Article citation info:
Sawczuk W, Rilo Cañás A-M, Kołodziejski S. Evaluation of weight wear of disc brake pads after test stands. Rail Vehicles/Pojazdy Szynowe. 2022;(3-4):53-59. https://doi.org/10.53502/RAIL-162350

Rail Vehicles/Pojazdy Szynowe

www.railvehicles.eu

Evaluation of weight wear of disc brake pads after test stands

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ARTICLE INFO

Received: 20 November 2022
Revised: 18 December 2022
Accepted: 24 December 2022
Available online: 28 December 2022

KEYWORDS

friction pad
pad wear by weight
brake test bench
camera IR

The article presents the results of the research of a railway disc brake in the evaluation of the weight wear of friction pads. The tests were carried out on a certified brake stand where the friction and mechanical characteristics of the brake are determined. The stand was additionally equipped with a thermal imaging camera to observe the contact of the pads with the brake disc. Attention was also paid to examining the influence of such parameters of the braking process as the contact surface of the pad with the disc, the thickness of the pads as a determinant of their initial wear, the pressure of the pads to the disc, braking mass and braking speed on the weight wear of the friction pads.

1. Introduction

When designing friction brakes in sequence, compliance with the requirements of UIC or EN-PN regulations [24, 25] should be ensured. This is very important due to the brake efficiency and the related braking distance [23]. Then, in the second stage, the wear characteristics of the friction material are checked and determined during tests and bench tests. Manufacturers of friction pads in their laboratories select such a composition of materials for friction pads to achieve a compromise in terms of the best friction-mechanical characteristics and acceptable wear. For several years, environmental issues have also started to concern manufacturers of brake pads. The first action in this regard was the elimination of asbestos from the production of friction materials, due to its carcinogenic properties, which was found in the 1970s [4, 5]. Despite the fact that as a material in combination with copper fibers, it creates the best material for brakes due to resistance to high temperatures and stability of friction characteristics of brakes [6, 21]. The issue of emission of wear products from the braking system to the environment has already been addressed by many researchers in works [3, 12]. The works drew attention to the issues related to environmental pollution as well as the impact of consumption products on the health of people in the vicinity of roads. For this purpose, additional measures were taken to eliminate or reduce the share of copper in the production of friction materials, which was included in the works [16, 18]. Therefore, many research centers have attempted to replace toxic and dangerous components for the environment with new organic materials or components such as titanium [2, 7]. In the field of new friction materials for disc brakes, tribological tests are carried out in the field of determining the friction-mechanical characteristics described in works [14, 28], checking the possibility of using the pads in a long-term operation in various braking conditions without adverse impact on the environment. An important issue raised in the work [13] is not only the impact of environmental pollution by solid and volatile particles from the braking process, but also their impact on the braking system and friction characteristics related to the efficiency of the disc brake. During these tests, the requirements regarding vibrations and noise generated by the braking systems de-

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scribed in [16] are additionally checked. The work [22] also indicated the dependence of the weight wear of the friction pads on the vibroacoustic signal generated by the disc brake. All the issues raised require knowledge of the mechanism of friction and wear of friction materials described in works [1, 9], which, in addition to models of friction and wear, are based on statistical tests carried out in bench and operational conditions [15]. In the field of rail vehicle brakes, the wear of the friction material is particularly important for vehicle owners for whom the purchase of friction pads and the related wear is assessed in the category of costs of a transport company. In the case of EMU electric multiple units or electric locomotives, the wear of the friction pads can be reduced due to the nature of the braking system operation. In these vehicles, most of the braking force is provided by the ED (frictionless) electro-dynamic brake using the traction motors in generator operation generating additional resistance. Only the missing braking force in the last phase is supplemented with the EP friction brake due to the characteristics of the traction motor. The share of the electrodynamic brake in relation to the friction brake varies in the range of about 80/20%. In this regard, many centers, both domestic and foreign [8, 26, 27] conduct research on the continuous increase in the effectiveness of electrodynamic braking. Thanks to this, the emission of dust and gases from the friction brake is generated to a minimum, which significantly reduces the weight wear of the friction pads of the pneumatic or electropneumatic brake.

The aim of the article is to present the dependence on determining the weight wear of friction pads as a function of the values characterizing the braking process. A regression model was determined to estimate the wear of friction pads based on a single braking.

2. Research object and methodology

The weight wear test of the friction pads was carried out on a certified inertial brake stand located at the Łukasiewicz Poznański Instytut Technologiczny. Figure 1 shows the view of the stand from the side of the tested brake disc and the measuring system for data recording. On the stand, it is possible to perform friction-mechanical tests of the railway block brake and the disc brake, reflecting the real conditions that occur during the braking of a rail vehicle. During the tests, the thermal imaging camera Flir e60 was additionally used to observe the temperature distribution of the friction pads after stopping braking.

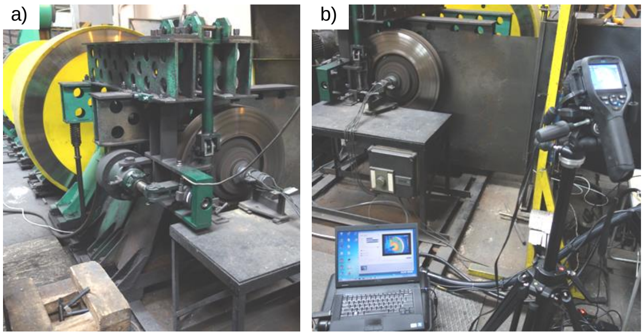


Fig. 1. Stand for testing a railway disc brake: a) view of the working part with rotating masses, b) view of the tested brake disc with thermal imaging measurement

The research was carried out based on the assumptions of an active experiment in accordance with [11]. During the tests, the input parameters (concerning the braking process) were deliberately changed and their influence on the changes in the output parameters – the weight wear of the friction pads – was observed. The tests covered organic friction pads of type 175 and 200 (Fig. 2) cooperating with a ventilated brake disc with dimensions of 640 × 110 made of gray cast iron.

Fig. 2. Friction pads used during the tests: a) type 175, b) type 200, view from the side of contact with the disc

The pads of the FR20H.2, in accordance with the manufacturer's procedure and the requirements contained in [24], were made of thermosetting resin, synthetic elastomer, metal and organic fibers and friction modifiers. Two types of pads, three sets of pads of different thickness, were used for the stand tests, which are the determinant of initial wear. The first set consisted of new pads with a thickness of 35 mm, the second and third sets consisted of worn pads to a thickness of 25 and 15 mm.

Bench tests were carried out in accordance with the UIC 541-3 card in the scope of selecting the pressure of the friction pads to the brake disc and the mass to be braked per one brake disc. The parameters changed during the friction pad wear tests were friction pad thickness, braking start speed (v = 50, 80, 120, 160 and 200 km/h, pressure of the pads to the disc (N =

16, 25, 26, 28, 36, 40 and 44 kN) and braking weight per disc (M = 4.4, 4.7, 5.7, 6.7 and 7.5 t).

After each braking of a pad of a certain thickness from a given speed, pressure, and mass to be braked, the pads were weighed on an electronic scale with an accuracy of 1 gram. During the friction-mechanical tests, 150 brake applications were performed, not counting the brakes related to the running-in of the friction pads.

3. Test results and analysis

The purpose of the bench tests was to determine the relationship between the weight wear of the pads as a function of such parameters as the contact surface of the pad with the disc, the braking start speed, the thickness of the friction pads, the pressure of the pads against the disc and the mass to be braked per one brake disc.

Table 1. Mass wear (in grams) of the brake pads after braking to a complete stop

Force N = 28 kN, mass to be decelerated M_h = 6.7 t			
Initial speed of deceleration, km/h	Pad thickness 35 mm (G_1)	Pad thickness 25 mm (G_2)	Pad thickness 15 mm (G_3)
50	0	0	0
80	1	1	1
120	2	2	3
160	4	5	5
200	9	10	10
Force N = 40 kN, mass to be decelerated M_h = 6.7 t			
50	0	0	0
80	1	1	1
120	3	3	3
160	6	6	7
200	12	13	14

First, for various cases of braking, i.e. a combination of pressure and pad thickness, the weight increase of pad wear as a function of braking speed was determined. Table 1 contains selected results from the weight measurement of friction pad wear for one of the selected braking combinations.

Analyzing the results contained in Table 1, it is found that with the increase in the braking speed, pressure and initial wear of the friction pads, the weight wear of the friction pads increases.

Figures 3 and 4 graphically show the dependence of pad wear on speed with a proposed approximation function. For the quadratic, power and exponential functions, the fit of the results from bench tests to the regression model based on the coefficient of determination R^2 was checked. Values of coefficients for individual regression functions are included in the graphs.

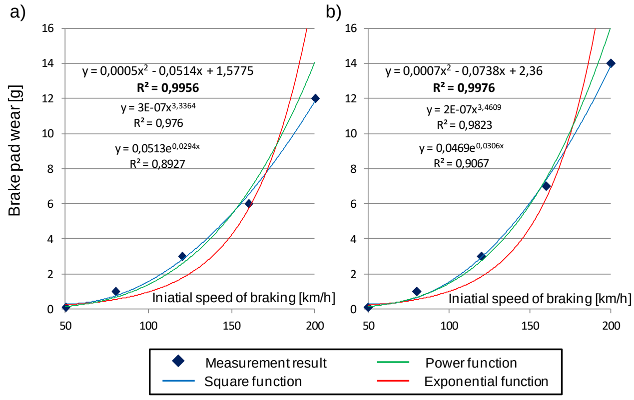


Fig. 3. The relationship of brake pad wear as a function of initial speed at N = 40 kN and M_h = 6.7 t, braking with: a) new 35 mm brake pads, b) 15 mm brake pads

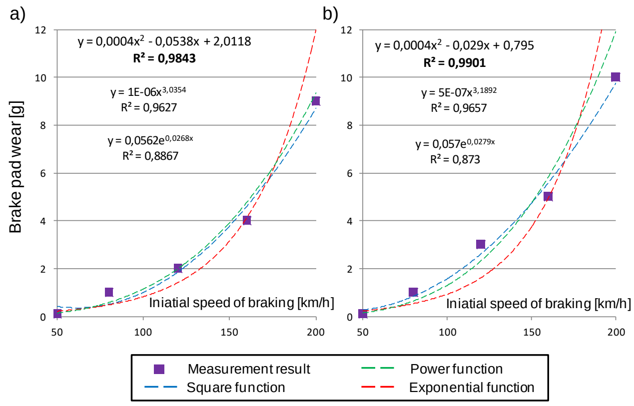


Fig. 4. The relationship of brake pad wear as a function of initial speed at N = 28 kN and M_h = 6.7 t, braking with: a) new 35 mm brake pads, b) 15 mm brake pads

Analyzing the graphs presented in Figures 3 and 4, it is concluded that, regardless of the applied pressure of the pads to the disc, the mass to be braked and the thickness of the pads, the weight wear of the pads as a function of the braking speed can be modeled using the regression quadratic function. In each case, the highest coefficient of determination was obtained in the range of 0.98–0.99. Equations (1–3) describe the braking wear with N = 40 kN and M_h = 6.7 t.

$$Z_{35\_40\_6.7} = 0.0005v^2 - 0.0514v + 1.577 \tag{1}$$

$$Z_{25\_40\_6.7} = 0.0006v^2 - 0.0717v + 2.425 \tag{2}$$

$$Z_{15\_40\_6.7} = 0.0007v^2 - 0.737v + 2.360 \tag{3}$$

The entry from equation (1) $Z_{35\_40\_6.7}$ is the wear in grams after one braking on a new pad with N = 40 kN and M_h = 6.7 t.

Figure 5 shows for selected braking combinations a summary of the weight wear of friction pads as a function of braking speed and thickness of friction pads for a given pressure and mass to be braked.

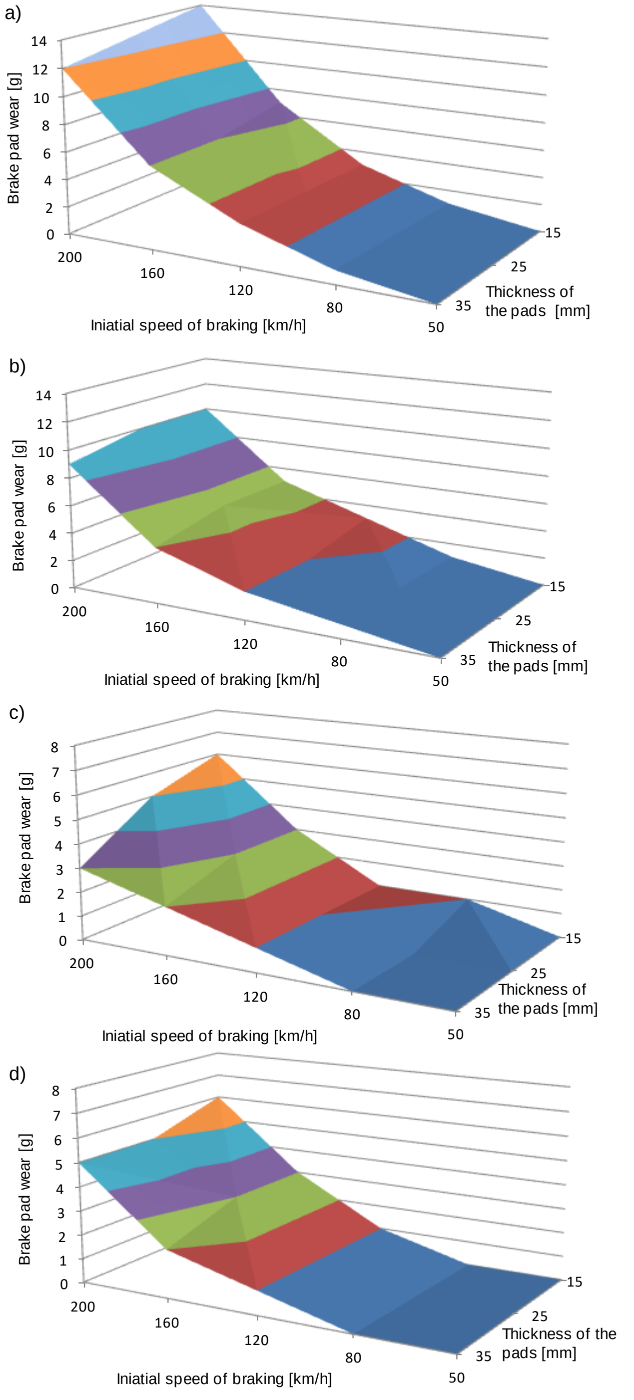


Fig. 5. Mass wear of the pads at: a) N = 40 kN, M_h = 6.7 t, b) N = = 28 kN, M_h = 6.7 t, c) N = 26 kN, M_h = 4.7 t, b) N = 16 kN, M_h = = 4.7 t

The bench tests carried out on 150 brake applications with various combinations in terms of speed, pressure, mass to be braked and pad thickness prove the existence of a relationship between the weight of pad wear and the input (set) parameters of the braking process.

4. Modelling of brake pad mass wear

On the basis of the results of tests of the weight wear of the friction pads, an attempt was made to model it based on the following input parameters, such as the contact surface of the pads with the disc, the thickness of the friction pads, the pressure of the pad to the brake disc, the braking mass per one disc, and the braking start speed.

A multiple regression model, also called polynomial regression, was used to model the weight consumption. It is a method in which the value of the random variable Y depends on the k-th independent features (X_1, X_2, ... X_k). Based on a given sample of test results, in accordance with [10], the invariant parameters β_0, β_1, ... β_k were determined using the least squares method. The following relationship was proposed to determine the weight wear of pads:

$$Z_w = \beta_1 P_O + \beta_2 G_O + \beta_3 N + \\ + \beta_4 M_h + \beta_5 v + \beta_6 v^2 + \beta_0 \quad [g] \tag{4}$$

where: P_O – contact surface of the pad with the disc (4 × 175 cm^2, 4 × 200 cm^2), G_O – thickness of friction pads (new 35 mm (G_1), worn down to 25 mm (G_2) and 15 mm (G_3), N – brake pad pressure (N = 16, 25, 26, 28, 36, 40 and 44 kN), M_h – mass to be braked per one disc (M = 4.4; 4.7; 5.7; 6.7 and 7.5 t), v – braking start speed (v = 50, 80, 120, 160 and 200 km/h).

The multiple regression parameters for the model (4) were calculated according to the relationship (5) [23].

$$r = \frac{\sum_{i=1}^{n}\left(x_i \bar{x}\right)\left(y_i - \bar{y}\right)}{\sqrt{\sum_{i=1}^{n}\left(x_i - \bar{x}\right)^2 \sum_{i=1}^{n}\left(y_i - \bar{y}\right)^2}} \tag{5}$$

where: $\bar{y}$, $\bar{x}$ – average values of features x and features y, y_i, x_i – describing variables.

At the same time, the empirical model described by relation (4) was verified and the significance of the system of regression coefficients and the significance of individual regression coefficients were checked. By verifying the empirical models described by dependencies (6) and (7), statistical tests were performed. Using the example of weight consumption, the statistical hypothesis regarding the significance of the system of regression coefficients was formulated as follows:

$$H_o : \sum_{k=0}^{n} \beta_k^2 = 0; \; (k = 0, 1, 2, ..., 6) \tag{6}$$

$$H_1 : \sum_{k=0}^{n} \beta_k^2 \neq 0; \; (k = 0, 1, 2, ..., 6) \tag{7}$$

Rejection of H_o means that there are statistical grounds for assuming that there is a linear relationship between the dependent variable and at least one explanatory variable. The Snedecor F distribution was used in the test of the significance of the regression model.

To determine the significance of individual regression coefficients, hypotheses written with dependencies were put forward:

$$H_o : \beta_k = 0 \tag{8}$$

$$H_1 : \beta_k \neq 0 \tag{9}$$

The Student's t-distribution was used to test the hypotheses regarding the significance of individual regression coefficients. If the significance of F is lower than the assumed significance level α (α = 0.05), there are grounds for rejecting the null hypothesis and assuming that there is a linear relationship between the explanatory variable and all explanatory variables included in the models.

Table 2 presents the values of the coefficients of the multiple regression function together with the coefficient of determination R^2 for the model of weight wear of the friction pads after statistical tests.

Table 2. Results of the statistical test for the pad wear model of a disc brake

Coefficient	Value	Value F*
β_1	$-2.11·10^{-2}$	**0.24**
β_2	$-3.40·10^{-2}$	**0.12**
β_3	$9.12·10^{-2}$	$4.56·10^{-5}$
β_4	$1.17·10^{-4}$	$2.15·10^{-11}$
β_5	$-4.94·10^{-2}$	$1.32·10^{-2}$
β_6	$4.86·10^{-4}$	$4.16·10^{-9}$
β_0	-3.22	**0.39**
R_2	0,81	
F^{**}	$6,31·10^{-49}$	

* significance for individual coefficients of regression
** significance for the entire system

Analyzing the results of the statistical test contained in Table 2, it is found that some coefficients, i.e. β_0, β_1 and β_2 of the model described by relation (1), do not meet the assumed significance level α = = 0.05. These coefficients were removed and the polynomial regression was recalculated without taking into account the variables related to the contact surface of the pad with the disc (Po) and the thickness of the pads (Go). Table 3 presents the results of the statistical test for the friction pad wear model after verification of its coefficients.

Table 3. Results of the statistical test for the brake pad wear model of a disc brake after validation of its coefficients

Coefficient	Value	Value F*
β_1	$9.00·10^{-2}$	$6.09·10^{-5}$
β_2	1.16	$3.04·10^{-11}$
β_3	$-4.94·10^{-2}$	$1.37·10^{-2}$
β_4	$4.86·10^{-4}$	$2.15·10^{-11}$
β_0	-8.11	$4.89·10^{-9}$
R_2	0.80	
F^{**}	$2.67·10^{-50}$	

* significance for individual coefficients of regression
** significance for the entire system

The final form of the weight wear model of the friction pads based on the given values describing the braking process after verification of the parameters of the multiple regression model is presented by the dependencies:

$$\begin{aligned} Z_w = {} & 9.00\times10^{-2}N + 1.16M_h - \\ & + 4.94\times10^{-2}v + 4.86\times10^{-4}v^2 - 8.11 \ [g] \end{aligned} \tag{10}$$

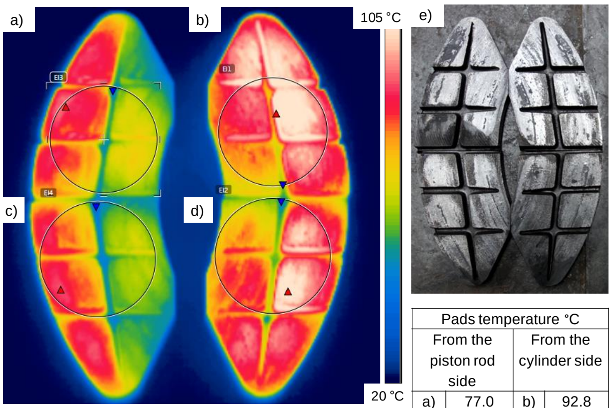
Then, the Pearson linear correlation coefficient (Table 4) was checked for the analyzed variables, i.e. the pressure of the pads to the disc, the mass to be braked and the braking start speed after verification of the coefficients of the weight model of friction pad wear.

Table 4. Correlation matrix for the model of mass wear variables

Variable	Clamping force N	Vehicle mass to be decelerated M_h	Speed v	Square speed v^2	Coefficient of correlation r
Clamping force N	1.0	0.29	0	0	0.24
Vehicle mass to be decelerated M_h	0.29	1.0	$3.75·10^{-17}$	$9.09·10^{-18}$	0.32
Speed v	0	$3.75·10^{-17}$	1.0	0.98	0.78
Square speed v^2	0	$9.09·10^{-18}$	0.98	1.0	0.81
Coefficient of correlation r	0.24	0.32	0.78	0.81	1.0

Analyzing the values of the correlation coefficient from Table 4, it is found that the changes in the values of weight wear of friction pads are most affected by the braking start speed (r = 0.81), which proves a strong dependence of Zw on v. have a weak effect on changes in Zw. The correlation coefficient for these variables is in the range of 0.2–0.4.

The thermal imaging tests carried out in parallel with the friction-mechanical tests proved that the contact between the friction pads and the brake disc was uneven, as shown in Fig. 6. After each braking, the friction pads were disassembled and the thermal image was recorded in order to determine the temperature distribution on the pads.



Pads temperature °C			
From the piston rod side		From the cylinder side	
a)	77.0	b)	92.8
c)	73.9	d)	87.2

Fig. 6. Temperature distribution on the brake pads, a) upper left pad, b) upper right pad, c) lower left pad, d) lower right pad, e) View of the set of brake pads after braking from the speed of 120 km/h

The regression model described by relation (10) refers to the determination of the weight wear of a set (4 pieces) of pads after one braking from a given braking speed. Thermal imaging studies have shown that the temperature of the pads is not the same in all friction elements. The maximum temperature value was 93ºC on the upper right friction pad, while the lowest contact temperature of the pad with the disc was 74ºC on the lower left friction pad. In the work [22], the problem of uneven distribution of pad pressures relative to the disc was explained by the change in the geometry of the linkage system and the uneven distribution of the masses of the right and left sides of the linkage mechanism.

5. Conclusions

The article presents the results of tests of the weight wear of friction pads on a certified brake stand for testing disc brakes of rail vehicles. On the basis of friction-mechanical tests, performing 150 brake applications with various combinations, e.g. pressure, speed, mass to be braked, a regression model of weight expressed in grams of friction pad wear after each braking was determined.

Based on the tests and modeling of friction pad wear, it was found:

a) from the parameters of the braking process, the significant increase in the weight wear of the friction pads is most affected by the braking speed, the correlation coefficient of the regression model was 0.80 and the mass to be braked with the correlation coefficient r = 0.32.
b) the pressure of the friction pads against the brake disc has a weak effect on the wear increase of the friction pads with the correlation coefficient r = = 0.24.
c) parameters of the braking process such as the contact surface of the friction pad with the disc and its wear defined by the thickness measurement do not affect the regression model and, consequently, the weight increase of the wear,
d) observation of the surface of the friction pads after each braking with the use of a thermal imaging camera proved the uneven distribution of pressures of the pads constituting the friction pair of the disc brake, which has already been described in other publications.

In further works, it is planned to take into account other variables from the group of parameters of the braking process for modeling the wear of friction pads, such as disc perforation. In particular, this type of brake discs is used in motor vehicles and, despite the improved contact between the pads and the brake disc, it is possible to expect a significant increase in the wear of the friction pads.

Acknowledgements

The investigations were carried out within the Implementation Doctorate Program of the Ministry of Education and Science realized in the years 2021-2025.

Nomenclature

IR	infrared
UIC	International Union of Railways

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