APPLICATION OF VIBROACOUSTIC SIGNAL TO DIAGNOSE DISCK BRAKING SYSTEM

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Abstract

Attempt to raise train speed involves application of greater braking power i.e. braking systems rapidly absorbing and dispersing stored heat energy. To maintain high efficiency of braking system in the whole operational process, it is necessary to control the friction set: brake and pad before reaching limitary wear particularly of friction pads. Stable and constant - in the whole speed range- coefficient of friction „μ" and realization of greater pad's pressures to the disc because of better conditions of warmth offtaking generated during braking into the athmoshere are basic advantages of disc brake systems [8]. Few disadvantages of disc brake include a lack of possibility of controlling the condition of the friction set: brake and pad in the whole operation time. It is particularly observable in rail cars, where disc brakes are mounted on the axle of the axle set between the wheels. To check the wear of friction pads and brake discs it is necessary to apply inspection channel to carry out inspections, and to carry out replacement of friction parts in case they reach their limitary wear. The purpose of this research is to apply vibration signal of pad calipers to assess the wear of friction pads of disc brake by defining characteristics of time, amplitude and selected frequency characteristics during tests at internal brake station.

Keywords: *disc brake, wear of friction pad, vibration signal, amplitude characteristics, frequency characteristics*

1. Introduction

Because of complex braking system in rail cars and locomotive, most often consisting of 8 individual brake cylinders, application of one diagnostic system to assess the wear of all friction sets is impeded [8]. A system for video inspection and diagnostics worked out in Rail Vehicle Institut „TABOR" in Poznań is the most advanced system do diagnose disc brake. Diagnosing system [1] provides complete information about the wear of friction pads and brake disc in each operation moment. Worked out solutions, becasue of complex and expensive measuring set consisting of a digital film camera and a software to convert the picture, after successful tests at reasearch station, have not been applied by railway industry yet.

In rail technique, also rail track stations are used to diagnose the wear of friction pad. At these stations friction set consisting of disc brake and friction pad is photograhed during train ride. However, it is not a very precise method because, on the basis of registered pictures the thickness of frction pads of disc brake is only assessed. When pads' thickness amounts to approx. 10mm tram driver receives information that limitary acceptable wear of pads on a certain axle of axle set has been reached [9]. Rail track stations to diagnose the wear of friction pads are used by German, British and French railways.

In railway vehicles, systems signaling braking process and easing process, visible for the service from the inside and outside of the vehicle, are the most often applied. Those systems enable to check during train ride in which car braking system is bloked. Nevertheless, rail technique lacks an objective method of quantitive assessment of the wear of friction pads.

The purpose of this research is to apply vibration signal of pad calipers to assess the wear of friction pads of disc brake.

2. Methodology research

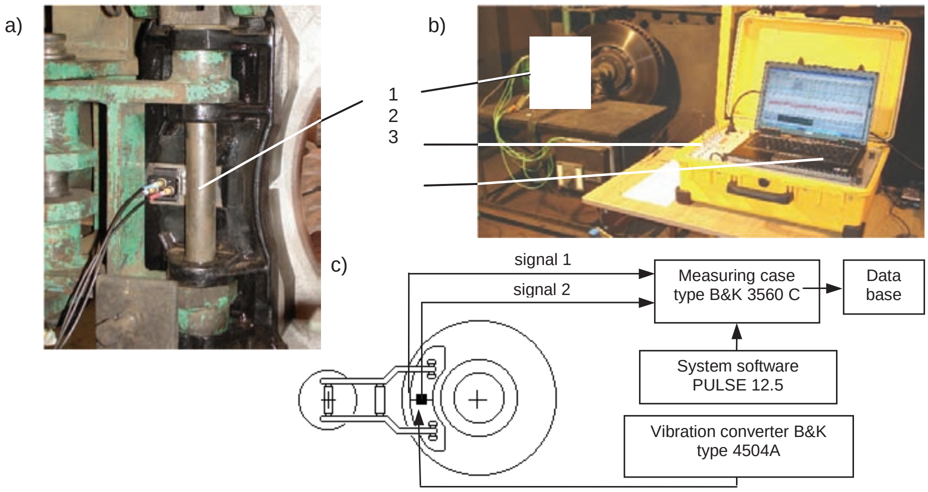

Fig. 1. Station for tests of railway brakes; a) pad calliper with accelerometer, b) view of measurement set of vibrations generated by calliper with pads, c) diagram of measurement track of vibrations generated by calliper with pads, 1-accelerometer, 2-measuring case type B&K 3560 C , 3- System software PULSE 12.5

The research was carried out at internal station for tests of railway brakes. A brake disc type 610×110 with ventilation vanes made by Kovis and three sets of pads type 200 FR20H.2 made by Frenoplast constitute the research object. One set was new - 35 mm thick and two sets were worn to thickness of 25 mm and 15 mm.

A reasearch program C (fast ride) according to instructions of UIC 541-3 was applied. The brakings were carried out from speed of 50, 80, 120, 160 and 200 km/h. During the research pad's pressures to disc N of 28 and 44kN were realized as well as braking masses per one disc of M=4.4T and 7.5T [7]. Vibration converters were mounted on pad calipers with a mounting clip, which is presented in Fig. 1a. During the research signals of vibration accelerations were registered in three reciprocally orthogonal directions. To acquire vibration signal a measuring system consisting of piezoelectric vibration accelerations converter and measuring case type B&K 3560 C with system software PULSE 12.5 was used. Fig. 1b presents the view of the measuring track.

Brüel&Kjær's vibration converters type 4504 were selected on the basis of instructions included in paper [3], the linear frequency of converter transit amounted to 13 kHz. During diagnostic tests signals in frequency from 0.7 Hz to 9 kHz [2] were registered. Sampling frequency was set at 32 kHz. This means that the frequency that was subject of the analysis in accordance with Nyquist relation amounted to 16 kHz.

This research was carried out in accordance with principles of active experiment. After carrying out a series of brakings at set speeds at the beginning of braking, pads' pressures to the disc and braking masses, the friction pads were changed and values of instantenuous vibration accelerations were registered.

3. Analysis of results of vibration accelerations by defining in time and amplitude domain

Research carried out at railway disc brake station showed that at each registration of signals of vibration acceleration in the time function it is possible to select from the whole registered signal a part referring only to braking process, independently of speed at the beginning of braking, pad's pressure to the disc and braking mass. Fig. 2 shows an exemplary signal of instantaneous values of vibration accelerations of caliper and pad registered in direction Y_1 (orthogonal to the disc) during station research.

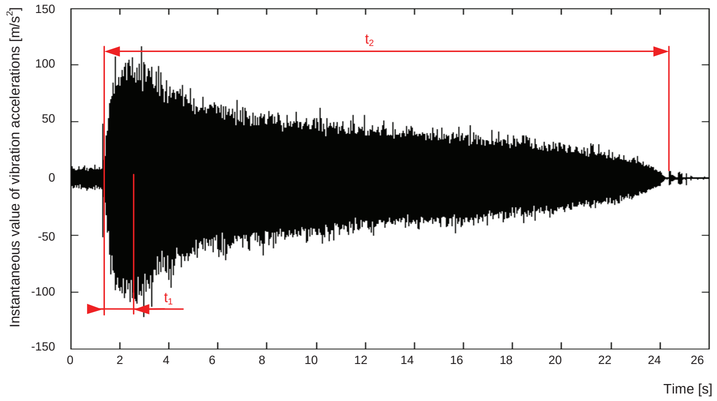

Fig. 2. Signal of vibration accelerations registered on pad calliper in direction Y_1: t_1- time of increase of pad's pressure to disc, t_2-time of braking

During simulated ride at speed *v* at brake station, the values of vibration accelerations of pad calipers put away from the disc by 2mm, come mainly from station vibrations driving the brake disc to required barking speed. Higher speeds before braking generate increase of caliper vibrations and for speed of 200km/h do not exceed 25m/s^2.

By registering continuously the value of vibration accelerations (ride at speed v, braking, and another ride at speed *v*) it is possible to recognize accurately the moment of the beginning of braking and its end. Before braking, on the diagram of vibration accelerations, an instantaneous increase of acceleration value (position 1 in Fig. 3a) stemming from operation of the lever set in contact of caliper and pad to the disc with increase of pressure in brake cylinder, is observed. A steep decrease of acceleration value determines disc stoppage (*v*=0); this is presented in position 2 in Fig. 3b. Instantaneous increase of vibration accelerations just after stoppage determined putting away the caliper from the disc with pressure decrease (position 3 in Fig. 3b).

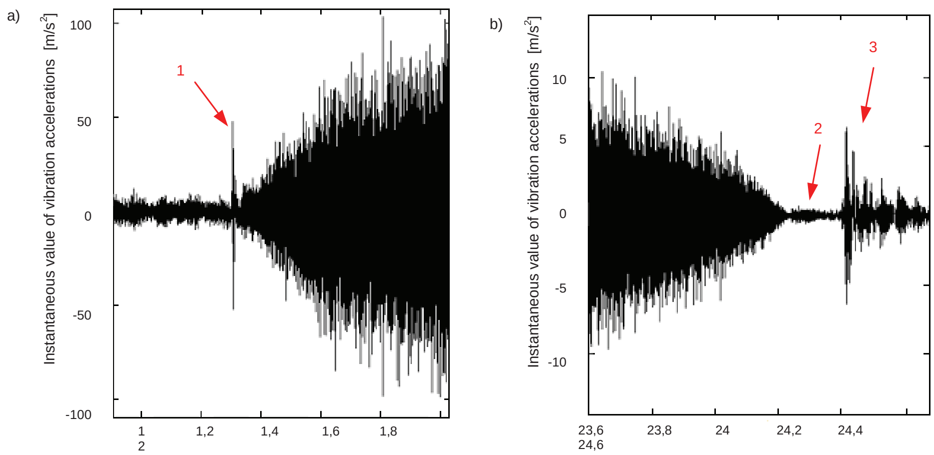

Fig. 3. Vibration accelerations of calliper with pads in various braking phases: a) beginning of braking, b) end of braking, 1- vibration signal impulse from calliper and pad contact to the disc, 2- end of braking, v=0, 3- vibration signal impulse of calliper and pad divergence from the disc

Further analysis in time domain enabled application of time flows of vibration accelerations of pad calipers to define braking time. Independently of pad's pressure to disc *N* and braking mass *M* and dependently on certain speeds at the beginning of braking *v*, it is possible to diagnose the wear of friction pads on the basis of registered signals of vibration accelerations. Carried out analysis had a quality assessing character of registered flows and it was carried out for three pads' thicknesses, i.e. G_1=35mm, G_2=25mm and G_3=15mm.

After carrying out time analysis of caliper and pad vibrations, it was found out that increase of vibration accelerations may be affected by the effect of self-excitation vibrations, which was observed in the pad worn to G_3=15mm. Fig. 4 presents the flow of the values of vibration accelerations in the braking time function for pad G_1 and pad G_3 with visible self-excitation vibrations in the last phase of braking (speed of 20km/h).

The effect of self-excitation vibrations observed for pad G_3 may testify for instability of the lever set of disc brake and may influence its future damages, such as cracks of calipers during operation time, which was described by Mr. Gruszewski in [5]. Self-excitation vibrations may be generated by resonance of caliper and pad set.

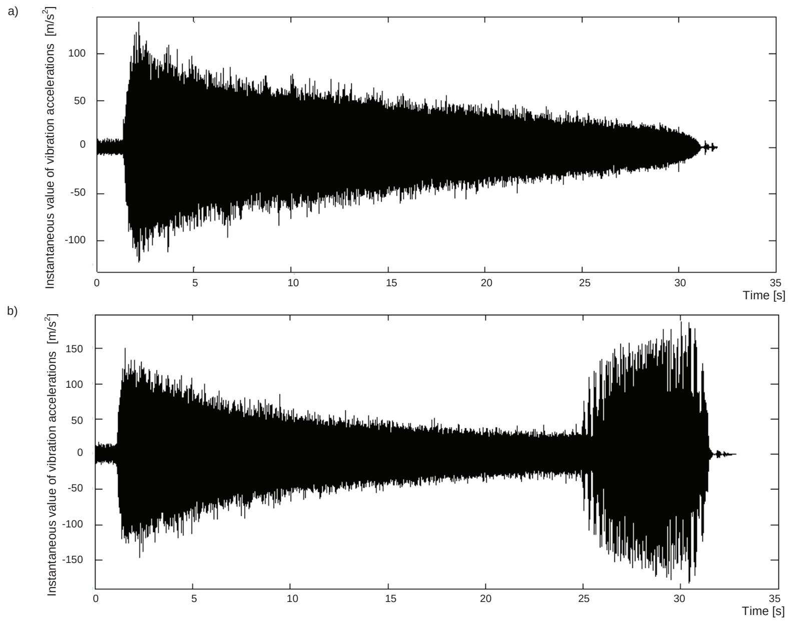

Fig. 4. Course of instantaneous vibration accelerations value of calliper with pads at braking speed of 160km/h: a) pad vibration G1=35mm b) pad vibration G3=15mm with self-excitation in last braking

In domain of amplitudes, the most common are the point parameters [4], which are used to describe displacement signals, speed signals and signals of vibration accelerations. Characterizing vibration signal with one number is an advantage of point parameters, thanks to which it is easy to define changes in vibroacoustic signal resulting from changes in technical condition of the tested object.

To diagnose the wear of friction pads of railway brake the following dimensional point parameters are applied:

- average amplitude, described with dependence:

$$S_{AVERAGE} = \frac{1}{T}\int_0^T |s(t)| dt \,, \tag{1}$$

where:
T - average time [s],
s(t) - instantaneous value of vibration accelerations [m/s^2].

- RMS amplitude, described with equation:

$$S_{RMS} = \sqrt{\frac{1}{T}\int_0^T [s(t)]^2 dt} \,, \tag{2}$$

- square amplitude, describe with dependence:

$$S_{SQuARE} = \left[\frac{1}{T}\int_0^T |s(t)|^{\frac{1}{2}} dt\right]^2 \,, \tag{3}$$

- peak amplitude, described with equation:

$$S_{PEAK} = \left[\frac{1}{T} \int_0^T |s(t)|^\infty dt \right]^{\frac{1}{n}} \quad dla \quad n \to \infty , \tag{4}$$

Before calculating point parameters from signals of vibration accelerations in program Matlab 7.0, a preliminary processing of signal in time domain was carried out. The reason of this processing was to select from the whole registered signal a part connected only with braking process. This process was also carried out to obtain required dynamics of changes essential for diagnostic purposes. Defining dependence of friction pad's thickness on selected point parameters was carried out through determining dynamics of changes for a certain parameter, which is presented in dependence (5) [6]:

$$D = 20\lg\left(\frac{s_2}{s_1}\right) , \tag{5}$$

where:
s_1 - the value of point parameter determined for pad G_3 or G_2 [m/s^2],
s_2 - the value of point parameter determined for pad G_1 [m/s^2].

The analysis of results of vibration tests showed that obtaining dependence of friction pads' thickness on the value of point parameters is possible by measuring vibration in directions Y_2 and Z_2 on a sensor mounted from the side of brake cylinder's case. Diagnostic tests with application of point parameters showed that inference about the wear of friction pads is dependant on type of braking with pressure *N* to the disc and on braking mass *M*. Realizing pressure N=44kN on the disk with M=4.4T enables to determine dependence of the wear of friction pads on the value of point parameters in the whole speed range at the beginning of braking i.e. from 50 to 200km/h. Moreover vibration tests showed that combinations of brakings with N=44kN and M=7.5T, N=28kN and M=4.4T, N=28kN and M=7.5T preclude assessment of the wear of pad on the basis of values of point parameters for considered speeds at the beginning of braking.

The greatest values of dynamics of changes were noticed by using the following point parameters: the RMS value and square value. Fig. 5 present dependence of (RMS) value of vibration accelerations in direction Z_2 on braking speed for various values of the wear of pad *G* with N=44kN and M=4.4T.

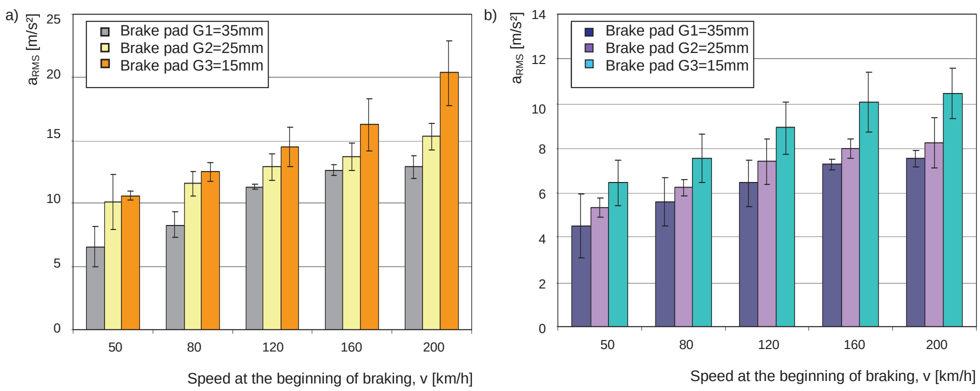

Fig. 5. Dependence of RMS value of vibration accelerations on braking speed for various values of the wear of pad G with N=44kN and M=4.4T: a) measurement in direction Y2, b) measurement in direction Z2

Because of increasing dependences of selected point parameters in the speed function at the beginning of braking, which was found out for each tested friction set, in the further analysis calculated point parameters were approximated against three friction pads' thicknesses. Dependences were approximated with the polynomial function or the linear function (depending on the speed at the beginning of braking) receiving the least values of correlation coefficient *R*. As a result of approximation mathematical models were obtained, which enable to calculate value of selected point parameters in the function of friction pad's thickness. Approximation was carried out in the measurement's orthogonal direction to the friction surface of the disc (Y_2) and tangential direction (Z_2) for RMS value, average value and square value, which resulted from the greatest dynamics of changes in enumerated point parameters.

To determine RMS value of vibration accelerations in direction Y_2, it is possible to use the following equations:

$$a_{RMS,(v=50)} = -0.2035 \cdot G_{(1,2,3)} + 13.823, \tag{6}$$

$$a_{RMS,(v=80)} = -0.2108 \cdot G_{(1,2,3)} + 16.074, \tag{7}$$

$$a_{RMS,(v=120)} = 31.64 \cdot G_{(1,2,3)}^{-0.2857}, \tag{8}$$

$$a_{RMS,(v=160)} = 36.326 \cdot G_{(1,2,3)}^{-0.2991}, \tag{9}$$

$$a_{RMS,(v=200)} = 87.348 \cdot G_{(1,2,3)}^{-0.5389}, \tag{10}$$

where:
$a_{RMS(\ldots)}$ - RMS value of vibration accelerations in direction Y_2 [m/s^2],
v - speed at the beginning of braking [km/h].

In case of measurement of instantaneous values of vibration accelerations in direction Y_2, it was found out that for lower speeds at the beginning of braking (to 80km/h), approximation of point parameters can be effected with linear functions, which was confirmed for each calculated parameter i.e. for peak value, RMS value, average value, square value and kurtosis. Higher braking speeds cause that the value of point parameters should be approximated with the polynomial function. In direction Z_2 of measurement of vibration accelerations of calipers with pads, average value should be approximated with the polynomial function in the whole range of speeds at the beginning of braking.

During station research, dynamics of changes of analyzed values of point parameters according to dependence (5) was defined, which is presented in tab. 1. On this basis it was found out that RMS value of vibration accelerations shows the best sensitivity towards change of pad's thickness against other point parameters at vibration measurement in directions Y_2 and Z_2.

Tab. 1. Dynamics of changes of selected point parameters in direction Y_2 and Z_2

Point parameter	Measurement's direction Y_2	Measurement's direction Z_2
RMS value	4.8	5.3
Average value	4.3	4.5
Square value	4.1	5.1
Peak value	3.9	4.4

Fundamental aim of station research of diagnostic character is to determine the wear of friction pads on the basis of values of vibration accelerations by applying approximating functions, on the basis of which, measured value of point parameter enables to define the wear of brake's friction pad. The wear of pads determinates pads' thickness, which in the carried out tests were diversified. Calculations were carried out for RMS value obtained in measurement direction Y_2 and for RMS value in direction Z_2.. For RMS value of point parameter, also obtained from measurement in direction Y_2 and Z_2 by using linear approximating functions described with dependences (11-20) for five speeds at the beginning of braking, the following equations defining friction pads' thickness were introduced:

$$G_{(Y2,\,v=50)} = -4.8508 \cdot a_{RMS,(v=50)} + 67.368, \tag{11}$$

$$G_{(Y2,\,v=80)} = -4.2915 \cdot a_{RMS,(v=80)} + 71.365, \tag{12}$$

$$G_{(Y2,\,v=120)} = -6.3257 \cdot a_{RMS,(v=120)} + 106.55, \tag{13}$$

$$G_{(Y2,\,v=160)} = -5.246 \cdot a_{RMS,(v=160)} + 99.461, \tag{14}$$

$$G_{(Y2,\,v=200)} = -2.5832 \cdot a_{RMS,(v=200)} + 66.829, \tag{15}$$

$$G_{(Z2,\,v=50)} = -10.312 \cdot a_{RMS,(v=50)} + 80.974, \tag{16}$$

$$G_{(Z2,\,v=80)} = -9.9328 \cdot a_{RMS,(v=80)} + 89.157, \tag{17}$$

$$G_{(Z2,\,v=120)} = -8.0383 \cdot a_{RMS,(v=120)} + 85.929, \tag{18}$$

$$G_{(Z2,\,v=160)} = -6.68 \cdot a_{RMS,(v=160)} + 89.416, \tag{19}$$

$$G_{(Z2,\,v=200)} = -6.3409 \cdot a_{RMS,(v=200)} + 80.448, \tag{20}$$

where:
$G_{(\dots)}$ - pad's thickness calculated on the basis of RMS value of vibration accelerations a_{RMS} [mm].

The value of correlation coefficient between measurement results of pad's thickness and the value obtained during calculations using dependence (11-20) during verifying tests on pads of thickness G_4=21.1mm and G_5=16.2mm amounted to 0.91-0.98 depending on braking speed.

Figure 6 presents dependence of friction pad's thickness of disc brake *G* on RMS value of vibration accelerations a_{RMS}.

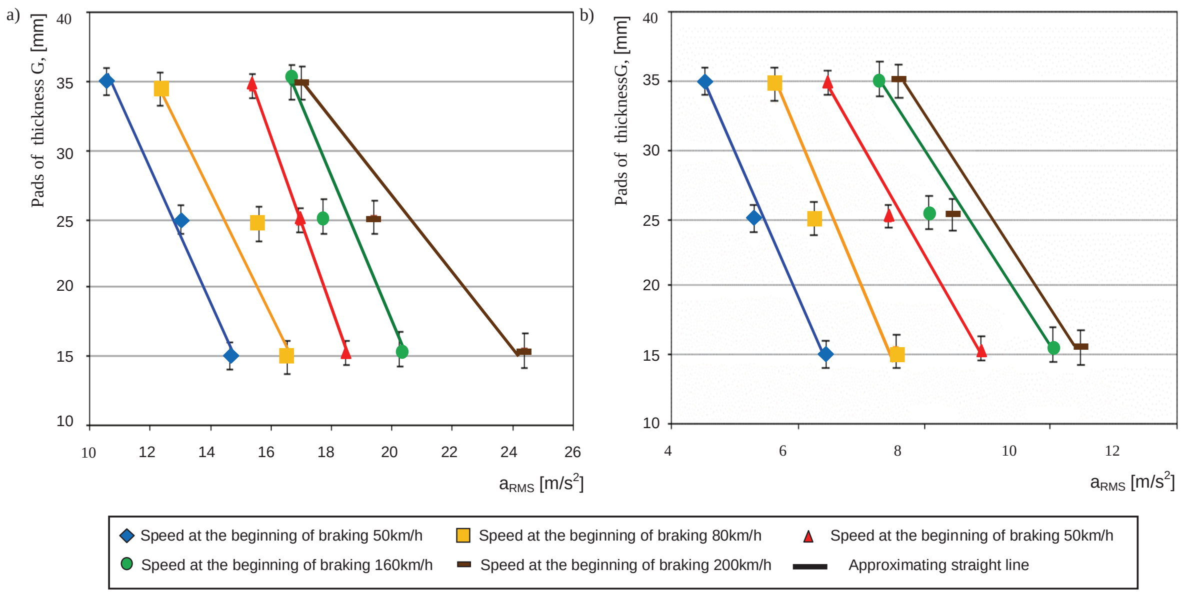

Fig. 6. Dependence of pad's thickness G on RMS value of vibration accelerations with N=44kN and M=4.4T: a) measurement in direction Y2, b) measurement in direction Z2

4. Analysis of results of vibration accelerations by defining in frequency domain

The purpose of spectrum analysis of signals of vibration accelerations was to determine frequency bands connected with change of pad's thickness during operation of braking system. Figure 7 presents exemplary amplitude spectra of vibration accelerations for various pad's thicknesses received during braking from speed of 160 km/h. Spectrum received on measurement of vibrations in direction perpendicular to friction surface of the disc (direction Y_2) with pad's clamp to the disc N=44kN and braking mass M=4.4t.

Research on measurement of vibration accelerations of brake callipers in frequency domain showed that it is possible to find frequency bands, in which dependence of RMS value of vibration accelerations a_{RMS} (equation (2)) on various pad's thicknesses in considered range of speeds at the beginning of braking is observed.

Table 2 presents frequency range, in which dependence of amplitude value of vibration accelerations on the wear of pads is observed. Additionally, dynamics of changes according to dependence (5) [6] of an examined diagnostic parameter for a certain frequency band and at a certain speed at the beginning of braking and values of correlation coefficients for linear dependence of amplitude value of vibration accelerations on examined friction pad's thicknesses is presented. On this basis it was concluded that diagnosing the wear of frictions pads can be carried out independently from the speed at the beginning of braking for certain frequency bands.

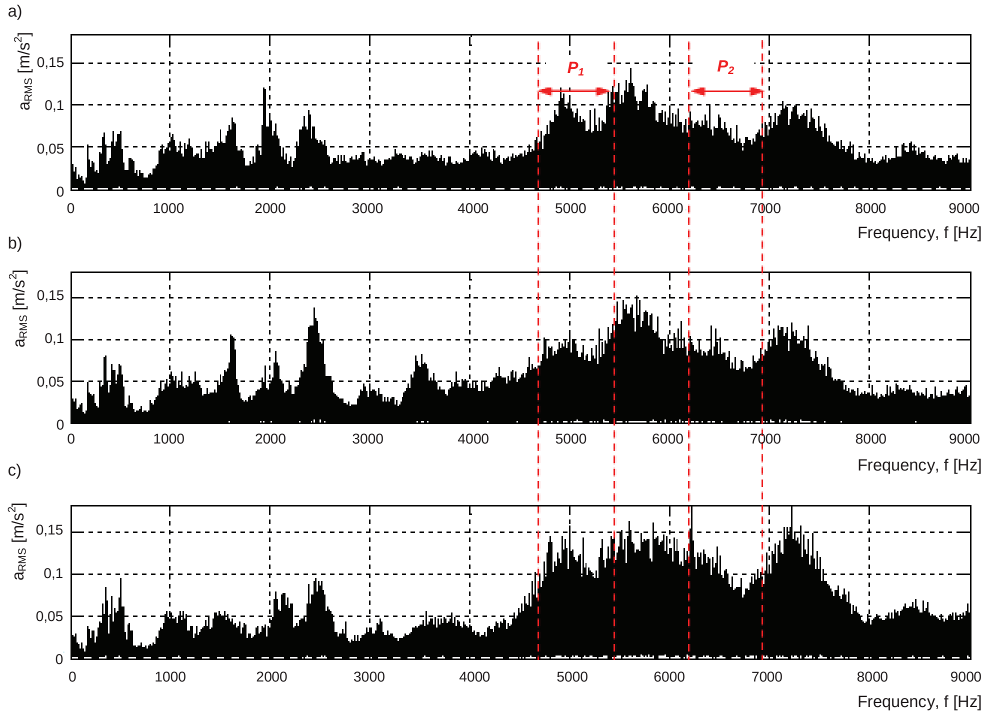

Fig. 7. Dependence of amplitude of vibration accelerations on frequencies for different pad's thicknesses for speed at the beginning of braking v=120km/h in direction Y2: P1-frequency band 4600-5100 Hz, P2-frequency band 6000-6700 Hz: a) pad's thickness G1=35mm, b) pad's thickness G2=25mm, c) pad's thickness G3=15mm

Tab. 2. Results from research on determining frequency bands N=44kN and braking mass M=4,4t, in which dependence of amplitude of vibration accelerations in function of friction pad's thickness occurs

Speed at the beginning of braking v [km/h]	Frequency band *f* [Hz]	Dynamics of changes *D* [dB]	Correlation coefficient *R*
50	4600-5100	6.26	0.89
80	4600-5100	5.83	0.99
120	4600-5100	2.89	0.93
160	4600-5100	2.68	0.94
200	4600-5100	7.61	0.99
50	6000-6700	5.79	0.99
80	6000-6700	4.75	0.97
120	6000-6700	2.94	0.97
160	6000-6700	3.78	0.97
200	6000-6700	5.08	0.92

Figure 8 presents dependence of friction pad's thickness of disc brake *G* on RMS value of vibration accelerations a_{RMS} in considered frequency bands.

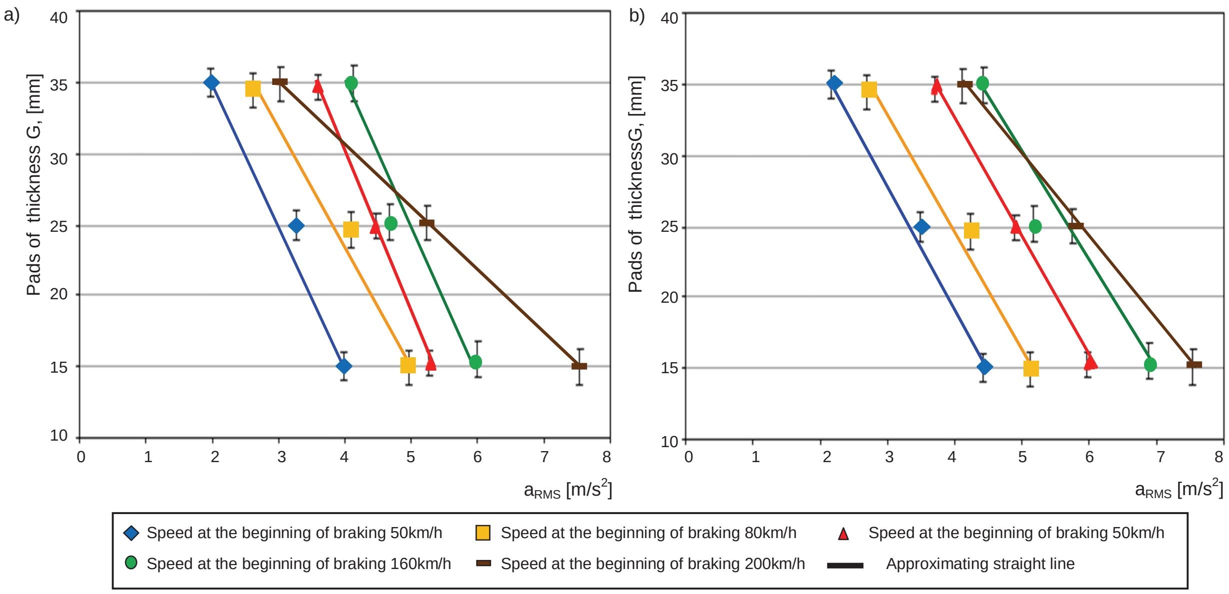

Fig. 8. Dependence of pad's thickness in function of RMS value of vibrations accelerations: a) for frequency band 4600-5100Hz, b) for frequency band 6000-6700Hz

On the basis of approximation function of the wear of friction pads against RMS value of vibration accelerations, linear dependences (21-30) were implemented for considered speeds at the beginning of braking enabling defining current friction pad's thickness.

$$G_{(v=50,\,4600-5100)} = -8.028 \cdot a_{RMS(v=50)} + 50.395, \tag{21}$$

$$G_{(v=80,\,4600-5100)} = -7.929 \cdot a_{RMS(v=80)} + 56.032, \tag{22}$$

$$G_{(v=120,\,4600-5100)} = -11.585 \cdot a_{RMS(v=120)} + 76.09, \tag{23}$$

$$G_{(v=160,\,4600-5100)} = -12.254 \cdot a_{RMS(v=160)} + 81.206, \tag{24}$$

$$G_{(v=200,\,4600-5100)} = -4.64 \cdot a_{RMS(v=200)} + 49.346, \tag{25}$$

$$G_{(v=50,\,6000-6700)} = -9.07 \cdot a_{RMS(v=50)} + 56.203, \tag{26}$$

$$G_{(v=80,\,6000-6700)} = -8.922 \cdot a_{RMS(v=80)} + 61.638, \tag{27}$$

$$G_{(v=120,\,6000-6700)} = -11.01 \cdot a_{RMS(v=120)} + 80.528, \tag{28}$$

$$G_{(v=160,\,6000-6700)} = -8.179 \cdot a_{RMS(v=160)} + 68.484, \tag{29}$$

$$G_{(v=200,\,6000-6700)} = -5.28 \cdot a_{RMS(v=200)} + 53.16, \tag{30}$$

Verification of stationary research on pads' thicknesses: G4=21.1mm and G5=16.2mm showed that diagnose error stemming from application of dependencies (21-30) equalled 4-7% for pad G5 and 7-12% for pad G4. Lower values of diagnose error were obtained for band 6000-6700Hz.

On this basis, it can be concluded that it is possible to define the wear of friction pads for selected speeds at the beginning of braking with known RMS value of vibration accelerations.
The analysis of results of research in frequency function showed that for certain frequency band it is possible to diagnose the wear of friction pads independently of the speed at the beginning of braking. The dynamics of changes of RMS values of vibration accelerations for pads: G_1, G_2 and G_3 can be found in the range between 2.7 and 7.6 dB.

5. Conclusions

Analyzing time flows for considered pads' thickness, occurrence of self-excitation vibrations for pads worn to thickness of 15mm was observed. The effect of self-excitation vibrations may be connected with change of dynamics properties of the system caused by change of caliper and pad's mass, which is particularly visible at the end of braking.

For this purpose the range of frequencies in which self-excitation vibrations occur together with self-vibrations of the pad and caliper, should be checked, which can be effected by modal analysis of the lever set. In the further stages of works it is planned to carry out mentioned analysis to check occurrence of resonance frequencies of braking system.

In the diagnostics of the wear of friction pad of disc brake, point parameters obtained from amplitude flows of vibration accelerations are easier to interpret. Analyzing results in the range of applying point parameters of signals of vibration accelerations to determine friction pads' wear determined by current pads' thickness in the moment of measurement, it can be found out that selected parameters allow to determine friction pads' thickness..

Measurement of vibration accelerations in direction Y_2 direction orthogonal to friction surface of the disc and mounting vibration converter from the side of brake cylinder's case, is characterized as the most sensitive towards direction Z and X, which is confirmed by values of coefficient of dynamics of changes defined with dependence (5). During verification of regression diagnostic models determined on the basis of point parameters of signals coming from pad caliper, differences in determining pads' thickness did not exceed 11% in direction Y_2 based on average value, 9% for RMS value and 14% for RMS value in direction Z_2.

Analysis of calliper vibrations in frequency domain enables to diagnose the wear of friction pads in two bands: 4600-5100 and 6000-6700Hz independently of speed at the beginning of braking.

For analysis in frequency domain, coefficients of dynamics of changes equal 2.7-7.6dB depending on the speed at the beginning of braking. Using RMS value of vibration accelerations it is possible to use diagnostic models to define the wear of friction pads at considered speeds at the beginning of braking.

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