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VIBROACOUSTIC DIAGNOSTICS OF DEFECTS OF COMBUSTION ENGINES VALVES

Key words: internal combustion engine, SI engine, CI engine, vibroacoustic signal, diagnostic symptom, damage.

Abstract: The problem of the diagnostics of mechanical defects of combustion engines, especially of exhaust valves damages, occurs in several papers. Authors of these papers apply various inference techniques and limit investigations to the one selected object under specific work conditions. The results of the active diagnostics experiment, in which it was shown that completely different signal processing algorithms allow obtaining reproducible diagnostics symptoms in SI (spark-ignition) and CI (compression-ignition) engines, when the threshold values are properly chosen, are presented in the paper.

Diagnostyka wibroakustyczna uszkodzeń zaworów samochodowych silników spalinowych

Słowa kluczowe: silnik spalinowy, silnik ZI, silnik ZS, sygnał wibroakustyczny, symptom diagnostyczny, uszkodzenie.

Streszczenie: W wielu publikacjach pojawia się problem diagnozowania uszkodzeń mechanicznych silników spalinowych, a w szczególności problem diagnozowania uszkodzeń zaworów wylotowych. Autorzy tych prac stosują różne techniki wnioskowania i ograniczają badania wyłącznie do jednego wybranego obiektu w specyficznych warunkach pracy. W artykule opisano rezultaty czynnego eksperymentu diagnostycznego, w którym wykazano, że zupełnie różne algorytmy przetwarzania sygnałów pozwalają na uzyskanie powtarzalnych symptomów diagnostycznych zarówno w przypadku silników ZS, jak i ZI, pod warunkiem właściwego doboru wartości progowych.

Introduction

Car engines equipped with on board diagnostics (OBD) systems and controlled by microprocessors are able, at present, to perform self-diagnostics in a very wide range. However, there is still a large group of mechanical defects which can remain undetected. The mechanism of the system operation is such that none of the observed parameters is a direct measurement of the possible mechanical defect. The diagnostics system will be activated when the selected work parameter exceeds its threshold value and either will indicate the failure or will make further work impossible, often without a possibility of the detailed diagnostics. Moreover, the OBD system operates realising the optimal work criterion, understood as minimizing contaminations caused by an improper combustion process. Thus, when the mechanical defect (leaking valve, pierced gasket, etc.) in its initial stage causes worsening of parameters of the engine work (e.g., a small pressure change in a cylinder), it can happen that the system will change the control parameters to improve the combustion quality, which mask this defect. The effect of masking mechanical defects by control systems was investigated by the authors [1]. Thus, the simple conclusion is such that, when using values recorded by the standard control and diagnostics system of a modern vehicle, the diagnostics of several mechanical defects is not possible. The problem, whether, in the age of the developed technologies and due to that a high reliability of engines, such defects are occurring and are dangerous for the safe usage remains open.

Based on the data obtained from the service station, several typical mechanical defects, including leaking exhaust valves, control system reactions, and visible external symptoms that do not allow detailed diagnostics are presented in paper [2] in the tabular formulation. The frequency of occurring individual defects and economic calculations determine the profitability of developing diagnostics systems. However, it should be taken into account that recently costs of even highly developed systems of numerical processing abruptly decreased and are still decreasing, while defects of valves, gaskets, ring-shaped pistons, and supply systems occur all the time. Thus, the problem of the diagnostics of these types of defects have become common enough to be discussed in several scientific papers. Among others, such works concern diagnostics of valve tightness on the bases of symptoms occurring when the observed signal of mechanical vibrations is processed. The superficial analysis of the latest papers [3, 4–8, 9–12] already allows one to draw conclusions that, on the one hand, the vibration signal contains information concerning the states of valves, which means that building of such system is possible, and, on the other hand, each author performed a different experiment and revealed the possible dependence using completely different methods and techniques of signal processing. The visible differentiation of SI and CI engines, for which investigations were performed under different conditions, is also seen.

Analysing achievements of this field, the question can be asked whether the algorithm proposed in one case would be the proper one in another case. Is there the need of differentiation the signal analysis methods when performing diagnostics of SI and CI engines? How can one efficiently build a diagnostic system in the given cases? To be able to answer the above questions, it was decided to compare two logically different algorithms and to verify their operations by artificially introducing two analogous defects of exhaust valves and performing observations – under on-board conditions – by means of the same recording and signal processing system.

1. Theoretical considerations

The fact that the valve defect should be visible as a change of the observed vibration process will be shown on the relatively simple example of the dynamic model. Let us consider the multi-cylinder system of the crank-piston mechanism in which we will treat masses of the connecting-rod piston and crank as concentrated masses, imposed to the crank centre and the crank shaft as an elastic element [6]. Limiting considerations to the simplest two-cylinder system, it means two cranks, and omitting damping, we will obtain the following:

$$M\ddot{x} + Kx = P(t)$$

$$M \cdot \ddot{x} = \begin{bmatrix} m_w & 0 & 0 & 0 & 0 & 0 \\ 0 & m_w & 0 & 0 & 0 & 0 \\ 0 & 0 & I_{kL} + m_w R^2 & m_w R & m_w R & 0 \\ 0 & 0 & m_w R & m_w & 0 & 0 \\ 0 & 0 & m_w R & 0 & m_w & 0 \\ 0 & 0 & 0 & 0 & 0 & I_{kP} \end{bmatrix} \cdot \begin{bmatrix} \ddot{u}_{n1} \\ \ddot{u}_{n2} \\ \ddot{\phi} \\ \ddot{u}_{\tau1} \\ \ddot{u}_{\tau2} \\ \ddot{\varphi} \end{bmatrix}$$

$$K \cdot x = \begin{bmatrix} k_{n1n1} & -k_{n1n2} & 0 & 0 & 0 & 0 \\ -k_{n1n2} & k_{n2n2} & 0 & 0 & 0 & 0 \\ 0 & 0 & k_{\theta\theta} & -k_{\tau1\theta} & -k_{\tau2\theta} & -k_{\theta\theta} \\ 0 & 0 & -k_{\tau1\theta} & k_{\tau1\tau1} & -k_{\tau1\tau2} & k_{\tau1\theta} \\ 0 & 0 & -k_{\tau2\theta} & -k_{\tau1\tau2} & k_{\tau2\tau2} & k_{\tau2\theta} \\ 0 & 0 & -k_{\theta\theta} & k_{\tau1\theta} & k_{\tau2\theta} & k_{\theta\theta} \end{bmatrix} \cdot \begin{bmatrix} u_{n1} \\ u_{n2} \\ \phi \\ u_{\tau1} \\ u_{\tau2} \\ \varphi \end{bmatrix} \tag{1}$$

$$P(t) = \begin{bmatrix} P_{n1}(t) \\ P_{n2}(t) \\ M_o(t) \\ P_{\tau1}(t) \\ P_{\tau2}(t) \\ 0 \end{bmatrix}$$

where

m_w – crank mass,
u_{ni} – radial strain of crankshaft,
$u_{\tau i}$ – tangent strain of crankshaft,
k_{njni} – reduced coefficient of flexural stiffness,
$k_{\theta\theta}$ – reduced coefficients of torsional stiffness,
$k_{\tau j\tau i}$, $k_{\tau j\theta}$ – reduced coefficient of flexural-torsional stiffness,
R – radius of crank,
I_{kL} – reduced moment of inertia of a flywheel and power transmission system,
I_{kP} – reduced moment of inertia of a power transmission system and engine fittings,
ϕ – angle of rotation of the left end of the crank shaft,
φ – angle of rotation of the right end of the crank shaft,
θ – relative angular displacement,
$P_{ni}(t)$ – forces of combustion gases in radial direction,
$P_{\tau i}(t)$ – forces of combustion gases in tangent direction,
$M_o(t)$ – anti-torque moment.

In this system, the forces of combustion gases are functions of the parameter of valve defect δ:

$$P_{ij}(t) = P_{ij}(t,\delta) \tag{2}$$

Relatively strong couplings (nonlinear) between generalised coordinates occurring in this system means that vibrations observed in an arbitrary direction will be dependent on gaseous force changes, thereby on the defect parameter. It means that the dynamic process observed on the head or engine block can play the role of the observed symptom. The question remains whether these changes are observable. The result of investigations showing the power change (not revealed by OBD) observed on the engine test bench for the engine with the efficient valve and the engine with the defected valve can constitute the answer (Fig. 1).

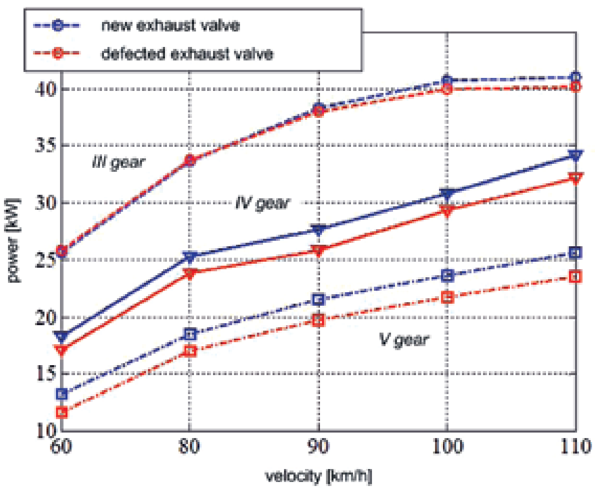

Fig. 1. Example of the power decrease caused by the exhaust valve defect, recorded on the roller dynamometer for various gears (SI engine)

Of course, the fact that the change of the force of combustion gases causes the change in the structure of the observed vibration signal does not mean that every algorithm will allow for the readable and repeatable diagnostics of the valve tightness degree. The appropriate analytical parameters should be selected and the following two conditions should be fulfilled:

1) Rotational speed should be synchronised;
2) Operators of the selection and averaging [13] in the time and frequency domains should be properly selected, which means that the preliminary processing of the signal should be performed according to the following relation:

$$x(\delta) = S_t S_\omega \left\{x(t,n,r,\Omega,\delta)\ldots\right\} \tag{3}$$

where

{x(…)} – observed signal, being generally the random process dependent on time t, location of measuring sensor r, machine specimen n, rotational speed Ω and defect parameter δ as well as other possible variables such as external disturbances, ambient temperature, etc., which should be eliminated or determined;

S_t – selection operator in the time domain containing the time window, the length of the observation time and sampling frequency;

S_ω – selection operator in the frequency domain (called filtration operator) primarily containing the selection of the observed band.

Since the diagnostics (regardless the algorithm) will be performed on the same vehicle and by the same recording system, the process dependence $\{x\}$ on variables n and r can be omitted.

Meeting the first condition is especially essential. Even when a driver (or a controller) maintains the constant rotational speed, this speed fluctuates intensely enough to "cover" efficiently small vibration changes accompanying the valve defect change. There are several ways of solving this problem. The authors apply the technique of signal resampling proposed in paper [14]. This technique operates according to the rule presented schematically in Figure 2. The technique idea is the

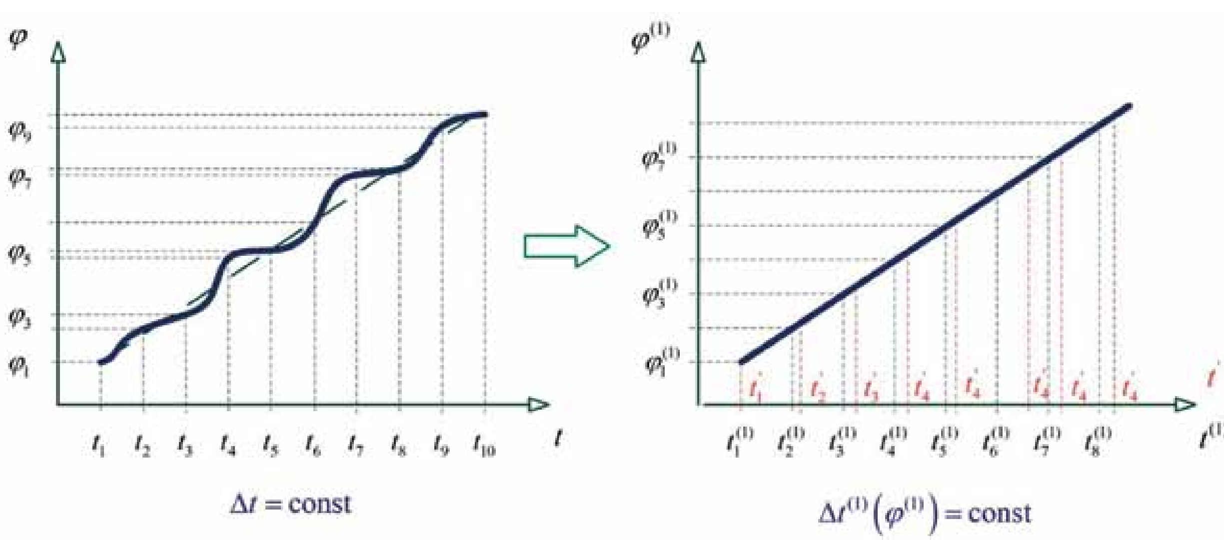

Fig. 2. Schematic presentation of the idea of the signal resampling

assumption that the constant time interval corresponds to the constant change of the angle of rotation, which requires satisfying the following dependence:

$$\Delta\varphi = 2\pi \frac{f_{obr}}{f_p} \tag{4}$$

where
f_p – frequency of signal sampling,
f_{obr} – assumed frequency value.

Thus, the presented technique reduces to the change of the sampling time step (generally the time scale) in such a way that

$$\frac{d\varphi}{dt} = \text{const}$$

The preliminarily processed signal can constitute the base of looking for the quantitative damage indicator, which, from the point of view of mathematics for the proposed algorithms, reduces itself to defining the proper metrics.

2. Experimental part

As the basis for investigations, the results of an experiment performed for SI and CI engines were assumed. Two classes of defects of the exhaust valve of the 1-st cylinder were introduced in these engines. Artificially introduced defects were made in such a way as to obtain their shape corresponding approximately to the real defect in the geometric sense as well as in the decrease of the compression pressure in the cylinder.

The defected valve is shown in Figure 3. The introduced defect caused the pressure decrease in the cylinder in accordance with Table 1.

Fig. 3. Defect of the exhaust valve

Table 1. Defect characteristic

	Change of the valve face surface	Decrease of the compression pressure in CI engine	Decrease of the compression pressure in SI engine
I-st valve defect	1.8%	8%	5.5%
II-nd valve defect	7.2%	17%	12%

The tested CI engine is a turbo-charged four-cylinder in-line engine with self-ignition with a displacement of 2.0 l., equipped with 2 camshafts. The power equals 66 kW, at 4000 rot/min, and maximum torque equals 245 Nm.

The tested SI engine is a four-cylinder in-line engine SPI 1.2 l with a power of 44 kW at 5500 rot/min and a maximum torque of 98 Nm.

Measurements were performed at constant rotational speeds of the engine in 2nd, 3rd, and 4th gears. The accelerations of vibrations in various directions and in various points of the engine head were measured using transducers B&K 4393. In addition, a signal from the camshaft position sensor was recorded in order to enable subsequent synchronization of the base rotational speed during the analysis of the signals.

The wide collection of the recorded courses was obtained. Some examples are shown in Figure 4.

3. Algorithms of drawing conclusions

Let us compare two algorithms of looking for the measurement of defects.

A) Algorithm based on combinations of simple signal measurements

For signals, which are represented in Figure 4, the sensitivity to the known simple measurements in time and frequency domains was checked.

The following measurements were analysed:
- average value – μ,
- variance – σ^2,
- asymmetry coefficient – WA,
- kurtosis – K,
- rms value – RMS,
- peak-to-peak value – $x_{p\text{-}p}$,
- peak factor – WS,
- clearance factor – WL,
- shape factor – WK,
- impulse factor – WI.

The signal was subjected to the preliminary processing. After resampling, taking into account the change of the engine rotational speed in time, averaging in the frequency domain was carried out. Then, the averaged signal was filtered in the selected band and the Hilbert transform was performed. This allowed determining the signal envelope as follows:

$$x_0(t,\delta) = \sqrt{\left(\hat{x}^2(t) + x^2(t)\right)} \tag{5}$$

where

$$\hat{x}(t,\delta) = \frac{1}{2\pi} \int_{-\infty}^{\infty} x(\tau) \frac{1}{t-\tau} d\tau \tag{6}$$

The basic measurements in the time domain were calculated for the processed signal. The sensitivity of these measurements to the analysed valve defect was checked. Examples of the investigation results are shown in Table 2.

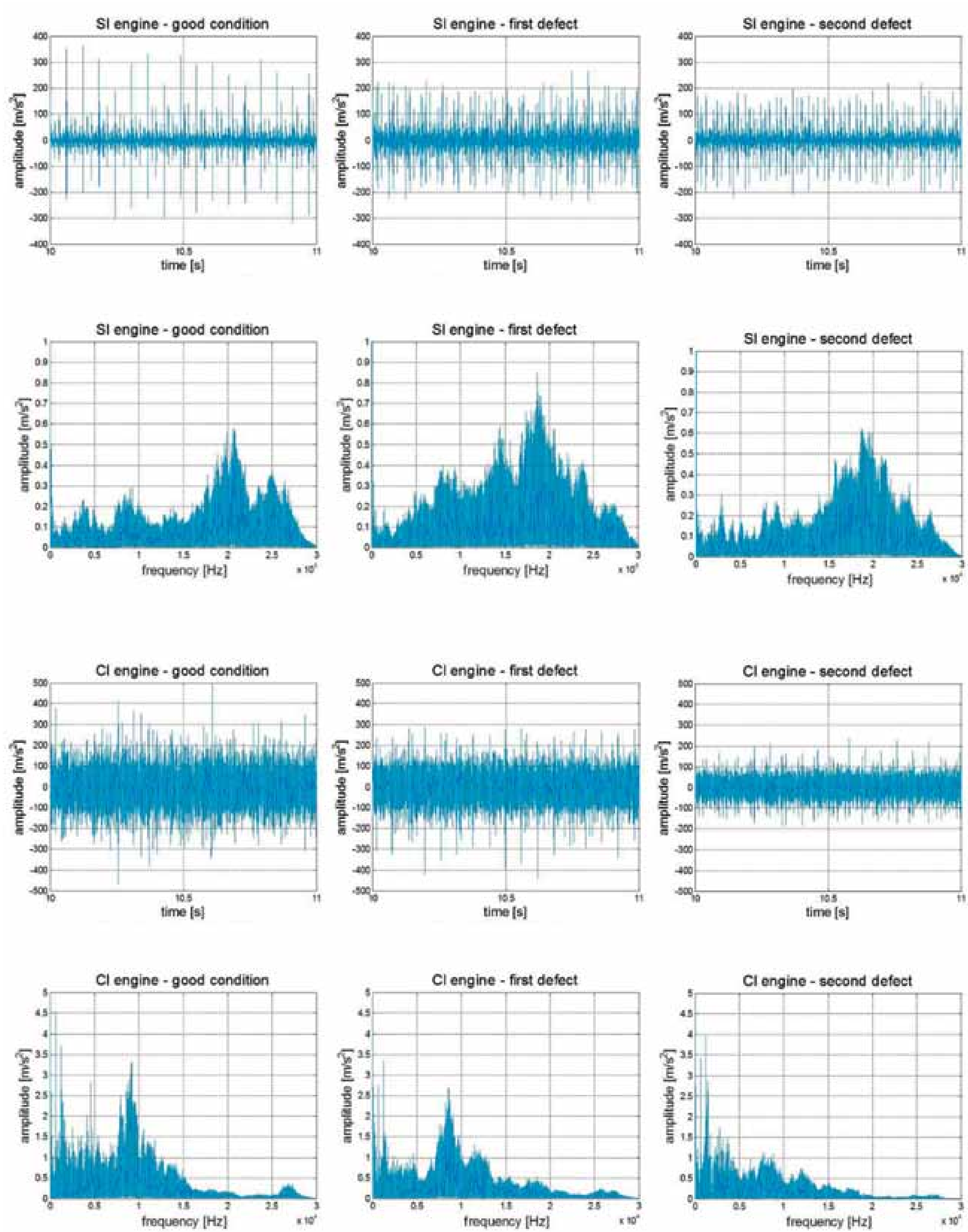

Fig. 4. Examples of preliminarily processed signals and their spectral densities for SI and CI engines in good condition and with various defects of the exhaust valve

Table 2. Sensitivity of simple measurements to valve defects

	S.I. engine				I.C. engine			
	I-st defect		II-nd defect		I-st defect		II-nd defect	
	III gear 2000 rot/min	IV gear 2000 rot/min	III gear 2000 rot/min	IV gear 2000 rot/min	III gear 2000 rot/min	IV gear 2000 rot/min	IV gear 2000 rot/min	IV gear 2000 rot/min
μ	→	→	→	→	↘	↘	↘	↘
σ	↘	↘	↘	↘	↘	↘	↘	↘
σ^2	↘	↘	↘	↘	↘	↘	↘	↘
WA	↗	↗	↘	↘	↗	→	↗	↗
K	↘	↘	↘	↘	↘	→	↘	↘
RMS	↗	↗	↗	↗	↘	↘	↘	↘
RMS^2	↗	↗	↗	↗	↘	↘	↘	↘
x_{p_p}	↗	↗	↗	↗	↘	↘	↘	↘
WS	↘	↘	↗	↗	↘	→	↘	↘
WL	↘	→	↘	↘	↘	→	↘	↘
WK	↘	→	↘	↘	↘	→	↘	↘
WI	↘	→	↘	↘	↘	→	↘	↘

The mentioned measurements were calculated for all selected rotational speeds of the engine in all gears. The simple variance measurement was selected for building the decision algorithm in consideration of its monotonic character in all cases.

B) Algorithm using the coherence analysis

The ordinary coherence function is the characteristic of single-input and single-output systems, having features of filtrations of input and output disturbances. This function achieves lower than unity values when the system is nonlinear. In order to apply this characteristic for solving the diagnostics task, a certain generalisation, utilising the fact that the observed process is, in fact, multidimensional, should be performed [15]. Thus, one can assume the observation result for the good condition as the conventional "input" into the system and the observation result for the defected state as the "output." Marking the following, respectively: G_{xx} – spectral density in the good condition, $G_{yy}^{(1)}$ – spectral density of the system with a small defect, $G_{yy}^{(2)}$ – spectral density of the system with a large defect and – as a formality – $G_{yy} = kG_{xx}$ – spectral density of a conventional "output". The value of the proportionality coefficient "k" for the good condition is meaningless, since γ^2 for the linear transformation remains equal to 1. Using $Z_{yy}^{(i)}$, we will mark the disturbance of the vibration process causing the change from the good condition to the state with the "i-th" defect. This assumption allows building the relation "input"↔"output" in accordance with Figure 5.

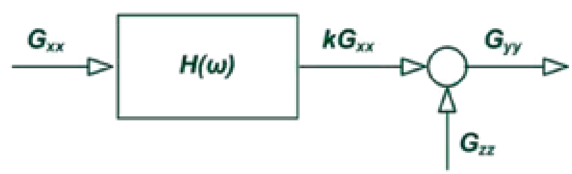

Fig. 5. Scheme of the coherence analysis of the system with an external disturbance

Applying dependences known from the coherence analysis, we will obtain the following:

$$H_{xy}^{(1)} = \frac{G_{yy}}{G_{yx}} = H\left(1 - \frac{G_{zz}}{G_{yy} - G_{zz}}\right) \tag{7}$$

$$H_{xy}^{(2)} = \frac{G_{xy}}{G_{xx}} = H \tag{8}$$

$$\gamma^2 = \frac{H_{xy}^{(1)}(f)}{H_{xy}^{(2)}(f)} \tag{9}$$

$$G_{zz} = \left(1 - \gamma^2\right) \cdot G_{yy} \tag{10}$$

Where the following markings are used:

G_{xx} – power spectral density of the input,
G_{yy} – power spectral density of the output,
G_{xy}, G_{yx} – cross power spectral densities from *x* to *y* and from *y* to *x*, respectively,
G_{zz} – power spectral density of the disturbance not correlated with the input ("good condition").

In this way, the part of the output signal, not coherent with the input, was separated. It means that, in accordance with the presented reasoning, the part of the signal, being the base for calculating the defect measurement, was separated. Examples of such signal processing are shown in Figure 6.

G_{zz} integral was assumed as the defect measurement in the obligatorily selected frequency interval

$$\mu = \left| \int_{\omega_1}^{\omega_2} G_{zz} d\omega \right| \tag{11}$$

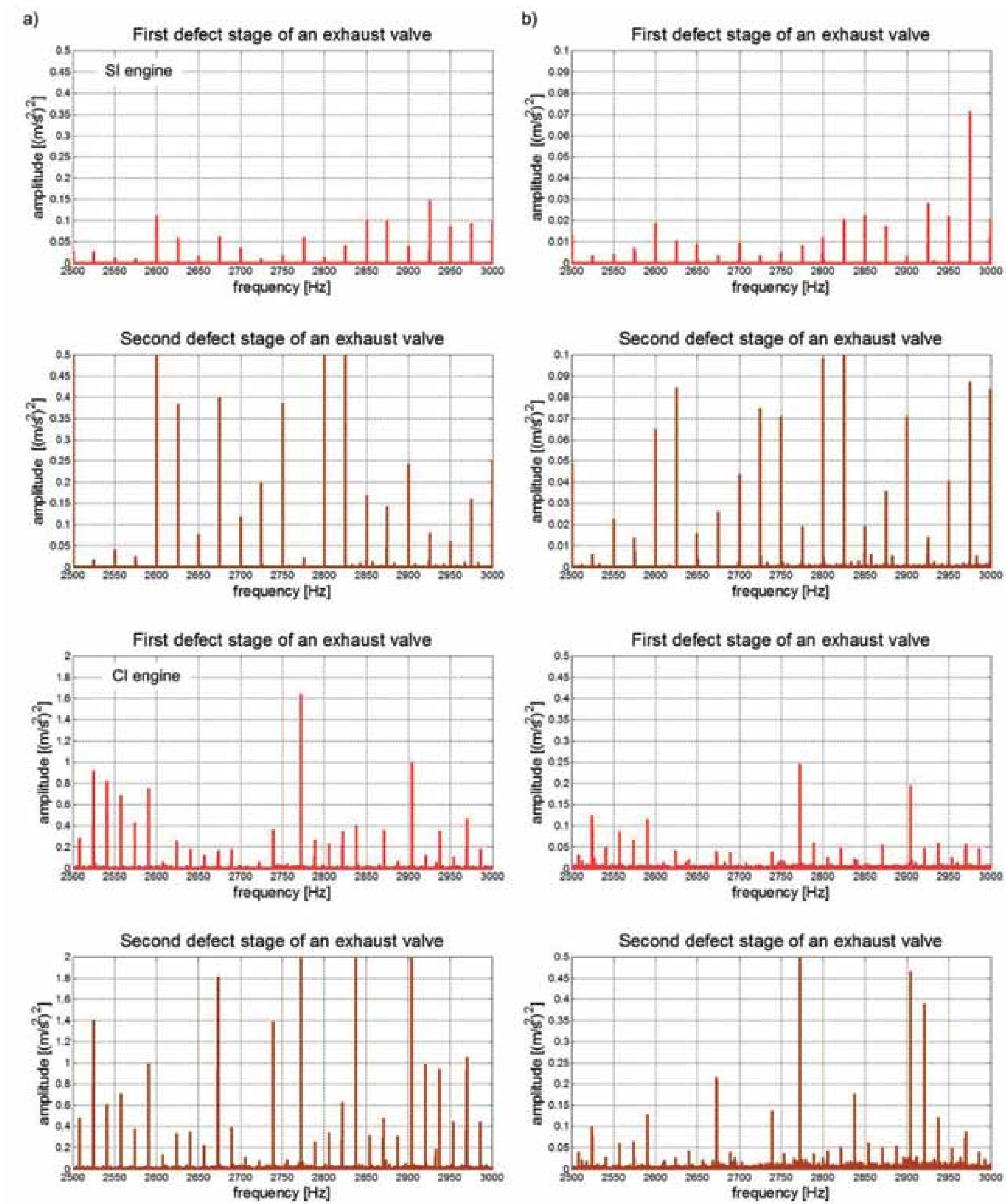

Fig. 6. Examples of the signal processing by the proposed procedure: a) the spectrum of the input signal, b) the spectrum of the incoherent signal with the input signal

Both applied algorithms are visually presented in Figure 7.

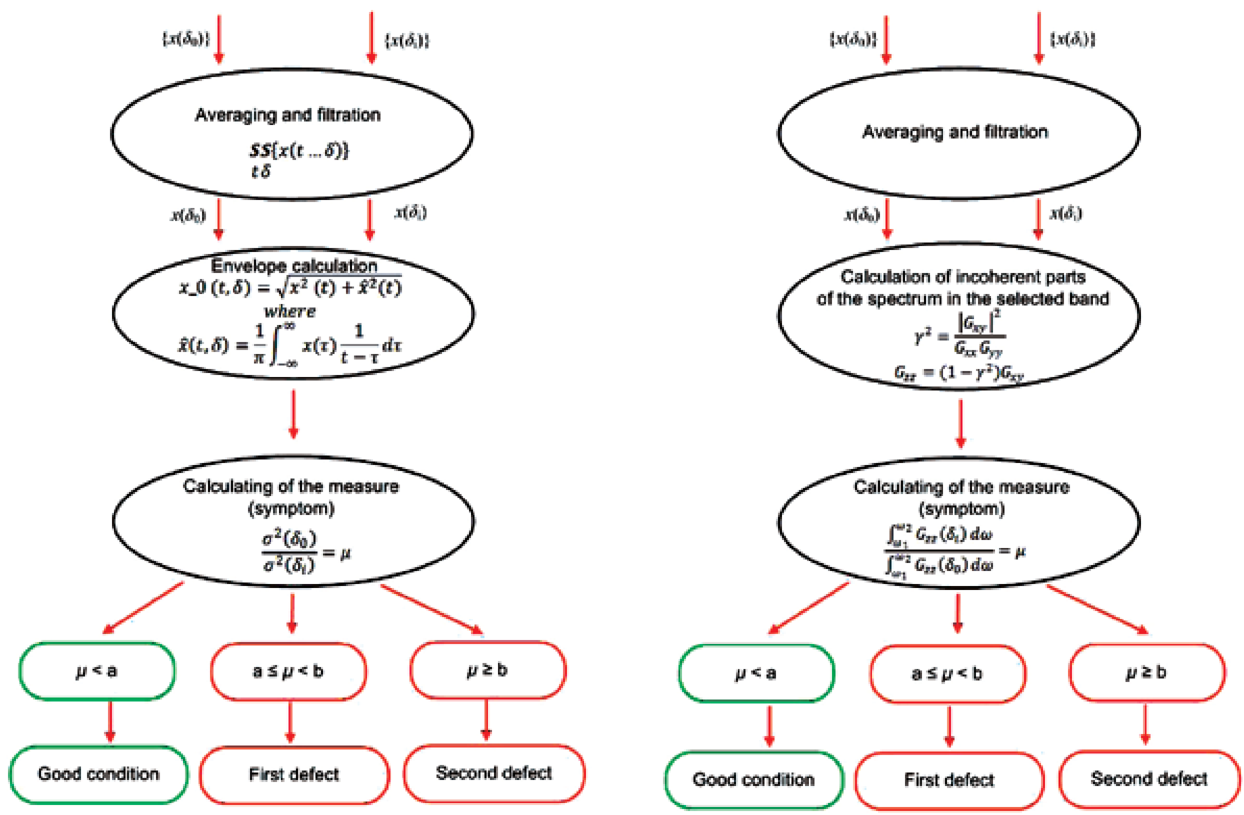

Fig. 7. Block diagrams of the applied algorithms

4. Final result

Applying the proposed algorithms, the attempt of diagnostics of both tested engines was undertaken under various work conditions (gear, rotational speed) and for differently selected frequency bands. The most reliable diagnostics was obtained for the following parameters:

- Compression-ignition engine: n = 2000 rot/min and 33<f<335 Hz for algorithm A
  n = 3000 rot/min and 2500<f<3000 Hz for algorithm B
- Spark-ignition engine: n = 2000 rot/min and 33<f<335 Hz for algorithm A
  n = 3000 rot/min and 2500<f<3000 Hz for algorithm B

The results are presented in Figure 8.

As can be seen for both applied algorithms, despite essential differences in their build, fully satisfying results were obtained. Thus, it seems that applying different measuring and signal processing techniques for SI and CI engines is pointless. Obtaining information on the valve state from the recorded vibration waveforms is possible by means of various techniques of signals processing, and the simplicity of the calculations should determine which technique should be applied. Direct applications of every proposed technique, in the "on-line" system as well as during measuring vibrations on the roller dynamometer, are possible. Only the threshold values should be matched to the mark and type of engine (vehicle). The precise choice of the selection operators is also essential, which, from the point of view of calculations, is relatively easy. Observations should be carried out for a relatively long duration (minimum 20 seconds), but maintaining the stable rotational speed during the measurement, on account of the applied resampling procedure to the same rotational speed, can occur with an accuracy of the vehicle counter.

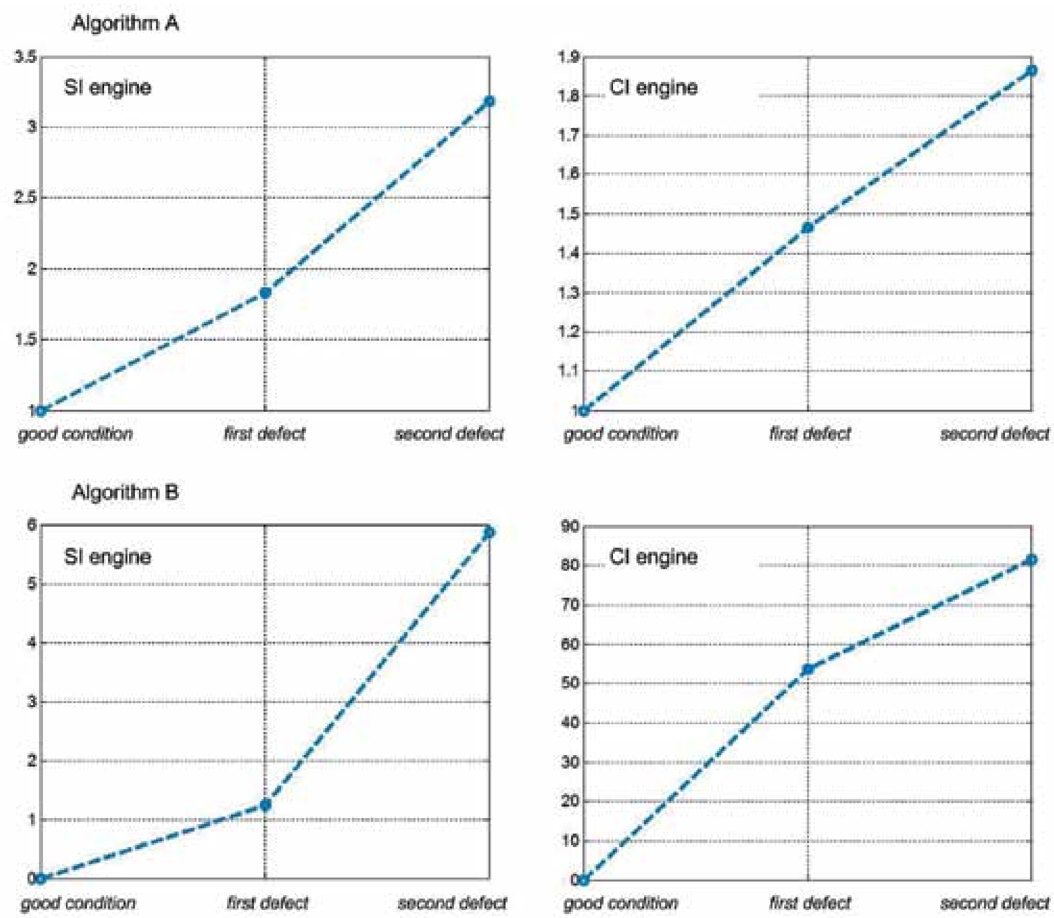

Fig. 8. Efficient diagnostics symptoms obtained by algorithm „A" and algorithm „B" for SI engine and CI engine

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