SELECTED ISSUES OF OPERATIONAL USE OF RAIL DISC BRAKE

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Summary

This article describes disc brakes, which in comparison to traditional block brakes are more often used in rail vehicles. Because of numerous advantages in comparison to a traditional air block brake, disc brakes, are more and more often utilized in passenger carriages and other railway vehicles. Stable and constant – in the whole speed range – coefficient of friction μ, with the value: μ =0.35 is a basic advantage of disc brake systems [7]. Attempt to increase train speed, triggers application of greater braking power i.e. braking systems rapidly absorbing and dispersing stored heat energy. For stimulation of cooling processes of friction set (brake disc and friction pad), discs with ventilation canals shaped by various vanes, are used. Thanks to such solution discs take off about 40% [4] of heat generated during braking. However, natural ventilation causes losses of energy for rotation of the disc during ride with switched off brakes, which is especially important for trains with little frequency of brakings and in cases when the profile of the track does not stimulate long-lasting braking.

This article presents selected issues of operational use of rail disc brake such us total resistance generated by brake disc divided into resistance of disc inertia as mass in rotational motion and resistance generated by disc's ventilator, and influence of blocked ventilation canals on braking process.

Keywords: disc brake, resistances of disc inertia in rotational motion, ventilation canals, disc temperature, coefficient of friction

1. Introduction

Operational use of disc brake showed that during braking from high speeds, heat load of braking system appears which as a consequence decreases efficiency of the brake and lengthens braking distance. To cool the disc, internal ventilation canals are used which take off certain amount of heat to the

Environment. However, discs with specifically shaped ventilation canals during rotation consume energy taken by the ventilator. What is more, condition of fast exchange of braking energy into heat, strictly depends on dirt inside ventilation canals and possibly on presence of broken stone or other foreign matters in disc's vane spaces. Concentration of heat energy on the brake disc influences deterioration of braking process, which

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in critical case may lead to loss of braking power. During overhauls of braking system of train cars in accordance with operation-maintenance documentation [2], particular attention is paid to dirt and presence of foreign matters in disc's ventilation canals.

The purpose of this article is to present calculations of resistances of brake disc inertia as mass in rotational motion taking into account resistance generated by brake disc's ventilator [6] and to present results of stationary research of brake disc with closed vane space simulating presence of foreign matters in ventilation canals.

2. Constructional characteristics of ventilated brake discs

Requirements for disc braking systems, such as braking from high speed and realization bigger pressure of friction pads to the disc, are higher because in such conditions, natural cooling is not sufficient to give away heat generated during braking.

a) b) c)

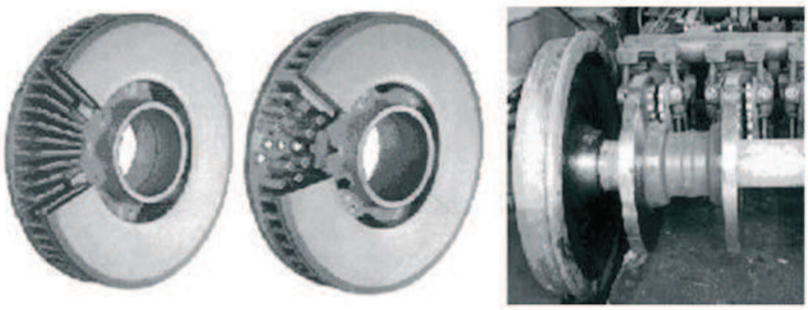

Fig. 1. Brake disc of rail vehicles: a) disc with ventilation vanes, b) disc with ventilation bars, c) solid disc (non-ventilated) [4].

As a consequence, deformation of brake disc appeared and this was connected with necessity to replace the disc before reaching acceptable wear of friction surface. To accelerate giving away heat accumulated during braking, discs with stimulated internal air flow started to be used. For this purpose spokewise arranged cooling vanes (Fig. 1a) or distance bars (Fig. 1b), are used.

Among ventilated discs, discs with ventilation vanes were the first ones to use for a large scale. Structure of such disc consists of two friction rings and ventilation vanes between the rings. From 60 to 100 vanes enable distribution of air, which flowing inside the disc takes away heat from inside of the disc to outside under the influence of centrifugal force.

In canals between the vanes, increase of circumferential speed of the air occurs, which flowing inside the disc takes away heat generated during braking. Friction ring is made as monolith most often of grey cast iron or nodular cast iron and is connected with cast steel hub with 4 or 6 bolts (solution of company Knorr-Bremse), which at the same time enable axial thermal expansion of the disc against the hub, which is presented in figure 2.

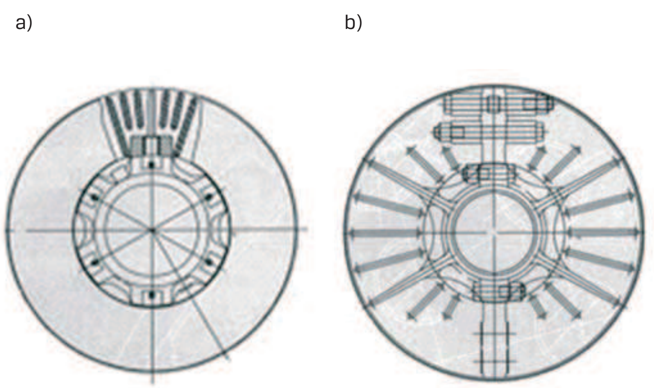

Fig. 2. Disc with ventilation vanes: a) monolithic friction ring, b) friction ring divided with brake hub into two [4].

A disc with ventilation bars is a more innovative solution. In comparison to discs with ventilation vanes, discs with ventilation bars show lower losses of energy resulting from stimulated ventilation without deteriorating the process of distributing heat energy generated during braking process. Such effect was obtained by replacing whirling vanes with densely arranged bars, by which, ventilated space was reduced by 1/3. Dense arrangement of bars in the whole area inside the disc enables good heat conductivity and rotations of the disc take heat outside. Ingested air is decayed by the bars and through continuously changing direction of the flow, takes away accumulated heat energy more effectively.

Discs with ventilation bars are produced in sectional and monolithic form (common solution). It should be mentioned that elements joining discs take some part of internal area reducing intensity of cooling and easiness of rings replacement.

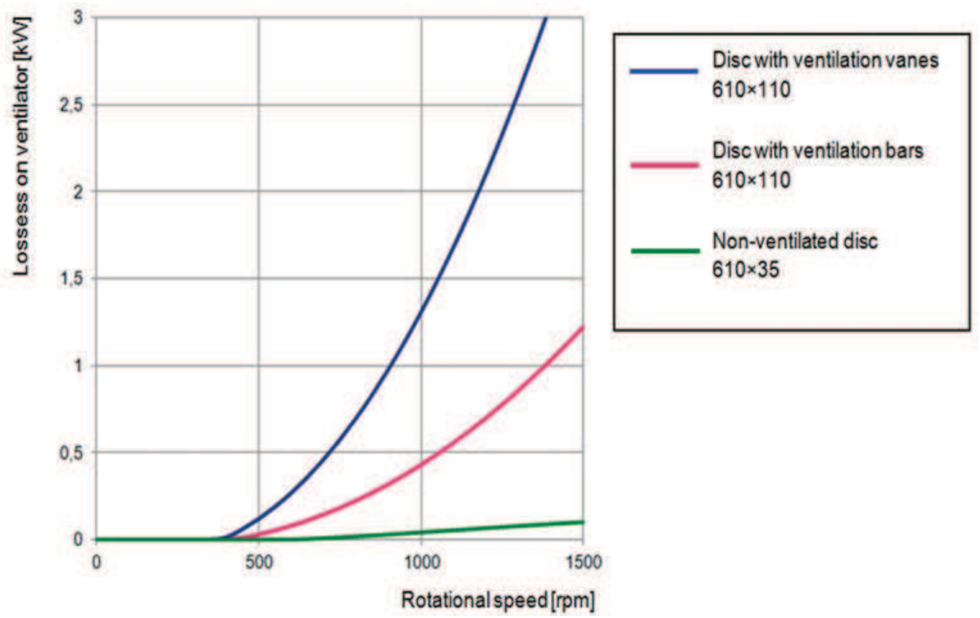

Fig. 3. Energy demand for ventilation process of brake discs [5].

For joining brake disc with the hub a solution of company Knorr-Bremse using 6 bolts is used. Disc constructed is such way is used in vehicles riding and speed of 160 km/h, as well as 200 km/h.

Tests [6] carried out on discs with ventilation bars and vanes showed that discs equipped with bars demand by 60% less energy because of natural ventilation than common discs with ventilation vanes. The tests also showed that in discs with ventilation bars it is possible to obtain by 3% higher average braking power and by even arrangement of bars in the whole disc area, braking heat is more intensively taken away to the environment. This means that despite weaker air pumping, in discs with ventilation bars, heat is taken away more effectively than in discs with ventilation vanes. Thanks to this, disc with ventilation bars is less sensitive to forming thermal cracks during sudden braking. Figure 3 presents comparative histories of the value of energy demand for pumping the air through the ventilator for discs with ventilation vanes and bars, and solid, non-ventilated disc in the function of rotational speed.

3. Method of assessing averaging power of energy dispersion by brake disc considered as solid body in rotational motion

In discs with ventilation vanes and bars, besides energy losses on the ventilator, resistance of disc inertia as mass in rotational motion also occurs. To determine characteristics of power taken by the disc, data from table 1 should be used. Calculus of disc power of certain mass is presented on disc type 610×110 (diameter×width) for one rotational speed.

Table 1. Initial data to calculate resistance of brake disc inertia in rotational motion.

Ordinal No.	Geometrical and thermodynamic value	Designation and values
1	Mass of disc with ventilation vanes	m_f=123,2 kg
2	Mass of disc with ventilation bars	m_p=112,2 kg
3	Mass of non-ventilated disc	m_n=83,4 kg
4	External diameter of ventilated and non-ventilated disc	d_z= 0,610 m
5	Internal diameter of ventilated and non-ventilated disc	d_w=0,193 m
6	Cast iron density	ρż=7200 kg/m^3
7	Steel density	ρs=7850 kg/m^3
8	Linear acceleration	a=1,0 m/s^2
9	Rotational speed of the disc	n=1000 obr/min

Disc volume is calculated according to the following formula:

$$V = \frac{m}{\rho} = \frac{123{,}2}{7200} = 0{,}017\,[m^3]\,, \tag{1}$$

where: m – mass of brake disc in kg, ρ – density of disc's material in kg/m^3.

Substitute thickness of working, annular part of the disc is determined from the following dependency:

$$g_z = \frac{V}{\left(\frac{\pi \cdot d_z^2}{4} - \frac{\pi \cdot d_w^2}{4}\right)} = \frac{0{,}017}{\left(\frac{3{,}14 \cdot 0{,}61^2}{4} - \frac{3{,}14 \cdot 0{,}193^2}{4}\right)} = 0{,}064[m] \tag{2}$$

Moment of inertia of working ring is calculated according to the following formula:

$$I = \frac{1}{2} \cdot m\left(\frac{d_z^2}{4} + \frac{d_w^2}{4}\right) = \frac{1}{2} \cdot 123{,}2\left(\frac{0{,}61^2}{4} + \frac{0{,}193^2}{4}\right) = 6{,}3\,[kg \cdot m^2] \tag{3}$$

Angular velocity of the disc, with assumed constant rotational speed n is determined from the following formula:

$$\omega = \frac{d\varphi}{dt} = \frac{2 \cdot \pi \cdot n}{60} = \frac{2 \cdot 3{,}14 \cdot 1000}{60} = 104{,}7\left[\frac{rad}{s}\right] \tag{4}$$

Angular acceleration of the disc, with assumed constant value of vehicle's acceleration a=1m/s^2, is calculated from the following formula:

$$\varepsilon = \frac{d\omega}{dt} = \frac{a}{\frac{d_z}{2}} = \frac{1}{\frac{0{,}61}{2}} = 3{,}28\left[\frac{rad}{s^2}\right] \tag{5}$$

Time of braking process can be assessed from the following dependency:

$$t = \frac{\omega}{\varepsilon} = \frac{104{,}7}{3{,}28} = 31{,}9[s] \tag{6}$$

Initial, instantaneous kinetic energy of disc's ring is calculated from the following formula:

$$E_k = \frac{I \cdot \omega^2}{2} = \frac{6{,}3 \cdot 104{,}7^2}{2} = 34530{,}6[J] \tag{7}$$

Averaging in time t, power of motion resistance of braking process is determined according to the following formula:

$$N = \frac{dE}{dt} = \frac{E_k}{1000 \cdot t} = 1{,}08[kW] \tag{8}$$

This is calculation of one point of brake disc operation. To define further points of characteristics, identical procedures are used by substituting further values of rotational speed of brake disc. Presented calculus enables to calculate power used by discs of various structure with known mass, geometrical values and density of brake disc's material at constant angular acceleration ж

Figure 4 presents averaging history of resistance of disc inertia in rotational motion. Figure 5 presents the sum of resistances of disc inertia and resistance generated by brake disc's ventilator in rotational motion.

Figure 6 presents percentage of dispersed energy of disc in rotational motion against percentage of total energy of examined system.

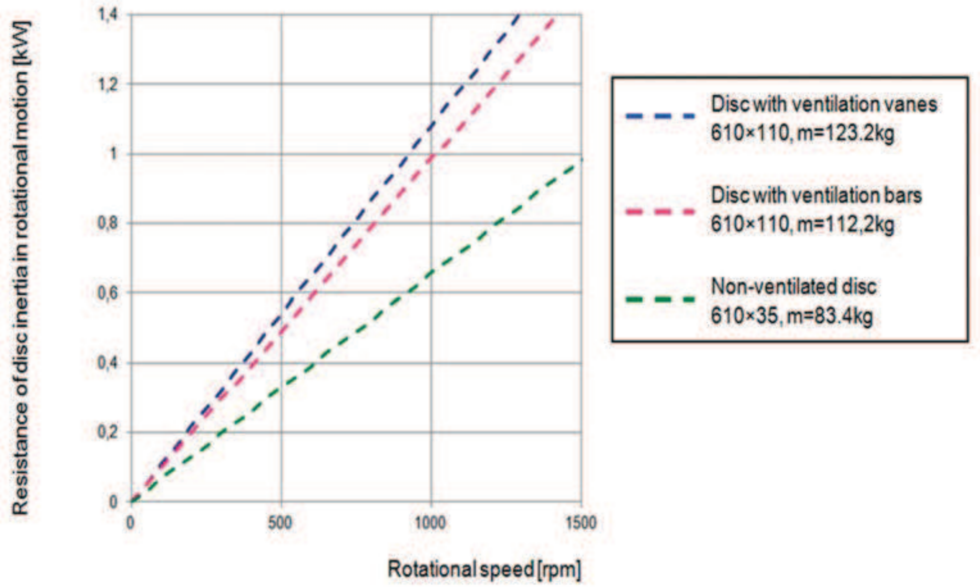

Fig. 4. History of averaging values of resistance power of brake disc motion in rotational motion in function of speed of this motion.

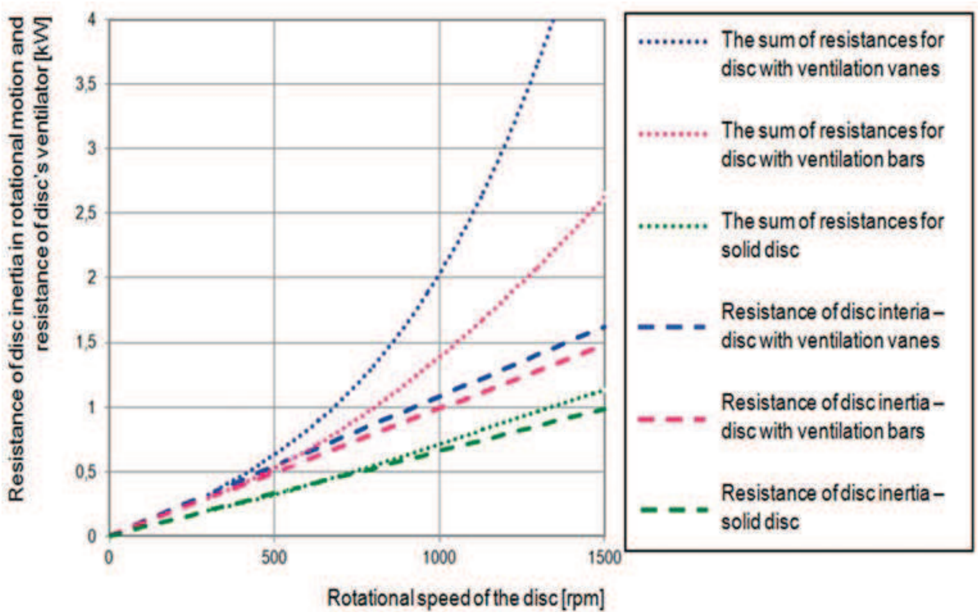

Fig. 5. The sum of resistances of brake disc in rotational motion (resistance of disc inertia in rotational motion and resistance generated by disc's ventilator).

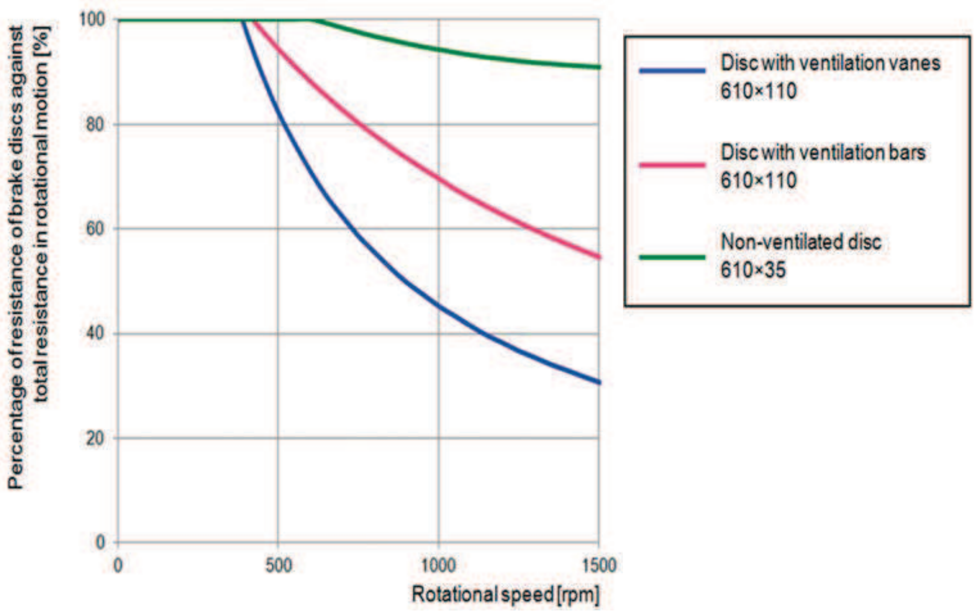

Fig. 6. Percentage of intertia resistance of brake disc against percentage of total resistance (resistance of disc inertia and resistance generated by brake disc).

4. The research on railway disc brake with closed ventilation canals of brake disc

4.1. Methodology and object of research

The tests were carried at an inertial station for testing rail vehicles brakes. A brake disc type 610x110 with ventilation vanes and a set of matched brake pads type 200 FR20H.2 35 mm thick were the object of the test. Research program C (fast drive) was applied for the tests, according to [1] brakings were performed at speed of 120, 160 and 200 km/h, with pad pressure of 44 kN to the disc and braking mass of 7,5 t per disc. Disc temperature in the whole range of braking time was registered by six thermocouples; three of them were mounted on two sides of the disc and placed every 120^{o} on three rays. The presence of foreign matters in the area between the vanes was simulated by placing a band clamp on ventilation vanes. The way how the ventilation areas were covered is presented in Fig. 7.

Before main tests were carried out a series of identification tests had been made on the disc with open ventilation canals, thanks to which undisturbed flow of cooling was provided. For each speed at the beginning of braking eight repetitions of braking were performed.

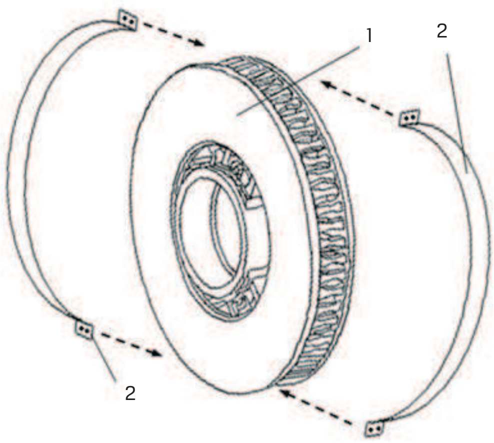

Fig. 7. Placing a band clamp on brake disc type 610×110: 1-brake disc, 2- right and left part of the band clamp.

4.2. Analysis of tests results

During tests the following parameters were measured: disc temperature at stoppage Fig. 8, average friction coefficient Fig. 9 and time of disc cooling to 60 ºC Fig. 10. Disc cooling was realized by simulating a car ride at 100 km/h.

Brakings of a disc with covered ventilation vanes could lead to a change in the structure of the material as a result of strong thermal load. On the friction surface of the disc, overheating occurred in a form of two rings in the area of inner and outer diameter of the disc. Discoloration of the friction area was observed yet after six brakings (Fig. 11).

During brakings on the disc with covered vane area, on one of the thermocouples overflow of temporary disc temperature over 400 °C was registered and reached scope between 403÷417 °C. According to [1] brake discs of rail vehicle during brakings should not reach temporary temperature over 400 °C, because this causes disc deformation and loss of required resistance and flexibility.

5. Conclusion

This article presents two aspects of operational use of rail disc brake. The first aspect refers to energy dispersed by brake disc's ventilator, the second aspect refers to influence of possibly existing ventilation canals on braking process.

Calculus presented in this article enables to assess averaging power of resistance of disc inertia as mass in rotational motion. Thanks to this, division of total disc's resistance into

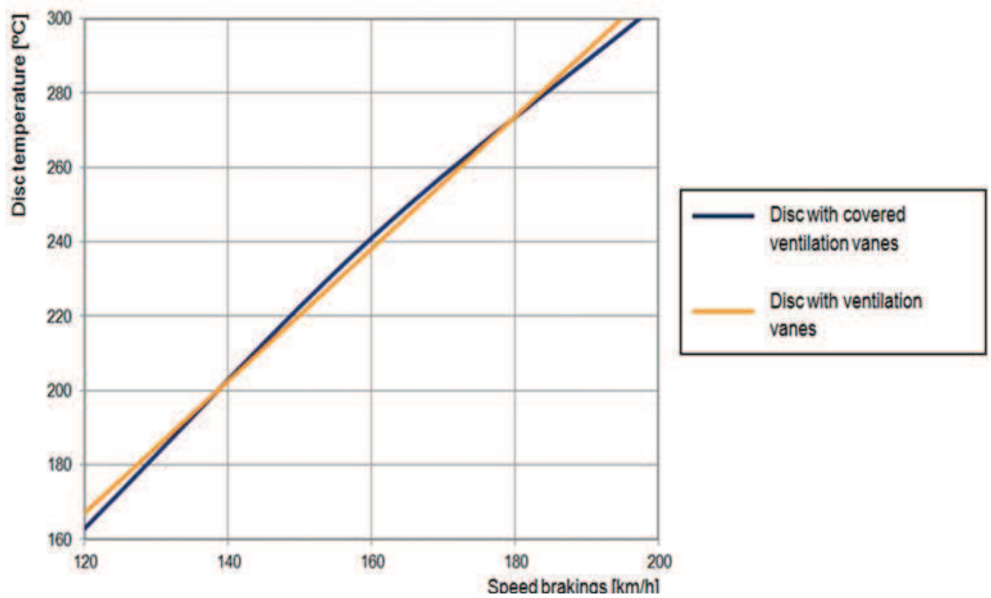

Fig. 8. Distribution of average temperature of brake disc type 610×110.

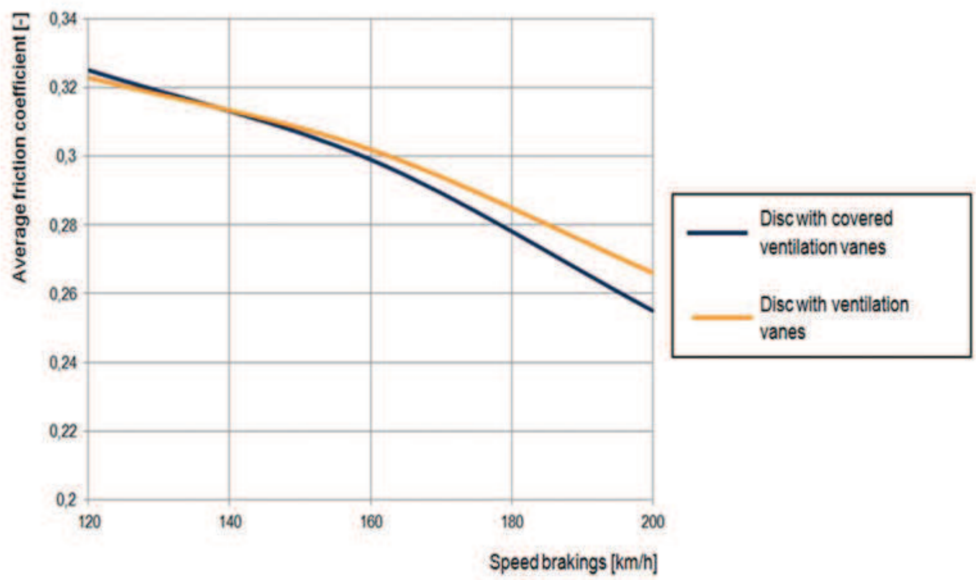

Fig. 9. History of average coefficient of friction.

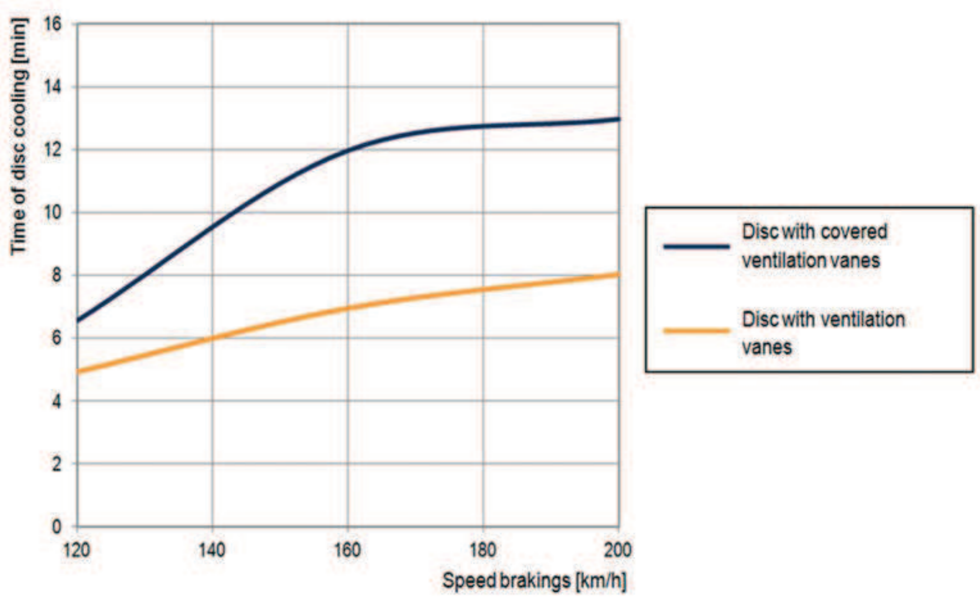

Fig. 10. History of cooling time of brake disc during simulated ride of train car at constant speed of 100km/h (disc rotational speed of 600rpm).

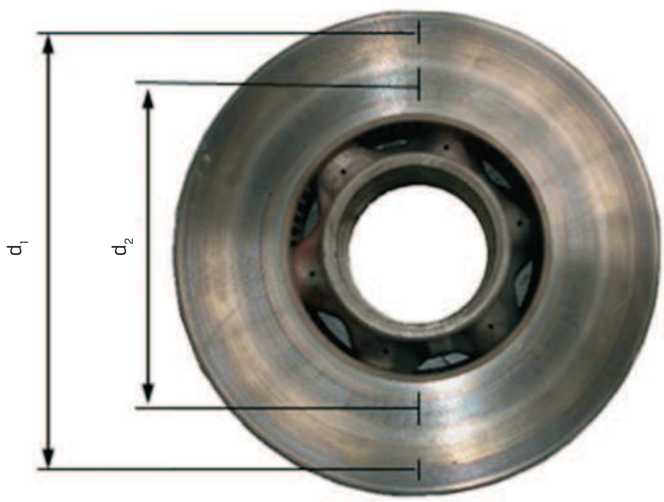

Fig. 11. The disc after series of brakings with covered ventilation vanes with visible surface overheating in a form of two rings of various diameters.

resistance of disc inertia as mass in rotational motion and resistance of disc's ventilator during pumping air for the process of disc cooling, is possible.

Analyzing graphs presented in figure 5 it can be found out that despite the same dimension of discs (610×110), disc with ventilation bars produces by 9% lower resistance of disc inertia than disc with ventilation vanes in the whole range of rotational speed.

The biggest portion of resistance of disc inertia as mass in rotational motion occurs at 450 rpm for ventilated disc and up to 600 rpm for solid disc. Further increase of rotational speed increased losses on disc's ventilator. Percentage of losses stemming from rotation of disc of a certain mass against total losses decreases from 100% to 30% for disc with ventilation vanes and from 100% to 54% for disc with ventilation bars. In solid discs, mass losses constitute from 90% to 100% of total losses depending on rotational speed.

After stationary tests [5] it was found that covering the vane area and simulating the presence of foreign matters between vanes has no substantial influence on temperature measured at stoppage and the value of coefficient of friction in comparison to a disc with undisturbed ventilation. The changes in temperature and coefficient of friction are contained in statistical error. The change in disc temperature with open and closed vane area has not been noticed, which may stem from long period of breaking heat discharge into the environment against time of singular braking, what was described in [3].

However, covering vane has significant influence on cooling time of the disc after braking. A disc ventilated during car ride with a simulated speed of 100 km/h cools by 25÷40% faster than a covered disc depending on speed at the beginning of braking.

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