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Automotive turbochargers power estimation based on speed fluctuation analysis

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Abstract

Turbocharging technology will play a crucial role in the near future as a way to meet the requirements for pollutant emissions and fuel consumption reduction.
However, optimal turbocharger control is still an issue, especially for downsized engines fitted with a low number of cylinders. As a matter of fact, automotive turbochargers are characterized by wide operating range and unsteady gas flow through the turbine, while only steady flow maps are usually provided by the manufacturer. In addition, in passenger cars applications, real-time turbocharger optimal control is even more difficult because of the lack of information about pressure/temperature in turbine upstream/downstream circuits and turbocharger rotational speed.
In order to overcome these unknowns, this work presents a methodology for instantaneous turbocharger rotational speed determination through a proper processing of the signal coming from one accelerometer mounted on the compressor diffuser, or one microphone facing the compressor. The presented approach can be used to evaluate both turbocharger speed mean value and the amplitude of turbocharger speed fluctuations caused by the pulsating gas flow in turbine upstream and downstream circuits. Once turbocharger speed has been determined, it can be used to estimate power delivered by the turbine.
The whole estimation algorithm has been developed and validated for a light duty turbocharged Common-Rail Diesel engine mounted in a test cell. However, the developed methodology is general and can be applied to different turbochargers, both for Spark Ignited and Diesel applications.

Keywords: Turbocharger speed; Pulsating flow, Turbine power, Turbocharger control

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1. Introduction

Due to the increasing request for pollutant emissions and fuel consumption reduction, the optimization of turbocharger control has become a critical issue in modern engine management systems. Prior research demonstrates that pollutant emissions reduction and higher engine efficiency can be achieved through a proper combination of turbocharging technique and engine downsizing [1]. However, turbocharger optimization is a critical task, mainly due to the few quantities related to turbocharger operating conditions that can be directly measured on-board. Moreover, optimal turbocharger control becomes even more difficult when turbochargers work over a wide operating range, since turbocharger manufacturers usually provide only steady flow maps that do not cover low rotational speeds (that may become predominant in standard driving cycles) [2]. Another important aspect to be addressed, especially for turbochargers coupled to downsized engines with a reduced number of cylinders, is the lack of information about unsteady operating conditions, since only steady flow maps are usually available for compressor and turbine [3].

In order to optimize turbocharger control, this work presents a methodology that allows extracting information about turbocharger operating conditions through a proper processing of the signal coming from one accelerometer mounted on the compressor diffuser or a microphone facing the turbocharger. In particular, the developed methodology is suitable for the estimation of both turbocharger rotational speed (mean value)and the amplitude of its instantaneous oscillations. Then, such oscillations can be used as an indicator of power delivered by the turbine. 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.

Many works, carried out over the past years, demonstrate that turbocharger speed can be estimated through a proper processing of vibration [4] or acoustic emission signals [5]. The main limitation of these methodologies consists in the fact that only turbocharger speed mean value can be estimated, while no information about the amplitude of speed fluctuations can be extracted. However, the on-board knowledge of this quantity would be very important to compensate the effects due to pulsating flows, such as errors in maps interpolation/extrapolation [6] or instabilities at low flow range [7]. Furthermore, this work demonstrates that real-time knowledge of turbocharger speed fluctuations provides information about turbine power, which could be a useful feedback for an optimal turbocharger control strategy.

Nomenclature

$\overline{\boldsymbol{\omega}}_{TC}$	Turbocharger average speed
J_{TC}	Turbocharger inertia
$\dot{\boldsymbol{\omega}}_{TC4}$	Turbocharger acceleration order 2 component
$\boldsymbol{\omega}_{TC4}$	Turbocharger speed order 2 component
ω_{TC}	Turbocharger speed
ω_{eng}	Engine rotational speed
γ	Specific heat ratio
T_3	Temperature in the turbine upstream circuit

β_T	Turbine pressure ratio
η_T	Turbine total efficiency
$\dot{m}_T$	Turbine mass flow
P_C	Power requested by the compressor
P_T	Power delivered by the turbine
N_B	Number of blades in the compressor rotor
f_B	Compressor blade frequency
bmep	Break Mean Effective Pressure
α	Proportional relationship coefficient between turbine average power and turbine pulsating power
η_o	Turbocharger mechanical efficiency
$\mathbf{P}_{T\,4}$	Turbine power order 2 component

2. Experimental setup

The estimation algorithm presented in this paper has been developed and validated starting from sets of specifically designed tests carried out running a 1.3L Common-Rail Diesel engine installed in a test cell at the University of Bologna.In order to investigate most of the operating range, 16 tests have been run at different speed and load levels. Table 1 reports the operating conditions of interest.

Table 1. Engine operating points taken into account in this work

Engine Speed	BMEP			
[rpm]	3 [bar]	8 [bar]	14 [bar]	20 [bar]
1500	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

The engine is equipped with a Borg Warner's BV35 turbocharger with Variable Geometry Turbine (VGT) and maximum rotational speed approximately equal to 250000 rpm (maximum rotational speed that allows avoiding mechanical damages).In order to measure turbocharger rotational speed directly, the compressor diffuser has been modified to install an eddy-current speed sensor (a Micro-Epsilon TurboSpeed-135 directly facing the rotor blades). This sensor returns oneanalog output (voltage signal proportional to turbocharger rotational speed) and two digital outputs (0-5V) characterized by a rising transition in correspondence of a compressor blade or a compressor rotation (10 blades, in this case) respectively. In order to maximize measurement accuracy, in this work, turbo speed measurement has been performed using the digital output corresponding to compressor rotor blades, detected using a data acquisition board with sampling frequency equal to 20 MHz. In addition, one high frequency ICP accelerometer (PCB Piezotronics Model 621B40) has been mounted on the compressor diffuser, while a microphone (PCB Piezotronics Model 378C01) has been fixed installed in front of the compressor diffuser. During each test, the signals coming from both sensors have been sampled at 500 kHz. All the sensors applied to the turbocharger have been fixed to the compressor, since upstream and downstream temperatures are much lower in the compressor circuit than in the turbine circuit.

3. Analysis of the relationship between speed fluctuations and turbine power

Turbocharger speed can be considered as the result of 2 contributions: a low frequency component (mean value over a complete rotation) and a high frequency component (fluctuation).The high frequency fluctuation is mainly due to the pulsating flow coming from the engine after exhaust valves opening. Consequently, the frequency of turbocharger speed fluctuation will always be approximately equal to engine combustion frequency, while its amplitudewill be correlated to engine load. In order to clarify this aspect, turbocharger can be modeled as a dynamic system kept in motion by a pulsating power with frequency equal to engine combustion frequency. The balance between power delivered by the turbine and power required by the compressor can be written in the angular frequency domainthrough Eq. (1). Average values of the Fourier expansion return the usual dynamic equation for the turbocharger.

$$\left(\overline{\mathbf{P}}_T - \overline{\mathbf{P}}_C\right) \cdot \eta_o = J_{TC} \cdot \overline{\boldsymbol{\omega}}_{TC} \cdot \dot{\overline{\boldsymbol{\omega}}}_{TC} \tag{1}$$

Since the engine under investigation performs 4 evenly-spaced combustions per cycle, turbocharger speed fluctuation will show a high frequency content in correspondence of engine order 2 (characteristic engine order, that corresponds to harmonic 4 calculated over the engine cycle). Considering that only turbine power will show a high order 2 fluctuation, due to the pulsating exhaust gas flow entering the turbine, compressor power fluctuation can be neglected:

$$\mathbf{P}_{T4} \cdot \eta_o = J_{TC} \cdot \overline{\boldsymbol{\omega}}_{TC} \cdot \dot{\boldsymbol{\omega}}_{TC4} \tag{2}$$

Assuming that a linear correlation between the amplitude of power fluctuation and power mean value exists [8,9] (α the proportional correlation coefficient) and taking into account that the order 2 fluctuation of turbocharger speed first derivative ($\dot{\boldsymbol{\omega}}_{TC4}$) can be correlated to the corresponding turbocharger speed fluctuation ($\boldsymbol{\omega}_{TC4}$), the equations above can be rearranged in order to express turbine power mean value ($\overline{P}_T$) as a function of turbocharger speed, yielding:

$$\alpha \cdot \eta_o \cdot \overline{\mathbf{P}}_T = 2.0 \cdot J_{TC} \cdot \left|\boldsymbol{\omega}_{TC4}\right| \cdot \omega_{eng} \cdot \overline{\boldsymbol{\omega}}_{TC} \tag{3}$$

The relationship expressed in Eq.(3) has been identified through experimental data acquired during the experimental tests run with the engine under investigation. With regard to power delivered by the turbine, it can be directly calculated through Eq.(4).

$$P_T = \dot{m}_T \cdot \eta_T \cdot T_4 \cdot \left(\beta_T^{\frac{\gamma-1}{\gamma}} - 1 \right) \tag{4}$$

Here, $\dot{m}$ is the reduced turbine mass flow, η_T is turbine efficiency(determined using the maps provided by the turbocharger manufacturer), T_4 is temperature measured in the turbine downstream circuit and β_T is the ratio between turbine inlet and outlet pressures.A unique correlation between turbine power and turbocharger speed can be obtained rearranging Eq.(3) as follows:

$$\frac{\overline{\mathbf{P}}_T}{\omega_{eng}} = 2.0 \cdot \frac{J_{TC}}{\alpha \cdot \eta_o} \cdot \left|\boldsymbol{\omega}_{TC\,4}\right| \cdot \overline{\boldsymbol{\omega}}_{TC} = K \cdot \left|\boldsymbol{\omega}_{TC\,4}\right| \cdot \overline{\boldsymbol{\omega}}_{TC} \quad (5)$$

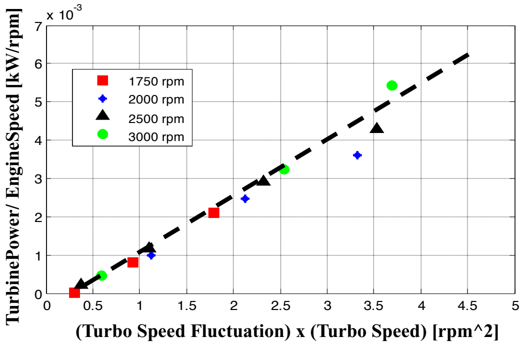

Fig. 1. Correlation between normalized turbine power and the product between the amplitude of turbocharger speed fluctuation and turbo speed mean value

K represents the linear correlation existing between normalized turbine power (with respect engine speed) and the product between the amplitude of turbo speed fluctuation and its mean value.The result reported in Figure 1 clarifies that the knowledge of instantaneous turbocharger rotational speed (usually unavailable onboard) provides important information about power delivered by the turbine, which would be very useful to optimize turbocharger control and cooling.

4. Instantaneous turbocharger speed estimation

As above discussed, the innovative approach presented in this work allows determining both turbo speed mean value and the amplitude of its fluctuation, which is directly correlated to power delivered by the turbine.The first step of the methodology consists in the estimation of turbocharger speed mean value through a proper spectral analysis of the signals coming from accelerometer and microphone. The main idea behind this part of the methodology is to set up an algorithm for automatic detection of the energy peak corresponding to the turbocharger blade frequency (f_B) [7]. Figure 2 shows the frequency spectra evaluated for acceleration and acoustic emission signals acquired during a test run at 2000 rpm and bmep=14 bar. As it can be observed, both spectra show a high energy peak corresponding to the compressor blade frequency (f_B), that is directly correlated to turbocharger speed mean value (over a complete rotation) through Eq.(6).

$$f_B\,[Hz] = \omega_{TC}\,[rpm] \cdot \frac{N_B}{60} \quad (6)$$

Therefore, turbo speed mean value can be real-time calculated simply tracking the energy peak associated to the compressor blade frequency in the spectra evaluated for acceleration or acoustic emission signals.

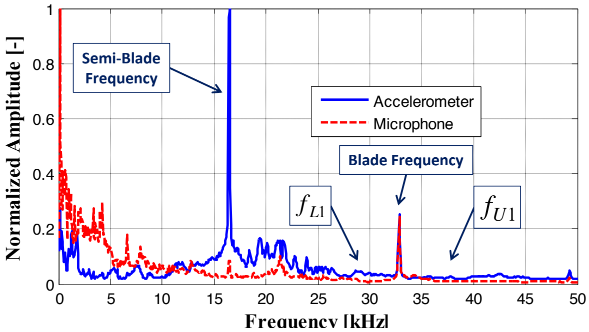

Fig. 2. Comparison between accelerometer (solid line) and acoustic emission (dashed line) frequency spectra

Unfortunately, the amplitude of the energy peak corresponding to the compressor blade frequency is usually lower than other frequency contents related to the combustion process (especially for the acoustic emission signal). Therefore, it is not possible to set up an automatic algorithm for blade frequency detection through the analysis of the complete spectrum. In order to overcome this problem, a proper band-pass filter has been applied to the acquired vibration signals (acceleration and acoustic emission). The cut-off frequencies of the band-pass filter can be established according to the compressor inverted characteristic map, that provides a rough turbo speed (i.e. blade frequency) estimation as a function of compressor pressure ratio and reduced mass flow. To do so, it has to be taken into account that the real operating range of the compressor can exceed the operating region characterized by the manufacturer [1,3,6]. To avoid interpolation errors, the compressor map has been extrapolated according to the methodology presented by El Hadef et al. [2].

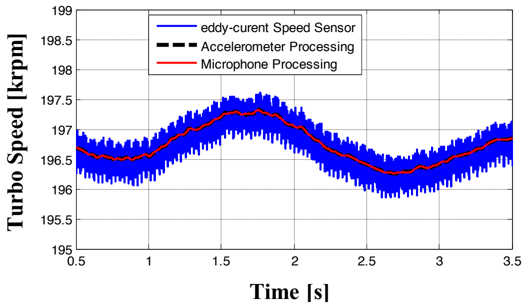

Fig. 3. Comparison between measured turbocharger speed (using the eddy-current sensor, blue line) and turbo speed estimated via accelerometer (black line) and audio signal (red dashed line) processing

Lower (f_{L1}) and upper (f_{U1}) cutoff frequencies have been chosen subtracting and adding 10% to the blade frequency estimated using the inverted compressor map [5,7], and the discussed band-pass filter has been applied to all the acquired signals (summarized in Table 1) and proved to be always effective for blade frequency tracking. The energy peak corresponding to the blade frequency can be automatically detected in the spectrum of the filtered signals. Therefore, turbo speed mean value can be estimated through the analysis of the acceleration or acoustic emission signals. Figure 3 reports, for the same test shown in Figure 2, a comparison between estimated (via accelerometer and acoustic signal processing) and measured (using the micro-epsilon eddy-current sensor) turbo speed.Even if the difference between measured and estimated mean turbo speed is negligible (root mean squared error lower than 0.1%), the methodology based on frequency spectra analysis is not suitable for speed fluctuation calculation.

Consequently, it is not possible to perform turbocharger power real-time estimation.

The limitations of thealgorithm based on real-time spectral analysis can be overcome using an innovative time-based procedure combined with the results obtained applying the methodology based on spectral analysis.First, it has to be considered that the information about compressor blade frequency, accurately determined using the frequency-based approach, can be used to set up a new band-pass filter for acceleration and acoustic emission signals. This filter can be more selective than the one based on turbocharger inverted map, since the actual value of compressor blade frequency (f_B) has been accurately determined. Therefore, it is possible to determine the lower (f_{L2}) and upper (f_{U2}) cutoff frequencies for the new filter. The choice of a proper band-pass filter allows selecting a limited region of acceleration and audio signal spectra in which the energy contribution due to the compressor blade frequency is highly predominant. Therefore, the output of the second selective filter is similar to a sine wave whose frequency is approximately equal to the instantaneous blade frequency.

Once the proper selective filter has been applied to acceleration or acoustic emission signals, the period (variable over time) of the filtered waveform can be identified simply by measuring the time interval between two corresponding zero-crossings (rising or falling). Finally, the calculated time intervals allow determining the corresponding instantaneous turbocharger speed(shown in Figure 4).

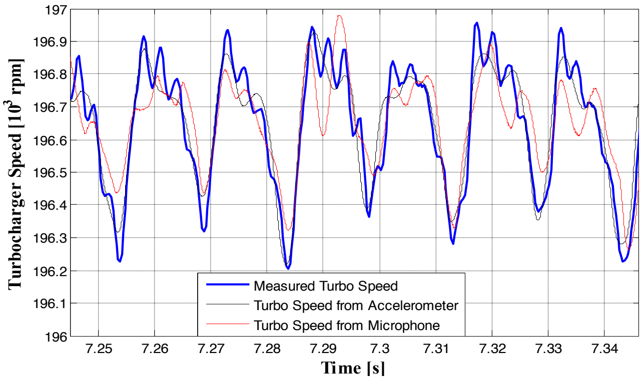

Fig. 4. Comparison between measured and estimated instantaneous turbo speed

As it can be observed in Figure 4, the complete estimation methodology presented in this work is suitable for instantaneous turbo speed estimation. Furthermore, the methodology proved to be effective for all the experimental tests summarized in Table 1. The accuracy of the obtained results seems to be compatible with on-board requirements for turbocharger control, since the root mean squared error between measured and estimated instantaneous turbo speed is lower than 0.1% both in case of acceleration and acoustic emission signal processing.

5. Conclusions

Turbocharger rotational speed is a very important quantity to be evaluated on-board, since it provides useful feedback information for optimizing turbocharger control strategies. As an example, many works demonstrate that the knowledge of turbo speed mean value allows optimizing compressor and turbine maps interpolation or extending turbocharger operating range. In addition, this work demonstrates that instantaneous turbo speed (i.e. the knowledge of both mean value and fluctuation) can be used to extract information about power delivered by the turbine.

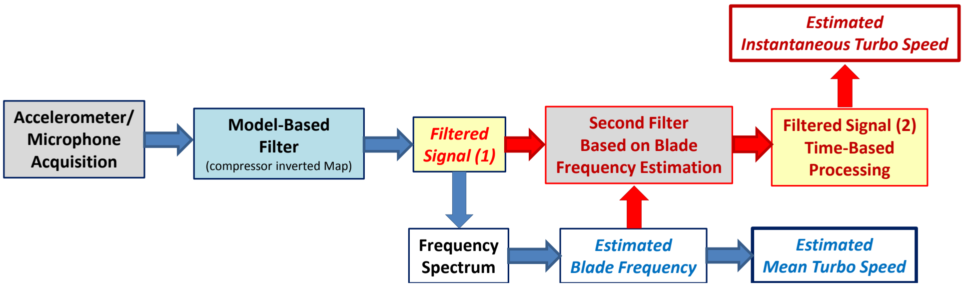

Fig. 5. Scheme of the complete instantaneous turbo speed estimation methodology

Therefore, this paper presents an innovative methodology for instantaneous turbo speed estimation. The presented algorithm is based on the analysis of the signals coming from one accelerometer mounted on the compressor diffuser, or from one microphone facing the turbocharger. The estimation methodology can be divided in 2 steps. The first step consists in the estimation of turbocharger speed mean value, performed through a proper spectral analysis of the acquired signals (accelerometer or microphone). The second step is a time-based processing, that starts from the estimated mean turbo speed to refine signal filtering and determine the instantaneous turbo speed. The scheme of the overall estimation procedure is reported in Figure 5. The developed methodology proved to be effective in all the operating conditions taken into account in this work.

Further investigations are currently being performed to implement the methodology in a Rapid Control Prototyping system, the goal being to develop an optimal turbocharger control strategy based on instantaneous turbo speed estimation.

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