EVALUATION OF THE WEAR OF FRICTION PADS RAILWAY DISC BRAKE USING SELECTED FREQUENCY CHARACTERISTIC OF VIBRATIONS SIGNAL GENERATED BY THE DISC BRAKE

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Summary

The article presents a new method for diagnosis of the wear of friction pads railway disc brake through analysis of the braking vibration during braking with the constant braking power and during braking to stop by making the analysis of signals in the frequency domain. At the time of research there are vibration acceleration generated by the caliper with brake pads during two types of braking.

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

OCENA ZUŻYCIA OKŁADZIN CIERNYCH KOLEJOWEGO HAMULCA TARCZOWEGO NA PRZYKŁADZIE WYBRANYCH CHARAKTERYSTYK WIDMOWYCH SYGNAŁU DRGANIOWEGO GENEROWANEGO PRZEZ HAMULEC

Streszczenie

Artykuł przedstawia autorską metodę diagnozowania zużycia okładzin ciernych poprzez analizę drgań układu hamulcowego na przykładzie hamowania na spadku jak i hamowania zatrzymującego, dokonując analizy sygnałów w dziedzinie amplitud. W czasie badań rejestrowano przyspieszenia drgań generowane przez obsady hamulcowe z okładzinami w czasie hamowania.

Słowa kluczowe: kolejowy hamulec tarczowy, diagnostyka hamulca, analiza widmowa

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]. 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 (Fig. 1) [9]. To check the wear of friction pads and brake discs it is necessary to apply specialistic station e.g. inspection channel to carry out inspections, and to carry out replacement of friction parts in case they reach their terminal wear.A system for video inspection and diagnostics worked out in Rail Vehicle Institut TABOR in Poznan is the most advanced system do diagnose disc brake. Diagnosing system [3] 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.

Fig. 1. View of two friction set mounted on axle passenger car

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. Rail track stations to diagnose the wear of friction pads are used by German, British and French railways.

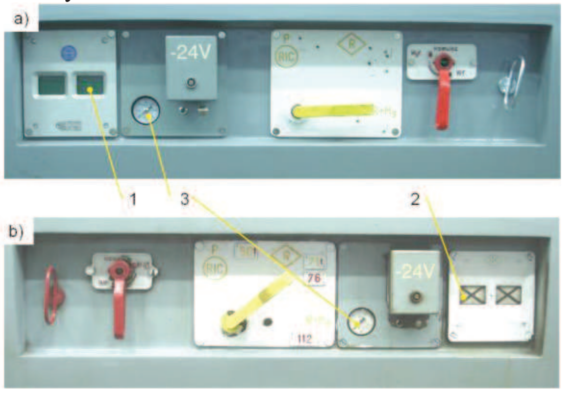

Fig. 2. System of signaling braking mounted on passenger car 136: a) excluded brake, b) other state of the brake: 1 – green colour it is excluded brake, 2 – rectangle crossed out that is other the state of the brake e.g. is missing airs in the brake installation, 3 – manometer pressures

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 (Fig. 2). Nevertheless, rail technique lacks an objective method of quantitive assessment of the wear of friction pads.

The purpose of this research is to apply frequency analysis to evaluatnion of the wear of friction pads on the base vibration signal generated by caliper with pads disc brake lever set.

2. METHODOLOGY AND RESEARCH OBJECT

The research was carried out at internal station for tests of railway brakes. 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 - 35mm thick and two sets were worn to thickness of 25mm and 15mm. A reasearch program 2B1 (II) according to instructions of UIC 541-3 was applied.

The braking was carried out from speed of 80km/h and it was the braking with the constant braking power P=45kW. During the research pad's pressures to disc N of 25kN was realized as well as braking masses per one disc of M=5.7T and during braking to stop [6].

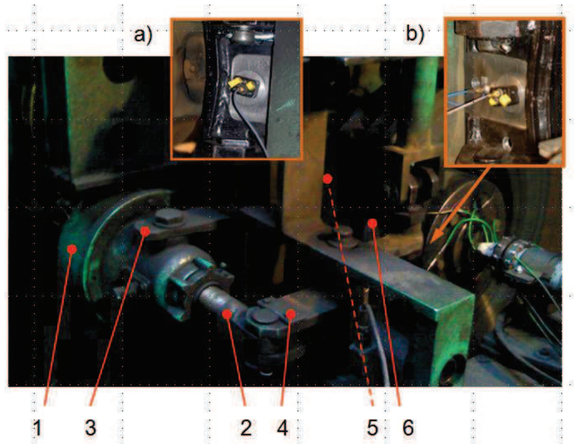

Fig. 3. View of disc brake lever set: a) caliper with vibration converter mounted in side of cylinder casing (is not visible in the picture), b) caliper with vibration converter mounted in side of piston rod, 1- brake cylinder casing, 2- brake cylinder piston rod, 3- left main level, 4- right main level, 5- caliper with vibration converter (is not visible in the picture)

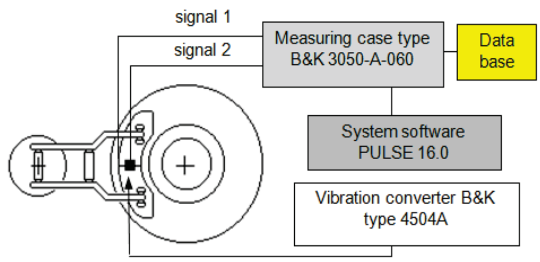

Fig. 4. Measurement set of vibrations generated by caliper with pads

This research was carried out in accordance with principles of active experiment [4]. After carrying out a series of brakings the friction pads were changed and values of instantenuous vibration accelerations were registered.
Vibration converters were mounted on pad calipers with a mounting metal tile, which is presented in Fig. 3a and 3b. During the research signals of vibration accelerations were registered in three reciprocally orthogonal directions [10]. To acquire vibration signal a measuring system consisting of piezoelectric vibration accelerations converter and measuring case type B&K 3050-A-060 with system software PULSE 16.0 was used. Fig. 4 presents the view of the measurement set [1, 2].

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. Figure 5, 6, 7 and 8 presents exemplary amplitude spectra of vibration accelerations for various pad's thicknesses received during braking with constant braking power from speed of 80km/h and braking to stop with the same speed braking. Spectrum received on measurement of vibrations in direction perpendicular to friction surface of the disc (direction *Y*).

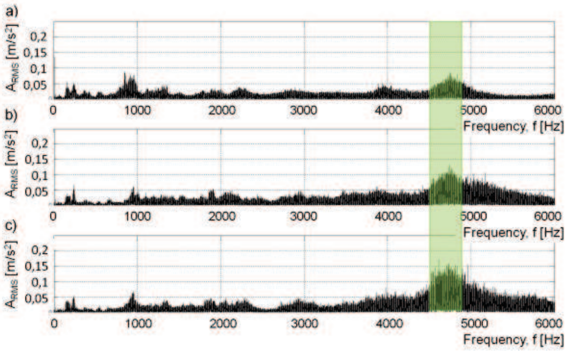

Fig. 5. Dependence of amplitude of vibration accelerations on frequencies for different pad's thicknesses during braking with constant braking power from speed of 80km/h in direction *Y1*: a) pad's thickness G1=35mm, b) pad's thickness G2=25mm, c) pad's thickness G3=15mm

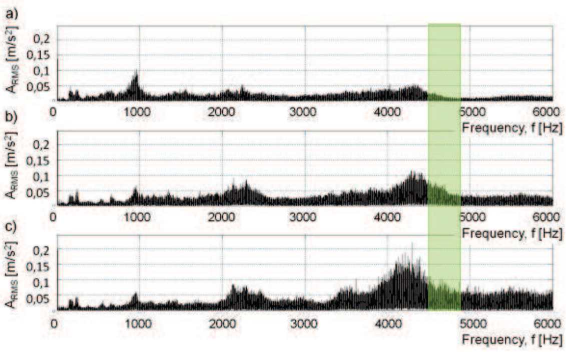

Fig. 6. Dependence of amplitude of vibration accelerations on frequencies for different pad's thicknesses during braking with constant braking power from speed of 80km/h in direction *Y2*: a) pad's thickness G1=35mm, b) pad's thickness G2=25mm, c) pad's thickness G3=15mm

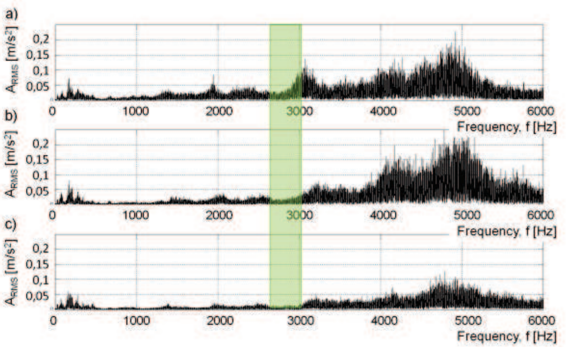

Fig. 7. Dependence of amplitude of vibration accelerations on frequencies for different pad's thicknesses during braking to stop (speed at the beginning of braking v=80km/h) in direction *Y1*: a) pad's thickness G1=35mm, b) pad's thickness G2=25mm, c) pad's thickness G3=15mm

Frequency analysis were part of the first 20 seconds with a constant braking power and the last 5 seconds to stop braking. In the case of braking to stop only last 5 seconds allows braking to observe changes in the spectrum of vibration accelerations.

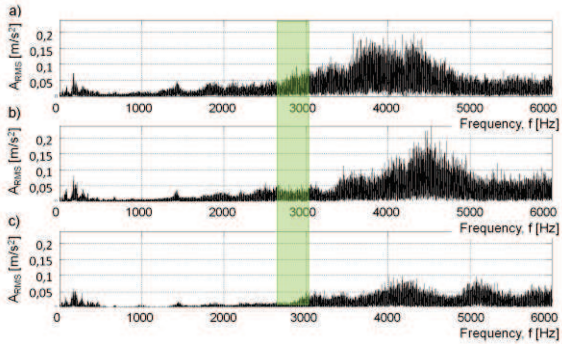

Fig. 8. Dependence of amplitude of vibration accelerations on frequencies for different pad's thicknesses during braking to stop (speed at the beginning of braking v=80km/h) in direction *Y2*: a) pad's thickness G1=35mm, b) pad's thickness G2=25mm, c) pad's thickness G3=15mm

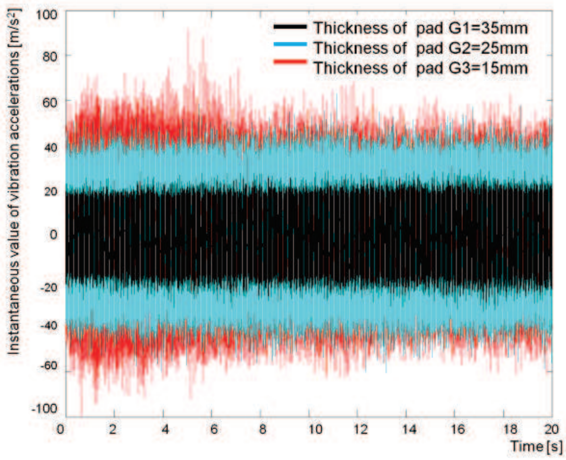

Fig. 9. Signal of vibration accelerations registered on pad caliper in direction Y_1 for different thickness of pads during braking with the constant braking power

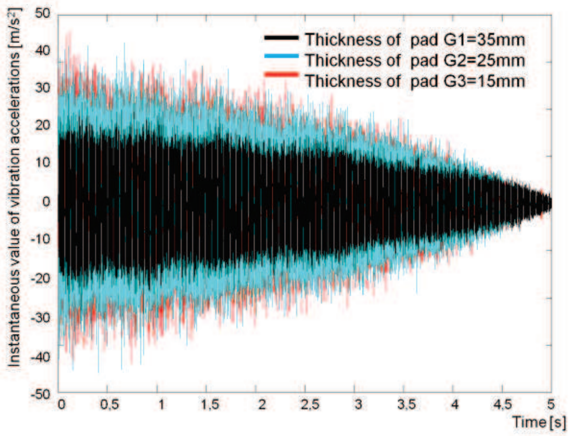

Fig. 10. Signal of vibration accelerations registered on pad caliper in direction Y_1 for different thickness of pads during braking to stop (Speed at beginning of braking v=80km/h)

Figure 9 and 10 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.

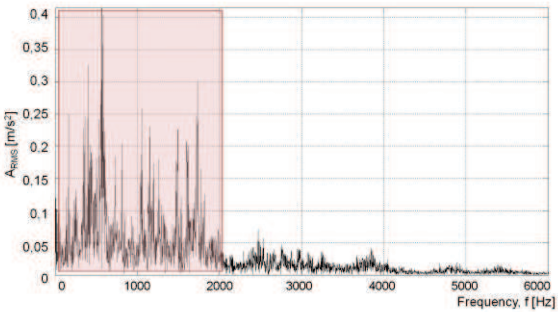

Fig. 11. Dependence of amplitude of vibration accelerations on frequencies for internal station for tests of railway brakes after brake (speed at the beginning of braking v=80km/h)

Frequency analysis has been also the vibrations generated by the internal station for tests of railway brakes during speed v = 80 km/h, it was found that 2kHz do not analyze the spectrum changes depending on the thickness of the friction pads

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} (equation (1)) [4] on various pad's thicknesses in considered range of speeds at the beginning of braking is observed.

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

where:

T – average time [s],

$s(t)$ – instantaneous value of vibration accelerations, in [m/s^2].

Table 1 and 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 (2) [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 type of braking for certain frequency bands.

$$D = 20\lg\left(\frac{A_2}{A_1}\right) \tag{2}$$

where:

A_1 – the value of point parameter determined for pad G_3 or G_2, in [m/s^2],

A_2 – the value of point parameter determined for pad G_1, in [m/s^2].

Table 1 measurement results for braking with the constant braking power

Measurement of vibrations in direction **Y$_1$**					
Frequency Hz	Value of RMS value m/s^2			Dynamics of changes dB	
	Pad 35mm	Pad 25mm	Pad 15mm	G2/ G1	G3/ G1
4600-4800	1,68	2,75	3,76	4,27	6,96
Measurement of vibrations in direction **Y$_2$**					
4600-4800	0,45	1,44	2,11	10,1	13,4

Table 2 measurement results for braking to sop

Measurement of vibrations in direction **Y$_1$**					
Frequency Hz	Value of RMS value m/s^2			Dynamics of changes dB	
	Pad 35mm	Pad 25mm	Pad 15mm	G2/G1	G3/G1
2800-3000	0,345	0,148	0,078	-7,36	-12,88
Measurement of vibrations in direction **Y$_2$**					
2800-3000	0,443	0,231	0,142	-5,65	-9,26

Figure 12 and 13 presents dependence of friction pad's thickness of disc brake G on RMS value of selected two bands frequency. selected point parameters of vibration accelerations. For RMS value of band frequency, also obtained from measurement in direction Y_1 by using linear approximating functions described with dependences (3-6) for speeds at the beginning of braking v=80km/h during braking with the constant braking power and during braking to stop, the following equations defining friction pads' thickness were introduced:

$$G = 69{,}682 \cdot A_{RMS\,(Y1,\,BS)} + 11{,}733 \tag{3}$$

$$G = -9{,}6426 \cdot A_{RMS\,(Y1,\,BP)} + 51{,}372 \tag{4}$$

$$G = 64{,}396 \cdot A_{AVERAGE\,(Y1,\,BS)} + 7{,}2718 \tag{5}$$

$$G = -11{,}862 \cdot A_{AVERAGE\,(Y1,\,BP)} + 40{,}871 \tag{6}$$

where:

G – thickness of pad [mm],

$A_{(..)}$ – RMS value from band frequency [m/s^2],

BS – braking to stop (band frequency 2800-3000Hz),

BP – braking with constant braking power (band frequency 4600-4800Hz).

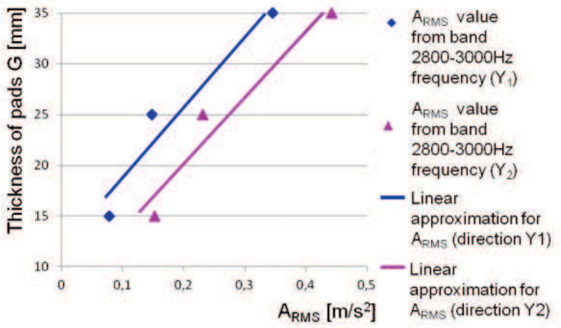

Fig. 12. Dependence of pad's thickness in function of RMS value of vibrations accelerations for frequency band 2800-3000Hz

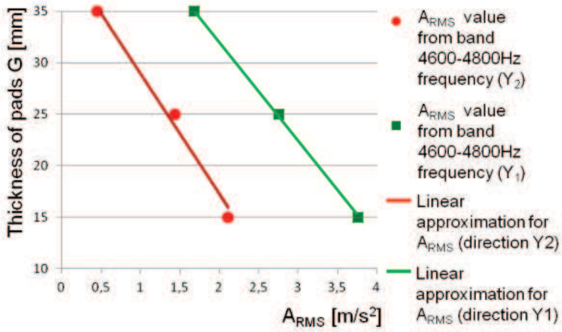

Fig. 13. Dependence of pad's thickness in function of RMS value of vibrations accelerations for frequency band 4600-4800Hz

The inaccuracy of the linear regression models described dependencies (3-10) present table 3 and 4.

Table 3 Error in % in the application models in estimating linear regression actual thickness of brake pad for braking with the constant braking power

Measurement of vibrations in direction Y_1			
Frequency Hz	For brake pad G_1= 35mm	For brake pad G_2=25 mm	For brake pad G_3=15 mm
4600-4800	0,29	0,87	0,74
Measurement of vibrations in direction Y_2			
4600-4800	1,44	5,08	4,82

Table 4 Error in % in the application models in estimating linear regression actual thickness of brake pad for braking to stop

Measurement of vibrations in direction Y_1			
Frequency Hz	For brake pad G_1= 35mm	For brake pad G_2=25 mm	For brake pad G_3=15 mm
2800-3000	2,16	11,84	12,73
Measurement of vibrations in direction Y_2			
2800-3000	2,16	11,43	12,20

The analysis of results of research in frequency function showed that for frequency band 4600-4800Hz it is possible to diagnose the wear of friction pads independently during braking with constant braking power P=45kW. 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 2,3 and 13,4dB.

4. CONCLUSION

The reseach 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.

Analysis of caliper vibrations in frequency domain enables to diagnose the wear of friction pads in band: 4600-4800Hz during braking with the constant braking power P=45kW. Another band 2800-3000Hz of frequency analysis of vibrations signals also checked during braking to stop at the beginning of speed braking v=80km, pressures to disc N of 25kN and braking masses per one disc of M=5,7T.

For analysis in frequency domain, coefficients of dynamics of changes equals 4-6dB (direction Y_1), 10-13dB (direction Y_2) for braking with the constant braking power. And coefficients of dynamics of changes equals 7-12dB (direction Y_1), 5-9dB (direction Y_2) for braking to stop. 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. Error diagnosis using regression diagnostic models equaled 0,3-5% for braking with the constant braking power (for both direction measurement) and 2-12% for braking to stop also for both direction measurement.

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Participation in numerous research work, including as Director of grants and research, development and grants KBN and work for industry.