# MATLAB BUILT ROUTINE FOR TURBOCHARGER VIBRATION SURVEY POST-PROCESSING

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**Abstract**

**The paper presents current procedure of the analysis and post processing of raw data measured during vibration survey on the turbocharger used in Honeywell Turbo Technologies (HTT). The test itself is standardized procedure of recording vibration data with the turbocharger physically mounted and operating on a combustion engine. The test is run under laboratory conditions with the combustion engine being controlled by dynometer. Vibration acceleration is recorded at several locations with standard commercial 1D or 3D accelerometers with common number of 15+ channels. Engine conditions are preprogrammed with varying both speed and load so that the collected data represent vibration signature during of the given product for wide range of operating regimes. Raw data are being stored for post–processing during which the focus is put on: structural resonant frequencies, harmonic components in the vibration, noise level and power spectral density courses. Results are being compared to the product specifications. Brief methodology is described with the example of actual data collected on a two stage turbocharged heavy duty application with the journal and ball bearing rotating assembly. Pattern of the harmonic frequencies of both turbocharger and engine is identified with the main contributing components plotted separately.**

## 1 Introduction

Turbocharger vibration signature is one of the key tests for both the turbocharger development and product qualification. The qualification intended data analysis is concentrated on overall peak values of the frequency response (peak-hold) and identification of the main structural resonances in the frequency range given by the formula

$$f_{\max} = N_{\text{harm\_max}} \frac{N_{\text{cyl}}}{2} N_{\text{ERPM\_max}}\,, \qquad (1)$$

where $f_{\max}$ is the maximum frequency in which the resonance could have impact on structural integrity, $N_{\text{harm\_max}}$ is the maximum engine speed harmonic count of the interest , $N_{\text{cyl}}$ is number of cylinders of the engine and $N_{\text{ERPM\_max}}$ is the maximum engine speed. The engine load is usually set to 100% over the entire rpm spectrum. Depending on the situation, more attention can be paid to the vibration load during concrete regimes, such as peak torque range, rated speed etc. Special case is the engine braking mode which requires separate measurement for each configuration as the engine braking strategies vary among the applications.

The gathered data provide fundamental information about the vibration load that the turbocharger will be exposed to during operation. Therefore the configuration at which the test is run should be as close as possible to the production design. There are several approaches in the post processing however all should have the standardized outputs such as peak-hold plots of acceleration, Campbell plots (combined with waterfall plot), PSD plots etc, effective values of the acceleration for given frequency range, RMS values etc.

Measurement and analysis for development purposes has different objectives than just a direct comparison to the vibration level specifications:

- Monitoring of the TC condition during accelerated testing
- Evaluation of the proposed design changes

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Vibration analysis is used as a diagnostic tool when comparing e.g. two design configurations against each other or baseline vs. new design.

## 2 Case study

The concrete data were analyzed to demonstrate methodology used. Data were collected during the development of the two stage application (high pressure – HP stage and low pressure LP stage) and comparative study of ball bearing vs. journal bearing rotor groups. Measurement was recorded under several engine load regimes as shown in Tab. 1.

Table 1:Test arrangement: configurations and engine loads

| | HP with support bracket | | | HP w/o bracket support | | |
|---|---|---|---|---|---|---|
| LP stage/HP stage JB/JB | 100% load | 0% load | Brake on | 100% load | 0% load | Brake on |
| LP stage/HP stage BB/BB | 100% load | 0% load | Brake on | 100% load | 0% load | Brake on |

### 2.1 Data Acquisition and analysis

Analyzed measurement data were recorded during the engine ramp up test on heavy duty application in Honeywell Turbo lab in Brno. The application was two stage 6 cylinder 13 liter diesel engine attached to the actively controlled dynometer of the max power of 640 kW. Five tri-axial accelerometers by PCB were used with the placement as per Fig. 1. and Tab. 2. The raw data were recorded by NI unit with PXI-4496 acquisition card (16 channels) in the sampling frequency of 10 kHz. Acceleration raw data were measured against the ramped engine speed – Total 16 channels were used for recording (one for engine speed track).

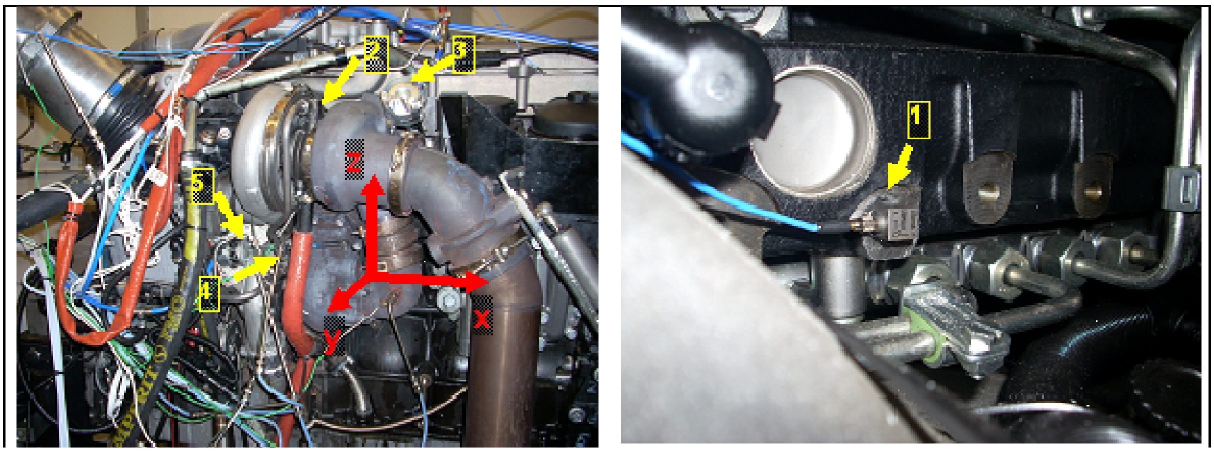
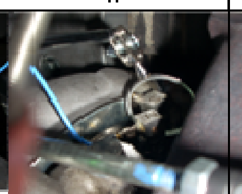



Figure 1: Location of the accelerometers during measurement

Table 2: Detailed description of the accelerometers location points

| 1. | 2. | 3. | 4. | 5. |
|---|---|---|---|---|
|  |  |  |  |  |
| Engine head | Turbo LP oil inlet | Turbo LP Actuator body | Turbo HP oil inlet | Turbo HP actuator bracket |

Recorded data were processed by the procedure programmed in Matlab based on built-in function fft(x). In the time domain each signal was divided into 'macro' windows of 1 sec and consequently, peak amplitudes were identified from each run @ each time window as well as a total RMS value of the signal and for each direction both peak value and RMS values were plotted versus the engine speed. In the frequency domain, FFT was performed under the following conditions:

- No. of samples per window: 2048
- Frequency resolution 1 Hz
- Hann filter over time window
- 75 % window overlap
- 1 Hz to 4200 Hz band pass digital filter in frequency domain

## 2.2 Data processing

The common outputs are:

1. overall peak hold plot (of the acceleration),
2. zero-peak plots for each channel and time window,
3. Campbell plot (and waterfall plot),
4. time domain (raw data) plots, effective value acceleration
5. PSD plots at required resolution,
6. Peak and effective value plot (RMS),
7. Band-pass effective value (denoted as $\mathrm{RMS_f}$) Eq. 2. or Root Mean Square ($\mathrm{RMS_t}$) as the effective value calculated from the time domain raw data as Eq. 3.:

$$\mathrm{RMS_t} = \sqrt{\frac{1}{N}\sum_{i=1}^{N}[x_i]^2} \tag{2}$$

$$\mathrm{RMS_f} = \sqrt{\int_{f_1}^{f_2} S_{xx}(f)df} \tag{3}$$

Fig. 2. shows the illustration of use of $\mathrm{RMS_f}$ and $\mathrm{RMS_t}$ values. Note that this is different measurement from the 'Case study': here the task was to run the validation of the electronic component that suits as a controller of the electric actuator. For that case the estimation of band-pass effective value $\mathrm{RMS_f}$ across 50-400 Hz was required. The left plot shows total RMS values measured in given locations whiles on the right is the band pass $\mathrm{RMS_f}$. After the extraction the high frequency content can be evaluated. This was important for the electric actuator PCB vibration load qualification.

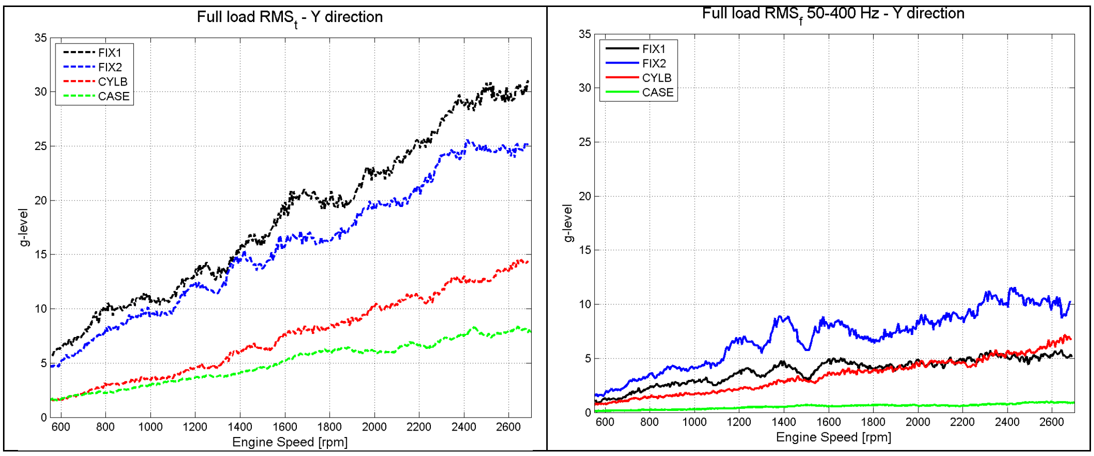



Figure 2: $\mathrm{RMS_t}$ and $\mathrm{RMS_f}$ values of acceleration from the actuator control electronic casing plotted against engine speed.

## 3 Turbo speed tracking

For some applications the turbocharger speed is measured by a speed sensor usually based on electromagnetic principle (Fig. 3.).

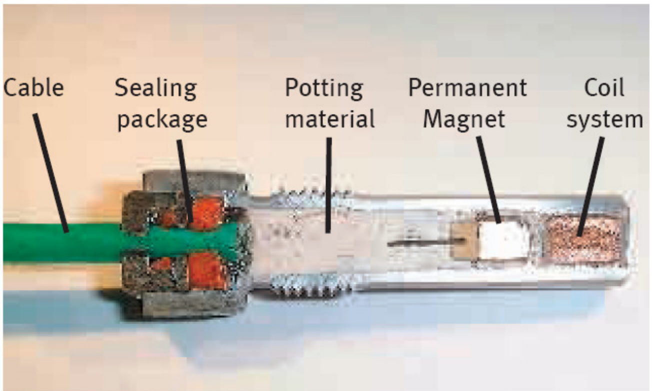



Figure 3: EM speed sensor by Jaquet AG

Many commercial applications do not have speed sensor installed, cost reduction being the primary reason and providing the turbocharger has sufficient operational margin for high elevation or limit regimes. If the speed sensor is available, the information represents additional data that can be used to evaluate turbo speed harmonic.

In the cases where the speed sensor is not present in standard configuration, it can be retrofitted for the testing purposes (optical or EM sensor) or the acceleration signal from the center housing (rotor housing) can be used as a source of turbo speed information. This is then extracted ether online during the measurement of during post processing by a simple routine that finds the peak in the vicinity of estimated turbo speed in the signal FFT. If 3D accelerometer was used for on the center housing, it's useful to determine which direction the turbo speed first harmonic vibration is prevailing. Not always this would be the vector that normal to the rotor axis. The general principle can be described by the following formula:

$$TSPEED = f\left\{\max\left[X(f)\right]_{f_{\mathrm{TC\_EST}}-f_{\mathrm{VIC}}}^{f_{\mathrm{TC\_EST}}+f_{\mathrm{VIC}}}\right\}\times 60 \qquad [rpm], \tag{4}$$

where $X(f)$ represents the signal window, $f_{\mathrm{TC\_EST}}$ is the estimate of the 1$^{st}$ harmonic frequency made by guess, $f_{\mathrm{VIC}}$ is the chosen vicinity at which the search is being made.

Statistical loop can be added and/or floating mean value filter applied to utilize signal in points showing ambiguous results. The result of the procedure is demonstrated in the Fig. 6.

The most frequent difficulty when the measurement is performed on engine is the presence of significant portion of vibration due to the engine harmonics (firing frequency pulsation and higher harmonics). The turbo speed amplitude is usually high enough to be extracted easily, however for some cases it might be a difficult task to do, e.g. turbo low speed run during ramp up from the idle. For the multistage application it is also difficult to distinguish between acceleration patterns due to the rotor speed of the stages. In such cases it's assumed that each stage has a dominant frequency of its speed rather than of the rotor speed of other stages. This might again be dubious if the stages are sequential and so the load of each of the stages is changing across the time comparing to the other as seen of Fig. 7.

## 4 Results

From the various tested arrangement here only full load engine regime and with the support bracket installed on HP stage is presented. The plots in the Fig. 4. show the Campbell diagram from the rotor housing for LP and HP stage with the tracked turbo speed for both high pressure and low pressure stage. Selected direction is normal to the turbo axis except for the journal bearing HP stage where the output in longitudinal direction (in sense of turbocharger axis) was more suited for the analysis. It is obvious that the pattern for ball bearing and journal bearing application has some common features and differs in details such as dynamics of rotor acceleration. Also one can conclude that the journal bearing configuration has uneven rotor speed harmonic components possibly due to the oil film [2] (HP stage). Zoomed views on Fig. 5. show that the most significant source of acceleration is the 3$^{rd}$ and 4 ½$^{th}$ engine speed harmonic which is equal to 1× and 1.5× the firing frequency of the 6-cylinder (in-row) engine.

Ball bearing harmonic frequency is not covered by the measurement as the values would be well above the studied frequency range. After [3] the frequency that corresponds to the specific vibration is for inner race failure caused component:

$$f_{\mathrm{IN}} = f_{\mathrm{ROT}} \frac{n_{\mathrm{ROLLERS}}}{2}\left[1+\left(\frac{d_{\mathrm{ROLLER}}}{d_{\mathrm{PITCH}}}\right)\cos\beta\right], \tag{5}$$

where $f_{\mathrm{ROT}}$ is the frequency of rotation, $n_{\mathrm{ROLLERS}}$ is number of rollers in the bearing, $d_{\mathrm{ROLLER}}$ is the roller diameter, $d_{\mathrm{PITCH}}$ is the pitch diameter for the rollers and $\beta$ is angle of contact

For the outer race failure induced vibration, the frequency is:

$$f_{\mathrm{OUT}} = f_{\mathrm{ROT}} \frac{n_{\mathrm{ROLLERS}}}{2}\left[1-\left(\frac{d_{\mathrm{ROLLER}}}{d_{\mathrm{PITCH}}}\right)\cos\beta\right], \tag{6}$$

and for the roller failure:

$$f_{\mathrm{ROLLER}} = f_{\mathrm{ROT}}\left(\frac{d_{\mathrm{PITCH}}}{d_{\mathrm{ROLLER}}}\right)\left[1-\left(\frac{d_{\mathrm{ROLLER}}}{d_{\mathrm{PITCH}}}\right)^2\cos^2\beta\right]. \tag{7}$$

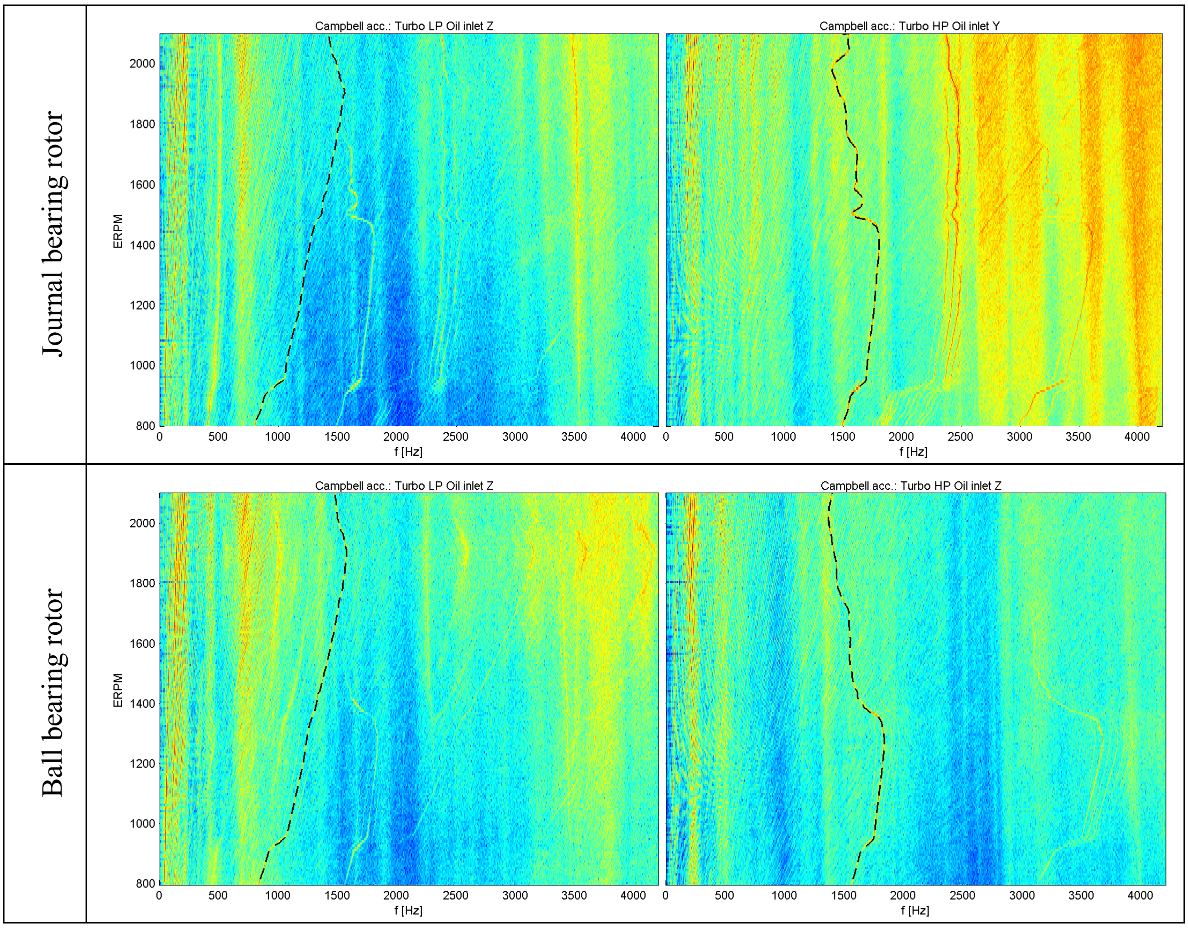



Figure 4: Samples of the turbo speed tracking for the test start (left) and end of test (right). Red star denotes the identified turbo speed peak.

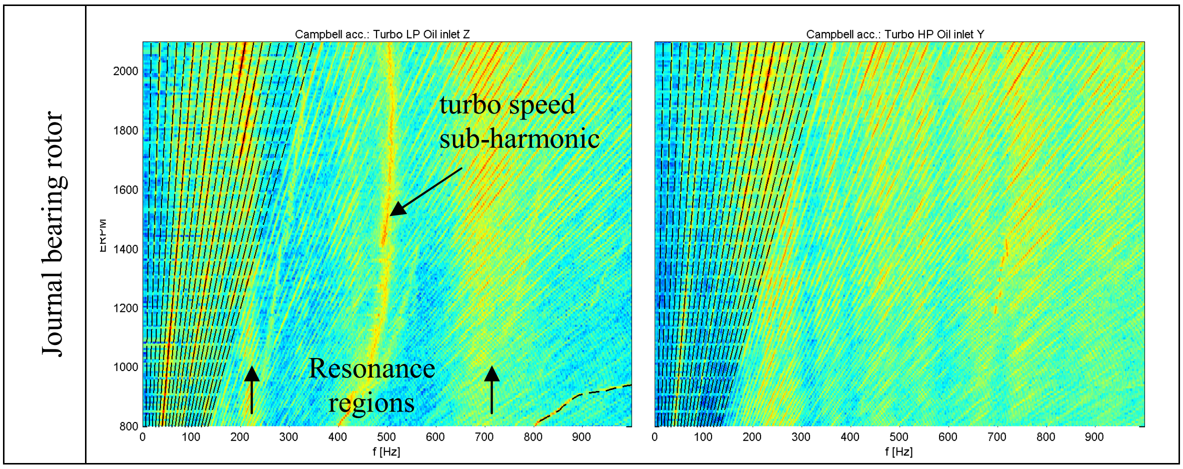



Figure 5: Zoomed Campbell plot with $f_{max}$ 1000 Hz. Dashed lines are the indicated engine harmonics starting with ½th and growing by step of ½ to 20$^{th}$. Resonance regions are represented as vertical strips.

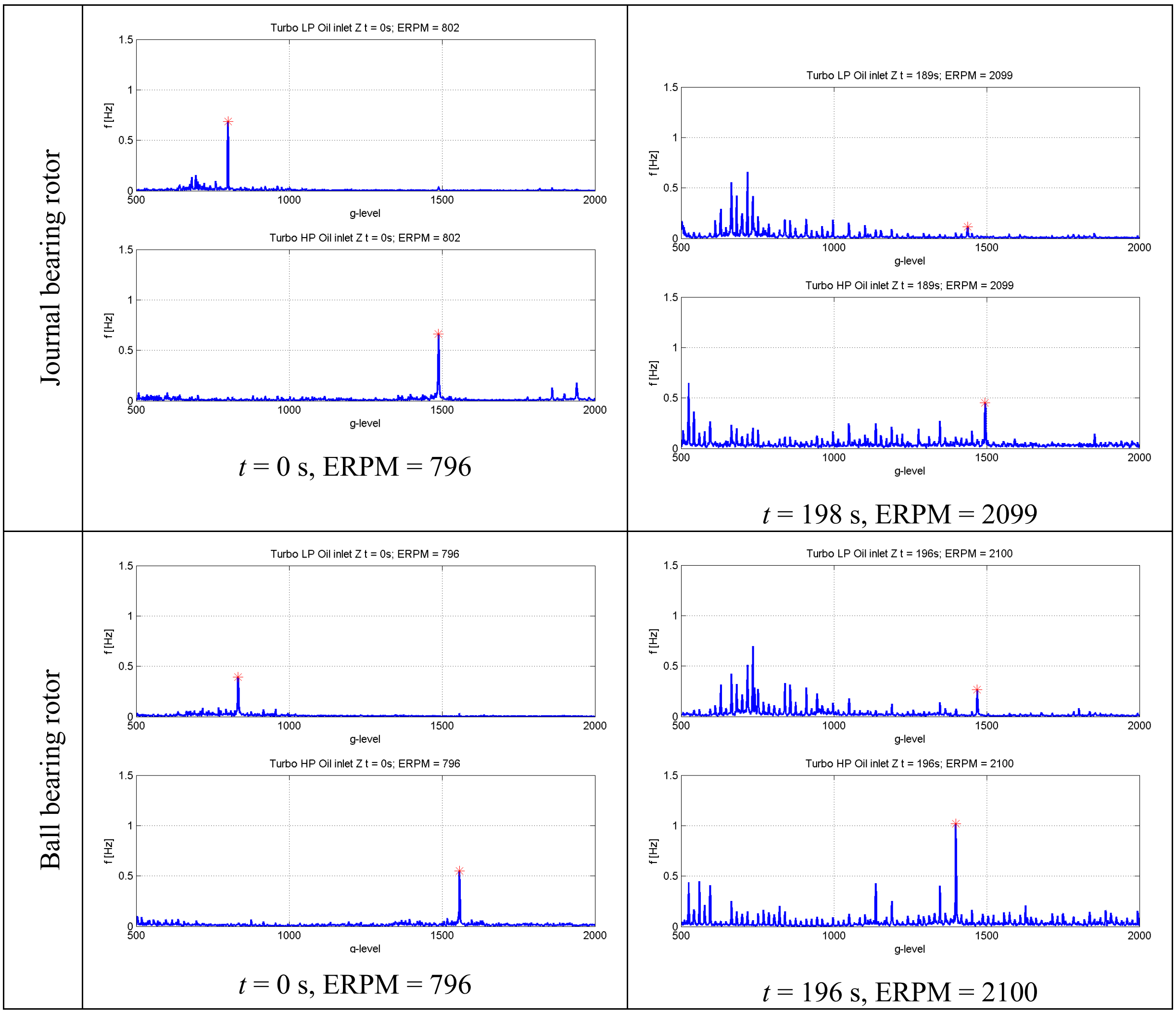



Figure 6: Samples of the turbo speed tracking for the test start (left) and end of test (right). Red star denotes the identified turbo speed peak.

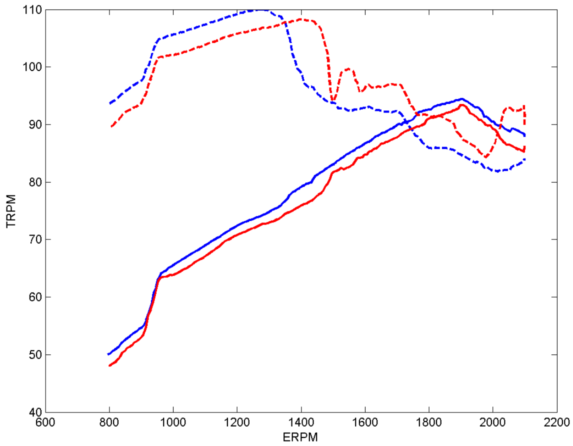



Figure 7: Tracked turbo speed vs. recorded engine speed for: dashed: HP stage; solid: LP stage; red: Journal Bearing application; blue: ball bearing application

## 5 Summary

The paper presents vibration data post-processing approach used in HTT Brno and that was build using Matlab scripts. The requirements for the analysis outputs are many times unique for given case and the purpose: structural qualification, fatigue damage investigation, electronic equipment validation etc. The program offers great option of making the customer specific changes in very short time and to provide clear and precise results. Procedure was illustrated on the example of the actual measurement on heavy-duty application which was the 6-cylinder in line diesel engine with the two stage sequential turbocharging. Comparison between option with journal bearing and ball bearing rotor groups was shown pointing the common features and differences. Approach of the extraction of the turbo speed from the acceleration data was also illustrated on the same example.

## References

[1] Anon. *Speed sensors for VG-Turbochargers and Turbo Brake.* Application Note, Jaquet AG, www.jaquet.com

[2] N. Aretakis, K. Mathioudakis, M. Kefalakis, K. Papailiou. *Turbocharger Unstable Operation Diagnosis Using Vibroacoustic Measurements*. Journal of Engineering for Gas Turbines and Power, Vol. 126, 2004

[3] M. Amarnath, R. Shrinidhi, A. Ramachandra, S. B. Kandagal. *Prediction of Defects in Antifriction Bearings using Vibration Signal Analysis*. IE(I) Journal, Vol 85, July 2004