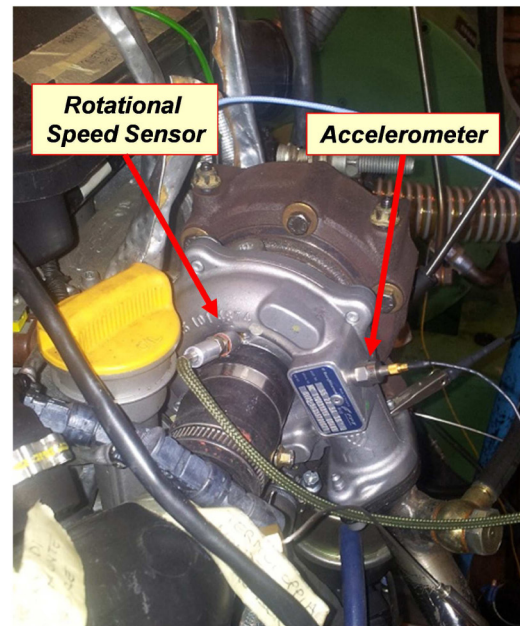


Figure 1. Speed sensor and accelerometer installation on compressor diffuser.

The last sensor applied to the turbocharger is a high frequency ICP accelerometer (PCB Piezotronics Model 621B40). During each test, the acceleration signal has been sampled at 100 kHz.

It is easy to understand that both speed and vibration sensors have been applied to compressor diffuser because upstream and downstream temperatures are much lower in compressor circuit than in turbine circuit. [Figure 1](#) shows rotational speed and accelerometer sensors location in the compressor diffuser.

TURBOCHARGER SPEED CALCULATION

In order to verify the feasibility of turbocharger rotational speed estimation, the frequency spectrum of the accelerometer signal has been investigated. In particular, it has been verified that the accelerometer signal spectrum shows a high energy content in correspondence of the revolution frequency (f_R) or one of its multiples, such as the blade frequency (f_B) [4,5]. In order to clarify this consideration, Figure 2 shows the frequency spectrum of the vibration signal acquired during Test 5, evaluated through the Short Time Fourier Transform algorithm.

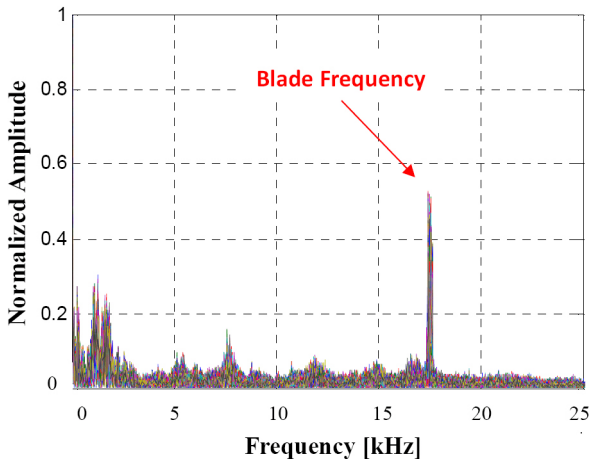


Figure 2. Comparison between the accelerometer signal spectrum and the blade frequency calculated from the turbocharger speed measurement.

The plot shows a significant energy in the low frequency range (lower than 5 kHz) and in correspondence of the blade frequency, that can be correlated to the rotational speed through Equation (1).

$$f_B [Hz] = \omega [rpm] \cdot \frac{N_B}{60} \quad (1)$$

Here f_B is the rotational speed (in rpm) evaluated using the Micro-Epsilon speed sensor, while N_B is the blade number (10 for the compressor under study). The same consideration applies to all the tests reported in Table 2, i.e. it is always possible to observe a significant energy concentration in correspondence of the harmonic component associated to the compressor blade frequency.

Due to the above considerations, in order to extract the information related to the turbocharger speed rotation from the vibration signal, it is necessary to identify the harmonic component corresponding to the blade frequency and to track it in every engine operating condition. Unfortunately, the vibration signal spectrum contains a high energy content also in the frequency range usually associated to phenomena due to the combustion process, i.e. in the frequency range lower than 5 kHz (clearly visible in Figure 2). Therefore, the vibration signal has to be filtered in a proper way to highlight the blade

frequency of interest. However, since the blade frequency to be tracked depends on turbocharger rotational speed, a proper accelerometer filter can be applied only if a rough turbo speed estimation can be carried out a priori, i.e. depending on the engine operating point.

To obtain a rough estimation of the frequency range to be considered for turbocharger rotational speed estimation, the compressor characteristic map shown in Figure 3 can be used. Here, the ratio between inlet and outlet compressor pressure is shown as a function of the reduced mass flow rate for different compressor speeds.

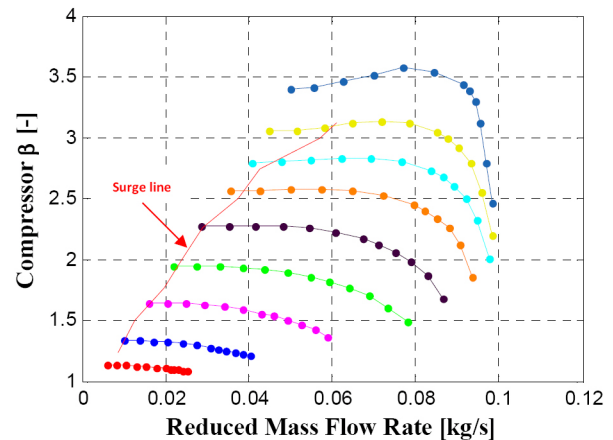


Figure 3. Characteristic map for the compressor under study.

The first aspect to be addressed is that the real operating range of the compressor can exceed the operating region shown in Figure 3 (the surge line can be exceeded) [1,3,6]. To avoid interpolation errors, the compressor map has been extrapolated according to the methodology presented by El Hadeif et al. [2]. Then, in order to obtain a rough turbocharger speed estimation, the characteristic map can be inverted to obtain a 2D lookup table that outputs compressor speed as a function of mass flow and pressure ratio. The obtained lookup table is reported in Figure 4. Since both mass flow and pressure ratio can be directly measured by the ECU, the inverted map can be used for turbocharger speed real-time estimation.

As it is well known, the accuracy of turbocharger speed estimation obtained interpolating the inverted compressor map is not compatible with the requirements for engine control [5,7], because it suffers from measurement errors and part to part variations. However, this quantity can be useful to set up the band-pass filter that allows maximizing the information related to the blade frequency in the accelerometer signal spectrum. In this approach, the upper and the lower cutoff frequencies of the band-pass filter have been determined respectively adding and subtracting 10% to the blade frequency calculated starting from the turbocharger speed evaluated using the inverted map. To clarify this consideration, Figure 5 shows the filtered accelerometer signal spectrum for the same test reported in Figure 2.