APPLICATION OF FFT FAST FOURIER TRANSFORM TO THE VIBRATION SIGNAL GENERATED BY THE DISC BRAKE TO ASSESS THE WEAR OF FRICTION PADS DURING BRAKING WITH THE CONSTANT BRAKING POWER

WOJCIECH SAWCZUK[1], FRANCISZEK TOMASZEWSKI[2]

Poznan University of Technology, Faculty of Working Machines and Transportation

Summary

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 limit wear particularly of friction pads. The purpose of this article is to present possibility to diagnose the friction set of disc brake by using selected frequency characteristics of vibration signal generated by brake caliper with 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 during braking with the constant braking power.

Keywords: railway disc brake, diagnostics of brake, frequency analysis

1. Introduction

In rail vehicle, because of constantly rising ride speed and to obtain required braking distance, disc brakes are used as primary brake.

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 [3]. To check the wear of friction pads and brake discs it is necessary to apply specialistic

[1] Poznan University of Technology, Institute of Combustion Engines and Transport, Division of Rail Vehicles, 3 Piotrowo Street, 60-965 Poznan, e-mail: wojciech.sawczuk@put.poznan.pl, ph. +48 61 665 20 23

[2] Poznan University of Technology, Institute of Combustion Engines and Transport, Division of Rail Vehicles, 3 Piotrowo Street, 60-965 Poznan, e-mail: franciszek.tomaszewski@put.poznan.pl, ph. +48 61 665 23 55

station e.g. inspection channel to carry out inspections, and to carry out replacement of friction parts in case they reach their terminal wear.

The purpose of this research is to apply vibration signal of pad calipers to assess the wear of friction pads of disc brake using selected frequency characteristics during station research.

2. Methodology and research object

The research was carried out at internal station for tests of railway brakes in Institute of Rail Vehicles TABOR in Poznan. A brake disc type 590×110 with ventilation vanes and three sets of pads type 175 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.

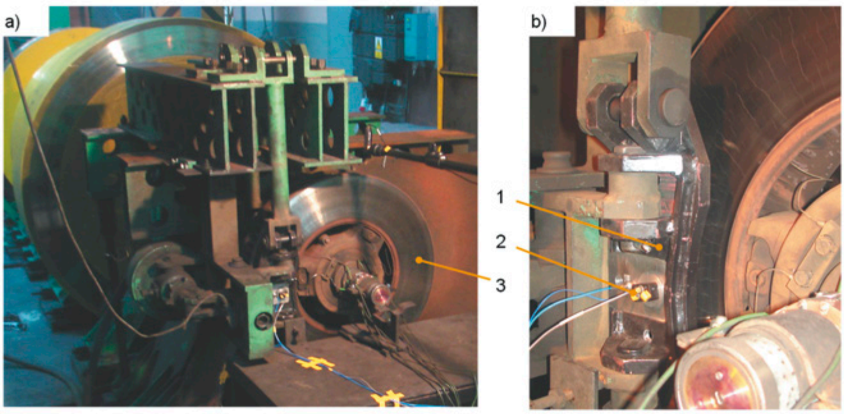


Fig. 1. Interial station for tests of railway brakes: a) view of level set of railway disc brake, b) pad caliper with acceleration; 1 – calliper with pad, 2 – accelerometer, 3 – disc brake type 590×110

A research program 2B1 (II) according to instructions of UIC 541-3 was applied (C.2 annex). The braking was carried out from speed of 80 km/h and it was the braking with the constant braking power P=45 kW [5]. This is a simulation of a train ride through the Gotthard pass in Switzerland, at a constant speed when braking During the research pad's pressures to disc N of 25 kN was realized as well as braking masses per one disc of M=5.7T. the total simulated time, exit the train at the time of the research amounted to t = 10 min. Vibration accelerometers were mounted on pad calipers with a mounting metal tile, which is presented in Fig. 1b [2].

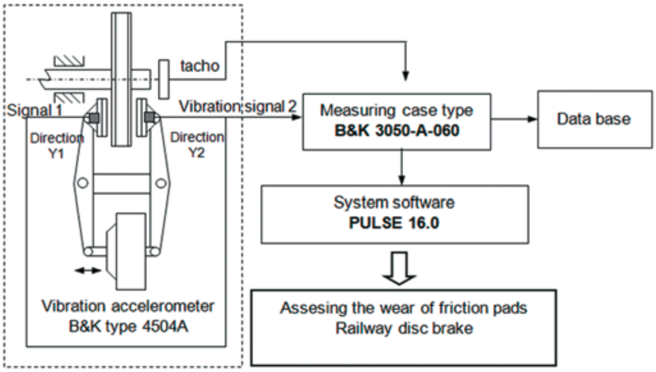


Fig. 2. Measurement set of vibrations generated by caliper with pads

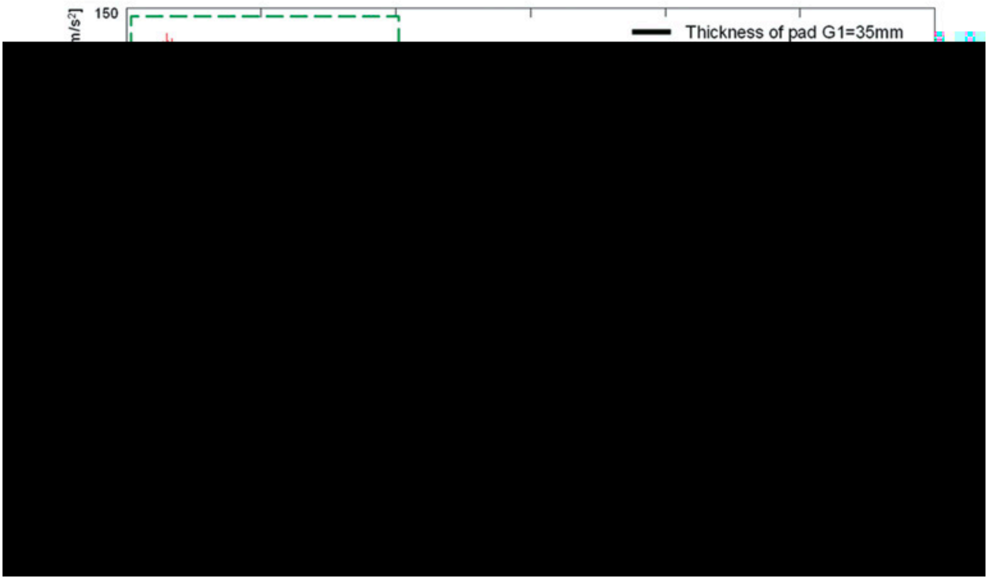
During the research signals of vibration accelerations were registered in one perpendicular direction. To acquire vibration signal a measuring system consisting of piezoelectric vibration accelerations and measuring case type B&K 3050-A-060 with system software PULSE 16.0 was used. Fig. 2 presents the view of the measurement set.

Brüel&Kjær's vibration converters type 4504 were selected on the basis of instructions included in paper [1], the linear frequency of accelerations transit amounted to 13 kHz. Sampling frequency was set at 131 kHz. This means that the frequency that was subject of the analysis in accordance with Nyquist relation amounted to 65 kHz.

This research was carried out in accordance with principles of active experiment. After carrying out a series of brakings the friction pads were changed and values of instantenous vibration accelerations were registered.

3. Research results

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. Spectral analysis was used the first a piece of the vibration signal from the brake braking disk brakes. Time analysis of instantaneous value of vibration acceleration shows that the time indicated that the braking time of 20 seconds from the beginning of braking is a period in which there is the largest dynamics of change (dependence (2)) vibration signal from the wear of friction pads. Spectrum received on measurement of vibrations in direction perpendicular to friction surface of the disc (directions Y_1 and Y_2). Vibration measurement was made on the cast from the brake cylinder and the cast from the brake cylinder piston rod.



[redacted]

Fig. 3. Instantaneous value of vibration acceleration as a function of time during braking with the constant braking power for the thickness of the friction pads: t_a – a piece of the analyzed signal spectral, t_a = 20s

Fig. 4. Dependence of amplitude of vibration accelerations on frequencies for different pad's thicknesses during braking with constant braking power for brake disc type 590×110 in direction Y_I: a) pad's thickness G1=35 mm, b) pad's thickness G2=25 mm, c) pad's thickness G3=15mm

Figure 4 and 5 presents exemplary amplitude spectra of vibration accelerations for various pad's thicknesses received during braking with constant braking power on the disc of an external diameter 590 mm (in both directions of vibration measurement Y_1 and Y_2).

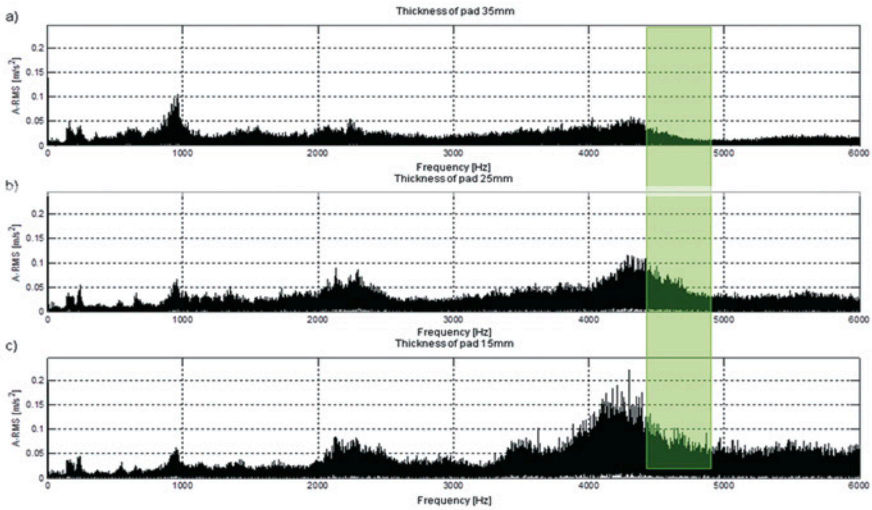


Fig. 5. Dependence of amplitude of vibration accelerations on frequencies for different pad's thicknesses during braking with constant braking power for brake disc type 590×110 in direction Y_2: a) pad's thickness G1=35 mm, b) pad's thickness G2=25 mm, c) pad's thickness G3=15 mm

Research on measurement of vibration accelerations of brake calipers in frequency domain showed that it is possible to find frequency bands, in which dependence of RMS value of vibration accelerations A_{RMS} (dependence (1)) [6] on various pad's thicknesses in the field of speed when the train with brakes.

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

where:
T – average time [s],
$s(t)$ – instantaneous value of vibration accelerations [m/s²].

Table 1. Results from frequency analysis of vibration acceleration signal caliper with friction pads

Frequency [Hz]	RMS from band frequency [m/s²]			Dynamics of changes [dB]		Correlation coefficient R
Brake disc type 590 × 110, measured in the direction of the Y_1 (measurement from the side of the brake cylinder)						
	Thickness of 35 mm	Thickness of 25 mm	Thickness of 15 mm	For pads with thicknesses of 35 mm and 25mm	For pads with thicknesses of 35 mm and 15mm	
4600-4650	0,6975	1,2502	1,8023	5,07	8,24	1,0000
4650-4700	0,7985	1,3527	1,8339	4,58	7,22	0,9992
4700-4750	0,9238	1,4588	1,9910	3,96	6,66	1,0000
4750-4800	0,9319	1,4433	1,8881	3,79	6,13	0,9992
Brake disc type 590 × 110, measured in the direction of the Y_2 (measurement from the side of the brake cylinder piston rod)						
4600-4650	0,2842	0,8684	1,1016	9,70	11,76	0,9706
4650-4700	0,2409	0,7895	1,1376	10,31	13,48	0,9918
4700-4750	0,1953	0,6365	1,0719	10,26	14,79	1,0000
4750-4800	0,1647	0,5529	0,9084	10,51	14,82	0,9997

Table 1 presents frequency range, in which dependence of amplitude value of vibration accelerations on the wear of pads is observed (measurement from the side of the brake cylinder (Y_1) and measurement from the side of the brake cylinder piston rod (Y_2)). Additionally, dynamics of changes according to dependence (2) [4] 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.

Vibration research disc brake during internal station for tests of railway brakes showed that frequency band, associated with a change in the thickness of the pads during braking, it do not need to set the method of spectral analysis of vibration acceleration signal recorded during the whole braking process (with pressure pads for brake disc).

Another, also an effective method of identifying those frequency bands, is registering the vibration acceleration in the last phase of braking, i.e.. at the time of departure pads from the disc at a total stopped (v = 0 km/h). In Figure 6 shows the process instantaneous time acceleration throughout the process of braking, with a visible pulse associated with the departure of pads brake disc on the pressure drop in the brake cylinder. Figure 7 shows the A_{RMS} spectrum of the impulse is offset from the lining of the disc.

Figure 8 present dependence of (RMS) value of vibration accelerations for thickness of pads in the frequency bands.

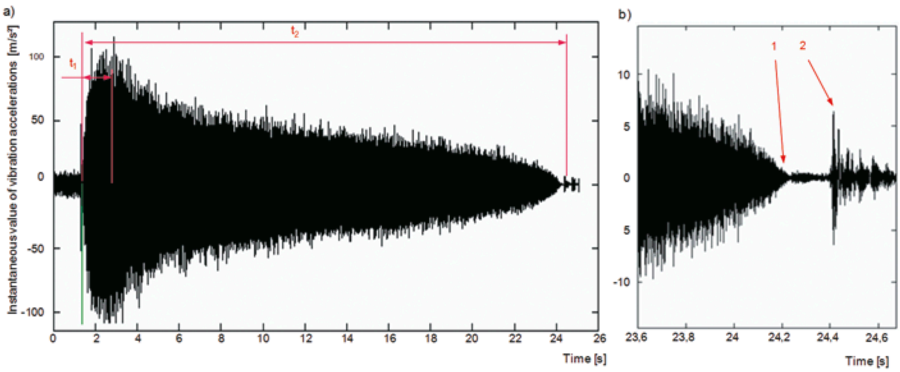


Fig. 6. Instantaneous value of vibration acceleration as a function of time: a) the whole process of braking, b) for the end of the braking; t_1 – pressure rise time pads for disc, t_2 –time of braking 1 – end of braking, v=0, 2 – the impact of the departure vibration acceleration from caliper with pads from the disc

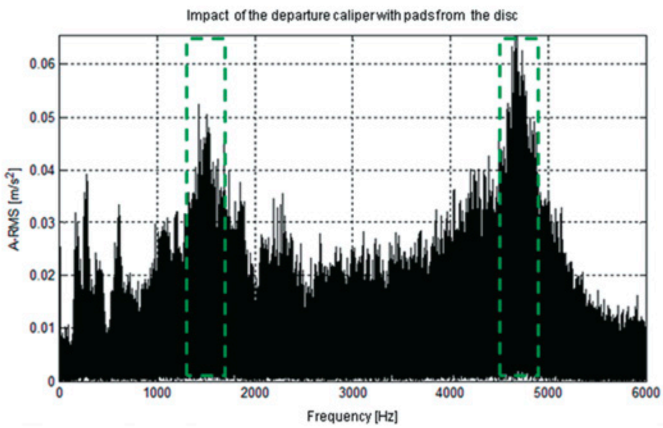


Fig. 7. Dependence of amplitude of vibration accelerations on frequency recorded for the time of departure caliper with pads from the brake disc type 590×110, for direction Y_1

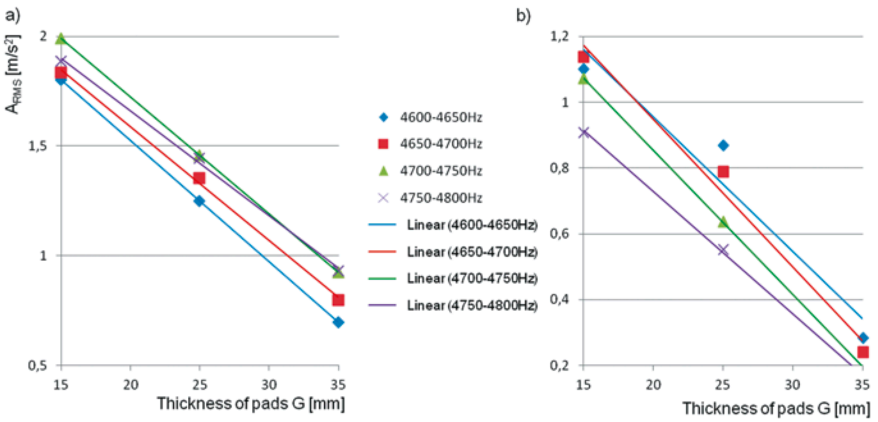


Fig. 8. Dependence of RMS value of vibration accelerations for thickness of pads: a) for disc brake type 590×110, measured in the direction of the Y_1, b) for disc brake type 590×110, measured in the direction of the Y_2

$$D = 20\lg\left(\frac{s_2}{s_1}\right) \tag{2}$$

where:
s_1 – the value of point parameter determined for pad G_3 or G_2,
s_2 – the value of point parameter determined for pad G_1.

In Figure 9 are presented depending on the thickness of the friction pads G disc brake from value of vibration acceleration of ARMS in the frequency bands from 4600 to 4800 Hz with a length of 50 Hz for band. For both directions of measurement analysis of vibration (i.e. for the caliper with friction pad connected with lever with the brake cylinder and the caliper with friction pad connected with a lever with brake cylinder piston rod) an approximation of a linear function depending on the thickness of the pads of the effective value of the acceleration.
On the basis of approximation function R_2 of the wear of friction pads against RMS value of vibration accelerations, linear dependences (3-6) were implemented for enabling defining current friction pad's thickness by measuring the vibrations generated by the caliper of side lever brake for brake cylinder. At the time of internal station for tests of railway brakes simulated braking with the constant braking power, the highest values were obtained correlation coefficient for measurement of vibration in the direction of Y_1, also for this direction resulted in the dynamics of change in excess of 6 dB for thickness of pad G3 = 15 mm relative to the new thickness of pad G1 = 35 mm.

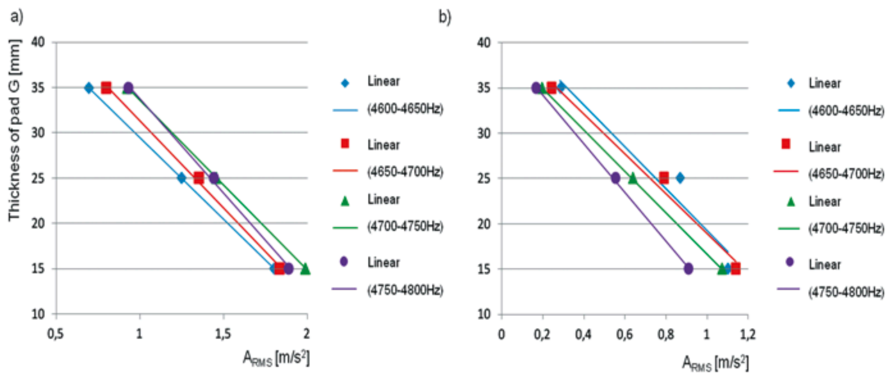


Fig. 9. Dependence of pad's thickness in function of RMS value of vibrations accelerations: a) for disc brake type 590×110, measured in the direction of the Y_1, b) for disc brake type 590×110, measured in the direction of the Y_2

$$G_{(590\,Y_1,\,4600-4650)} = -18{,}103 \cdot A_{RMS(4600-4650)} + 47{,}629 \quad R^2 = 1 \tag{3}$$

$$G_{(590\,Y_1,\,4650-4700)} = -19{,}284 \cdot A_{RMS(4650-4700)} + 50{,}617 \quad R^2 = 0{,}99 \tag{4}$$

$$G_{(590\,Y_1,\,4700-4750)} = -18{,}741 \cdot A_{RMS(4650-4700)} + 52{,}321 \quad R^2 = 1 \tag{5}$$

$$G_{(590\,Y_1,\,4750-4800)} = -20{,}882 \cdot A_{RMS(4750-4800)} + 54{,}676 \quad R^2 = 0{,}99 \tag{6}$$

where:

$G_{(...)}$ – thickness of pad,
$A_{RMS(...)}$ – RMS value of vibrations accelerations in m/s^2 for direction Y_1.

The inaccuracy of the linear regression models described dependencies (3-6) present table 2.

Table 2. Error in % in the application models in estimating linear regression actual thickness of brake pad

Frequency [Hz]	Brake disc type 590 × 110, measured in the direction of the Y_1		
	For thickness of pad G_1= 35mm	For thickness of pad G_2=25 mm	For thickness of pad G_3=15 mm
4600-4650	0,01	0,01	0,01
4650-4700	0,62	1,87	1,65
4700-4750	0,02	0,07	0,05
4750-4800	0,61	1,85	1,63

The analysis of results of research in frequency function showed that for each of the frequency band it is possible to diagnose the wear of friction pads at the designated A_{RMS} value of the acceleration. 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 6 and 8 dB. With a view to the practical application of the presented in the article methods for diagnosis of friction pads, it is necessary to register change in thickness of pads in the whole process of braking and creating a database, from the moment the new braking pads until it reaches an acceptable state and maximum wear friction pads after which replace the pads.

4. Conclusion

The research at internal station for tests of railway brakes showed that it is possible to diagnose the wear of friction pads by using analysis of the values of the vibration acceleration caliper by defining in frequency domain. For the purposes of diagnosis wear of friction pads it is sufficient to register only the first 20 seconds of braking process.

Analysis of caliper vibrations in frequency domain enables to diagnose the wear of friction pads in band: 4600-4800 Hz with a length of 50 Hz for band during braking with the constant braking power. Change the amplitude of vibration acceleration spectrum depending on the wear of friction pads are visible regardless of the vibration acceleration mounting on the caliper of the side of the brake cylinder or the side of the brake cylinder piston rod. For analysis in frequency domain, coefficients of dynamics of changes equals 6-8 dB for braking with the constant braking power. Using RMS value of vibration accelerations it is possible to use diagnostic models to define the wear of friction pads during braking with the constant braking power. The maximum error in estimating the wear of the friction pads for three of the thickness of the pads shall not exceed 2% for the disc type 590×110. Carried out diagnostic at internal station for tests showed that the choice of the frequency band associated with a change in the thickness of the pads can also be made after the impact test, the time of departure from the brake disc pads after braking. Registering instantaneous vibration acceleration, and then to frequency analyse the this portion of the braking also provides information on the amplitude change caused by changing the thickness of the friction pads.

In both cases, the full spectral analysis of braking process, and the time of departure after pad the braking pressure in the brake cylinder to zero showed that frequency 4600-4800 Hz can be used to diagnose the wear and friction pads irrespective of direction of vibration measurement (Y_1 and Y_2).

Impact test demonstrated that also in the 1600 Hz there has been a change in the amplitude of ARMS as a function of the thickness of the pads, obtaining a satisfactory dynamic of changes in excess of 6 dB, but only at the time of measurement in one direction Y_1.

The project is funded by the National Center for Science, N N504 644840

References

[1] Brüel&Kjær: *Piezoelectric Accelerometer Miniature Triaxial Delta Tron Accelerometer* – Type 4504A, oferta firmy Brüel & Kjær, s. 2.

[2] Brüel&Kjær: *Measuring Vibration*. Revision September 1982, s. 16.

[3] GRUSZEWSKI M.: *Wybrane zagadnienia eksploatacji hamulca tarczowego*, Technika transportu Szynowego 6-7/1995, s. 84-86.

[4] GRYBOŚ R.: *Drgania maszyn*, Wydawnictwo Politechniki Śląskiej, Gliwice 2009, s. 214.

[5] KodeksUIC 541-3., *Hamulec-Hamulec tarczowy i jego zastosowanie. Warunki dopuszczenia okładzin hamulcowych*, Wyd. 6, listopad 2006, s. 23.

[6] SAWCZUK W., TOMASZEWSKI F.: *Assessing the wear of friction pads in disc braking system of rail vehicle by using selected amplitude characteristics of vibration signal*. Vibration In Physical Systems, Volume XXIV, s. 335-361.