

# Modelling and Analysis of Thermal and Stress Loads in Train Disc Brakes – Braking from 250 km/h to Standstill

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*Thermal and stress analysis of disc brakes under specific loads (driving downhill and braking to a standstill) was calculated. The FEM (Finite Element Method) was used to carry out the analysis. The analysis dealt with centrifugal load for two cases of braking, braking to a standstill on a flat surface and braking downhill, maintaining constant speed and afterwards braking to a standstill. The main boundary condition in both cases was the entered heat flux on the braking surface of the disc and the force of the brake clamps. Two different discs were used, one brand new (unused) and one with permitted wearing.*

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## 0 INTRODUCTION

To increase the average travelling speed of Slovenian railways major reconstruction of the existing network must be performed. For this purpose the railway vehicles must also be suitable. The most important part of the vehicle is the braking system. Today the railway vehicles use disc brakes. Braking from 250 km/h to a standstill requires good, reliable brakes. Different loads are applied to the disc during braking. Centrifugal, thermo-elastic, friction and brake clamp loads affect the brake disc at the same time. Fast and hard braking of a whole train of cars causes a rise in temperature because the loads and the heat flux rise up instantly. The difference in temperature causes large stresses in the material – the consequence is thermal shock. Rapid aging and fatigues are the result. With correct design and correct choice, two major goals have to be assured:

- safe braking,
- reliability of brakes during all working regimes.

Wrong design and selection of the production procedure can lead to catastrophic results [2] and [6]. An overview of past research showed that brake discs are mostly tested for thermal loads and their effects (thermal cracks, thermal deformation). Besides that, there is great stress on brake pads.

The purpose of our model and analysis was to determine the effect of thermal loads on the temperature field and the effect of centrifugal load in a specific brake disc. To determine the

boundary conditions, parts of car brake disc calculation were used. This assumption can be applied because the working physical principal is the same.

The main goal of this analysis was to define a model for the thermal and centrifugal load. With this model all the necessary parameters (stresses as a consequence of thermal and centrifugal loads), defined by the maker, would be calculated.

## 1 NUMERICAL CALCULATION OF THE DISC BRAKE MODEL

The disc brake is symmetrical, therefore the model includes 1/12<sup>th</sup> of the whole disc brake. Two discs were considered, one disc was brand new (unused) and one had permitted wearing of 7 mm on both sides of the disc. The loads, used for this analysis are divided into three groups:

- centrifugal load at a constant temperature,
- centrifugal and thermal load combined during braking on a downhill track,
- centrifugal and thermal load combined during braking on a downhill track and afterwards braking to a standstill on a straight, horizontal track.

The material of the disc brake is spherical graphite, defined according to SIST EN 1563:1988 and with the characteristics according to EN-GJS-500-7 (EN-JS 1050) with surface roughness of  $R_a = 3.2 \mu\text{m}$ . Disc brakes were machined on CNC machine tool with cutting conditions which were previously optimized by an intelligent optimization software [11]. Surface roughness and cutting forces acting on the disc

during machining were kept constant by continuous adaptation of cutting parameters [12] to current machining conditions.

The disc brake was screwed to the hub. The hub was press fit onto the axle of the vehicle. Only one braking cycle was taken into consideration.

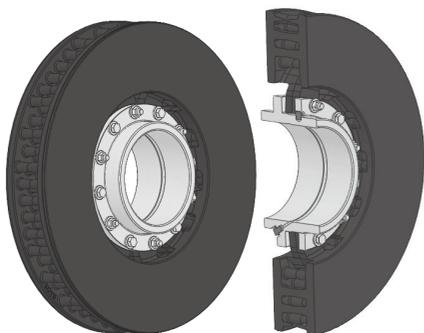


Fig. 1. Brake disc assembly

### 1.1 Modelling and Preparing the 3D Model of the Disc Brake

The main shape and geometry are shown in Figure 2. The worn disc is 14 mm thinner (7 mm on each side) than the new disc.

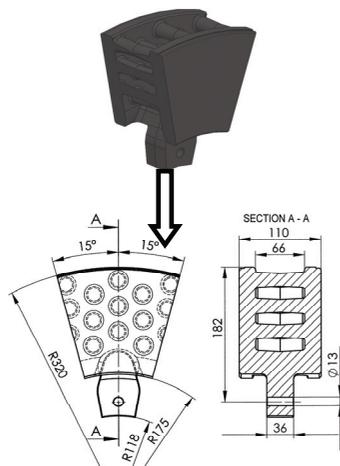


Fig. 2. Part of worn disc

### 1.2 Load Determination

The disc was analyzed for three different load cases:

- centrifugal load at constant speed and constant temperature of the surrounding area,

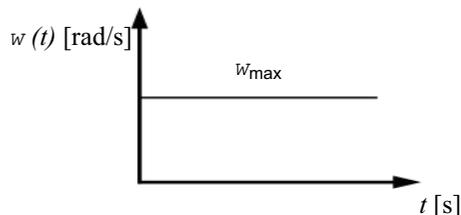


Fig. 3. Centrifugal load

- braking from the maximum velocity of 250 km/h to a standstill with the temperature of the surrounding area constant at 50 °C,

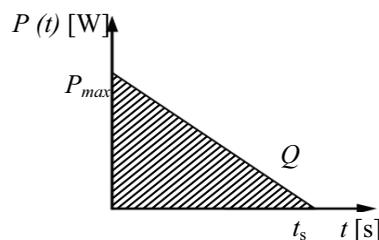


Fig. 4. Heat input during braking to a standstill

- braking from the maximum velocity of 250 km/h on a downhill track, maintaining constant speed and afterwards braking to a standstill. The temperature of the surrounding area is constant at 50 °C.

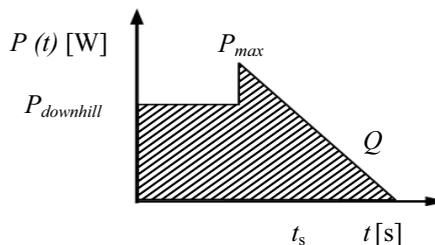


Fig. 5. Heat input in case of maintaining constant speed

In our analysis a brake cycle was dealt with, taking into consideration that the disc had already been warmed up.

Braking on a horizontal track was the most challenging load case. Because of speed reduction a smaller heat transfer coefficient (10 W/(m<sup>2</sup> K)) was considered.

To maintain a constant speed during descent on a downhill track and considering convection, also the medium value for the heat transfer coefficient (100 W/(m<sup>2</sup> K)) was considered. In this part of the analysis the disc was checked for temperature arrangement. On the part of the track with the most disadvantageous (11% inclination) the prescribed temperature of 350 °C was not exceeded. This temperature was not reached even with subsequent braking to a standstill. Both coefficients for the heat transfer were considered.

In all three cases, the air humidity and also the influence of the heat radiation was not considered.

### 1.3 Determination of the physical Model

The data, used to run the analysis, are shown in Table 1.

Table 1. *Material properties*

Mass of the vehicle – $M$ [kg]	70 000
Maximal load per axle [kg]	17500
Number of axles per vehicle	4
Number of discs per axle	3
Start speed – $v_0$ [m/s]	70
Deceleration – $a$ [m/s <sup>2</sup> ]	1.4
Braking time – $t_s$ [s]	50
Effective radius of the disc brake – $r_{disc}$ [m]	0.247
Radius of the wheel – $r_{wheel}$ [m]	0.460
Friction coefficient disc/pad – $\mu$ [/]	0.4

#### Angular velocity:

The consequences of centrifugal loads (caused by the high travelling speed of the vehicle) applied to the disc were determined and calculated in the analysis. The results were stresses. To begin the calculation, the angular velocity was needed:

$$\omega = \frac{v}{r} = 151 \text{ [s}^{-1}\text{]} \quad (1)$$

#### Heat flux:

A physical model, considering the heat flux in dependence of the braking time, was used to determine the brake influence. Values for calculation were taken from Table 1. The vehicle has one front and one back bogie. An assumption was made that during the braking, a part of the whole weight is distributed in a ratio

of 60/40 to the front bogie [3]. Because of that, the disc brakes on the front bogie take over most of the weight and braking energy and that is why only the disc from the front bogie was analyzed.

In consideration of the weight distribution and the fact that the bogie consists of two axles, one brake disc from the front bogie carries 10% of the whole braking force, where one disc from the back bogie carries only 6.67% (Figure 6).

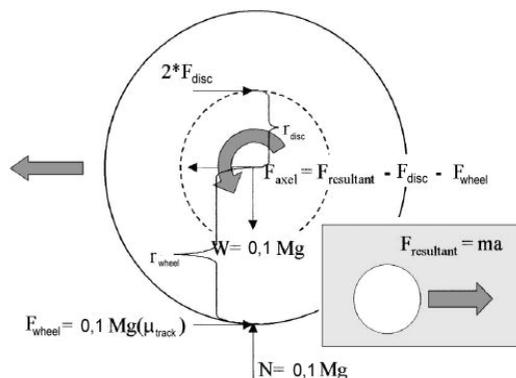


Fig. 6. *Representation of forces acting on wheel and disc brake, where  $M$  represents the mass of rail car and  $m$  represents mass of disc and wheel value 0.1 stands for 10% of weight distribution*

The kinetic energy for one wheel (disc brake) is equivalent to the energy balance [3]:

$$0.1 \cdot \frac{1}{2} \cdot M \cdot v_0^2 = \int_0^{t_s} P(t) dt = 2 \cdot F_{disc} \int_0^{t_s} v_{disc}(t) dt \quad (2)$$

The energy change in the disc at the moment is equal to the heat flux on the surface of the disc. The Eq. 2 is valid in the case of a constant braking deceleration. The braking force on the disc is equal to Eq. (3) [3]:

$$F_{disc} = \frac{0.1 \cdot \frac{1}{2} \cdot M \cdot v_0^2}{2 \cdot \frac{r_{disc}}{r_{wheel}} \cdot \left( v_0 \cdot t_s - \frac{1}{2} \cdot a \cdot t_s^2 \right)} = 9125.5 \text{ [N]} \quad (3)$$

The heat flux at the moment, which affects one half of the disc, is calculated according to the Eq. (4):

$$\begin{aligned} \dot{Q}(t) &= F_{\text{disc}} v_{\text{disc}}(t) = F_{\text{disc}} \frac{r_{\text{disc}}}{r_{\text{wheel}}} (v_0 - a \cdot t) = \\ &= 343000 - 6860t \quad [\text{W}] \end{aligned} \quad (4)$$

In case of braking on a horizontal track to a standstill, the analysis was carried out in 26 steps, each step being 2 s long.

In case of braking on a downhill track, a physical model was used to determine the heat flux in dependence of the changing potential energy. The vehicle descended on a downhill track and maintained a constant speed of 250 km/h. Consequently, the heat flux entering into the disc was constant. Considering the heat flux distribution in 12 discs (3 discs per axle, 2 axles per bogie), the whole energy balance is equal to the following equation:

$$M \times g \times \Delta h = \dot{Q} \times t_s \quad (5)$$

Considering the trigonometry, and the boundary condition that the vehicle should move at a constant speed, the braking power for one disc from the front bogie was obtained:

$$\dot{Q} = \frac{M \cdot g \cdot v_0 \cdot \sin \delta}{10} = 42332 \quad [\text{W}] \quad (6)$$

In Eq. 6, the value 10 represents 10% of weight distribution.

In case of braking on a downhill track and maintaining a constant speed, the analysis was carried out in 52 steps, each step being 2 s long. Additional 26 steps were added to simulate braking and stopping on a horizontal track. Altogether this case included 78 steps.

### Determining the pressure of the brake clamps

The brakes work on the brake disc by means of pneumatic system. The surface pressure between the disc brake and the brake pads was determined on the basis of the calculated braking force. In case of braking on a horizontal track, the pressure was:

$$p = \frac{F_{\text{disc}}}{A_c \cdot \mu} = 1.14 \quad [\text{MPa}]. \quad (7)$$

In case of braking downhill, the heat flux power for one disc was considered. The braking force on one disc was:

$$F_{\text{disc}} = \frac{\dot{Q} \cdot \mu}{2 \cdot v_0} \cdot \frac{r_{\text{wheel}}}{r_{\text{disc}}} = 233 \quad [\text{N}]. \quad (8)$$

In this load case, the surface pressure was 0.03 MPa. These two boundary conditions were considered in both cases (braking on a horizontal and downhill track). Because of their low values, they could be disregarded.

### 1.4 Determining of Loads, Fixing, Mesh and Material Properties

**Loads:** the symmetrical boundary condition on the edge of the selected section was modelled with slide supports in radial direction. The heat flux did not flow thru those supports.

**Fixing:** fixing was made in the points, where the disc was mounted (screwed) onto the hub (Fig. 7).

**Mesh:** the creation of a mesh volume was conducted automatically by the software package Abaqus CAE 6.7.1. The mesh consists of 84354 tetrahedral elements (element code C3D4AT – allows linear thermo – deformational analysis). The average size of the elements is 6mm and the number of nodes is 17639.

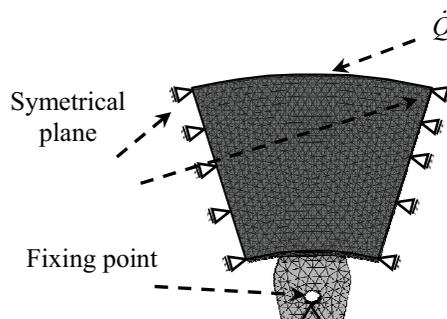


Fig. 7. Load, fixing and mesh of the selected section

**Material:** for the analysis of the discs certain physical properties of materials, given in Table 2, were required.

Table 2. Material properties

Heat conductivity – $\lambda$ [W/mK]	35.2
Density – $\rho$ [kg/m <sup>3</sup> ]	7100
Specific heat – $c_p$ [J/kgK]	515
Module of elasticity – $E$ [MPa]	169000
Poisson number – $\nu$ [/]	0.275

## 2 ANALYSES OF RESULTS

### 2.1 Analysis of Centrifugal Loads

In the first analysis the impact of centrifugal loads was taken into account. The analysis refers to the maximum speed of  $v = 250$  km/h. Based on the analysis made, the comparative stresses according to Mises were obtained. The results for sections of both discs are shown in the Figs. 8 a and b.

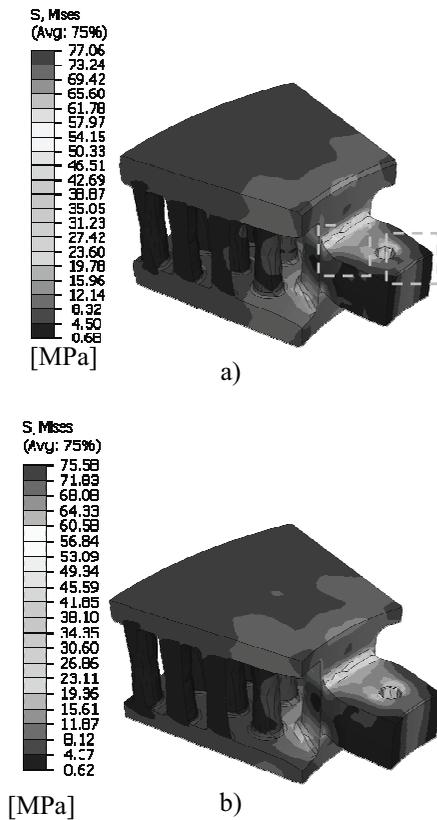


Fig. 8. Stress field of centrifugal loads of a) new disc, b) worn disc

The biggest stresses are located in the spots, where the disc is screwed onto the hub and where the disc passes into the fixing eye (marked spots in Fig. 8). The stress values rise to 77.06 MPa for a new disc and to 58 MPa for a worn disc.

The maximum displacement in a new disc is 0.0119 mm whereas in a worn disc this value is 0.0126 mm (Fig. 9).

### 2.2 Analysis of Thermal Load

This analysis of the disc was implemented for two examples; braking on a horizontal track to a standstill and maintaining the downhill speed and afterwards braking to a standstill. Following temperatures were taken into account; the initial temperature of the disc, disc fixture and the ambient temperature, all equal to 50 °C.

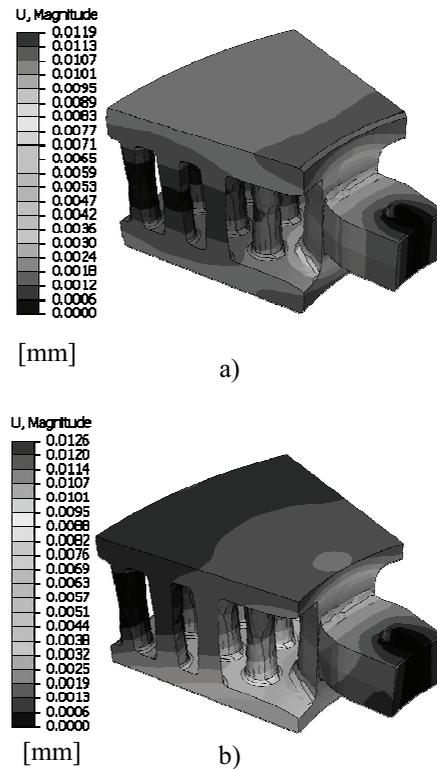


Fig. 9. Deformation field of centrifugal loads of a) new disc, b) worn disc

In case of braking on the horizontal track, the surface temperatures of the new disc after 30 s are 174 °C. Given the lesser and lesser input of the heat flux, the temperature of the disc decreased (lower speed) to a final temperature of 154 °C (after 52 s).

It was taken into account that the temperature of the cooling ribs and channels did not notably rise during the period of one braking. This fact comes from the limited heat conduction speed of the cast material.

For the worn disc with equal load after 38 s the highest temperatures amounted to 211 °C, but not on the entire surface of the disc. They occurred in areas where the two parts subject to load (disc rims) were not connected to cooling ribs. In that case the ribs received much more heat, since the heat conduction is more intensive due to smaller thickness of disc rims. Therefore, their temperature slightly increased, i.e., for 30 to 40 °C. After expiration of 52 s the temperature was equal to 211 °C (Fig. 10 b).

In case of braking downhill the highest temperature on the new disc after expiration of 104 s was equal to 154 °C, namely on the disc - brake pad contact surface (Fig. 11 a). During further braking to a standstill on the straight track the brake disc warmed up to the highest temperature of 251 °C within 132 s after the start of braking (Fig. 11 b).

For the worn disc the highest temperatures after 104 s of maintaining constant speed during braking downhill amounted to 166 °C (Fig. 12 a). By further stopping on flat surface the disc temperature increased to the maximum temperature of 298 °C within 32 s after the start of stopping on a flat surface (Fig. 12 b).

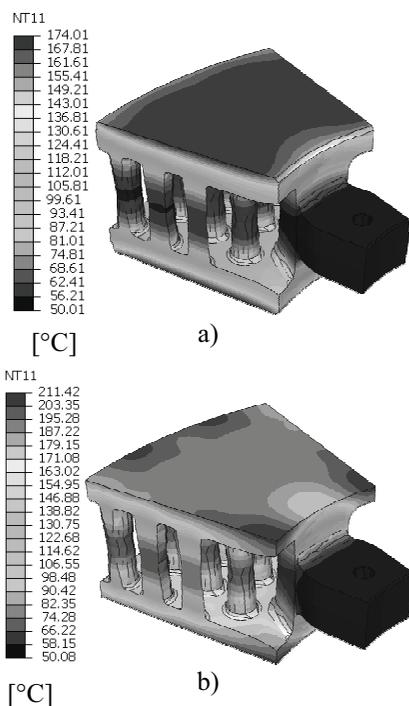


Fig. 10. Temperature field, braking on straight track a) new disc, b) worn disc

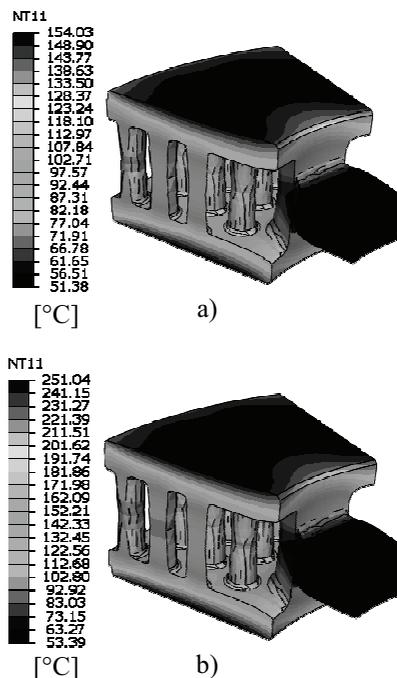


Fig. 11. Temperature field of a) new disc – braking downhill b) stopping on flat surface

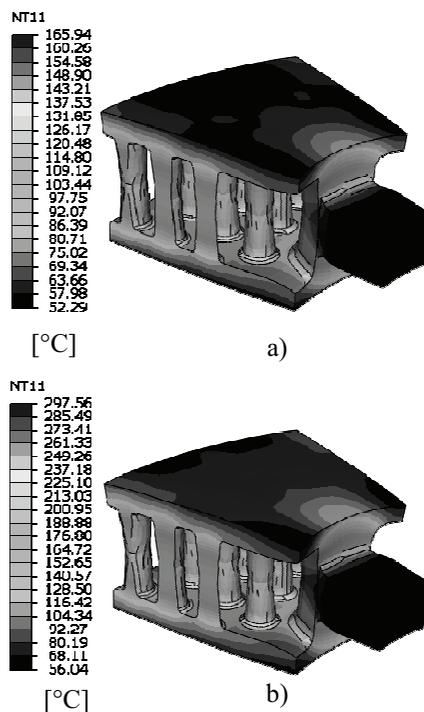


Fig. 12. Temperature field of a) worn disc – braking downhill b) stopping on flat surface

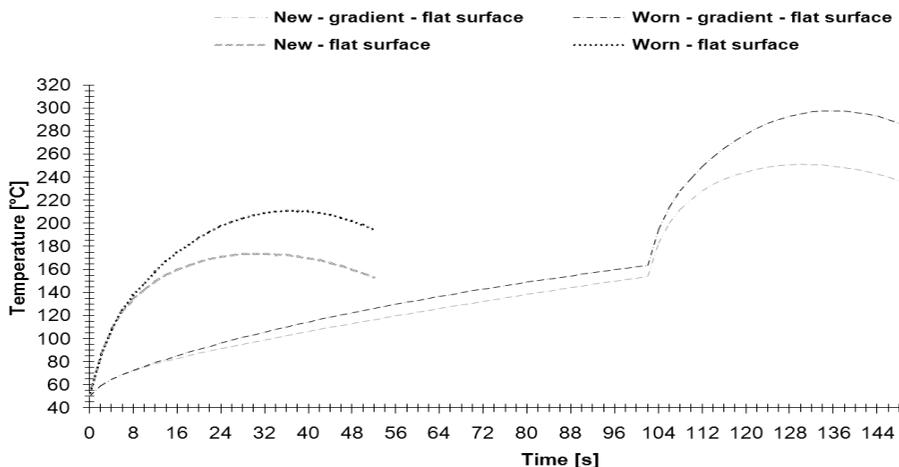


Fig. 13. Curves of max. temperatures of new and worn disc. The two left curves represent braking from 250 km/h to a standstill, the two right curves represent braking downhill and afterwards braking to standstill on a flat surface

The most loaded area is located immediately under the brake pad where the temperatures are the highest.

It can be observed that towards the point of fixing, the temperatures drop fast so that that point is thermally not loaded and also the effect of thermal loads on the hub itself is negligible, which is important for the press fit. Fig. 13 shows the comparison of curves of maximum temperatures of both load cases.

### 2.3 Analysis of Stress Loads

Thermal stresses appear due to the rise of the disc working temperature. However, in addition to thermal loads themselves also the effect of centrifugal loads on the disc brake and the effect of mechanical pressure of brake clamps were considered. The purpose of analyses was to determine the effect of centrifugal loads on the disc compared with thermal loads themselves. The stated stresses are comparison stresses according to Mises.

In case of braking on flat surface, by considering the centrifugal loads in the new disc, the highest stresses amounted to 185 MPa, namely beside the point of transition of fixing teeth. At the points with the highest heat load the stresses were equal to 185 MPa (Fig. 14 a).

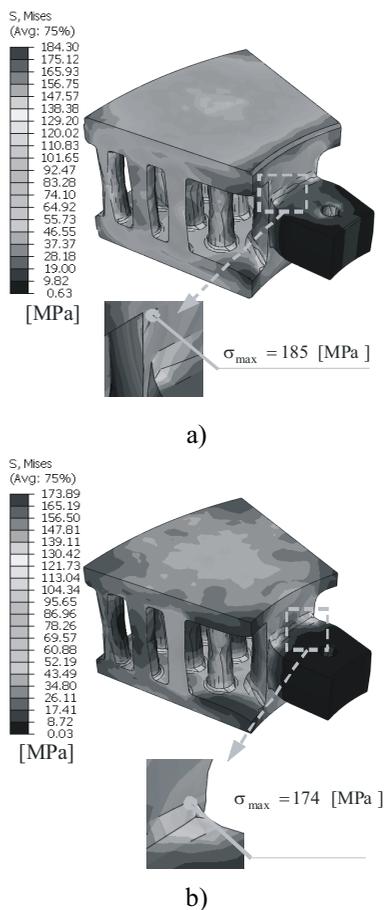


Fig. 14. Stress field of disc braking on flat surface a) new and b) worn

With that loading (braking on a flat surface) the highest stresses for worn disc were equal to 174 MPa. Also here, the highest stresses appeared at the points of transition of fixing teeth (Fig. 14 b). The largest disc displacements during braking on a straight surface amounted to 0.1536 mm for the new disc and 0.2288 mm for the worn disc.

In case of braking downhill and later stopping on a flat surface the highest stresses amounted to 201 MPa for the new disc, which is beside the point of transition of fixing teeth (Fig. 15 a). They occurred at the stage of stooping after expiration of 42 s after the start of stopping.

When analyzing the worn disc, the largest stresses were equal to 198 MPa (Figs. 15 a and 15 b). Also here, they occurred at the points of transition of fixing teeth, but slightly sooner (after expiration of 34 s after the start of stopping).

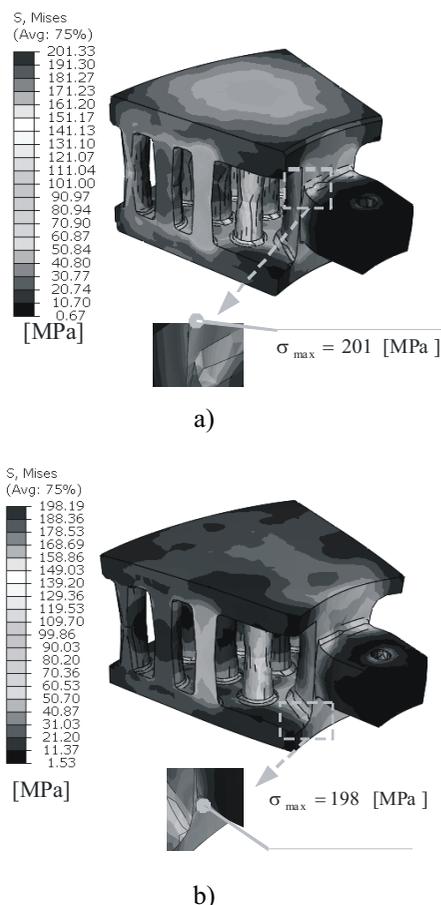


Fig. 15. Stress field of new a) and worn b) disc – braking downhill with stopping on flat surface

At the end of maintaining constant speed the highest stresses were lower. They amounted to 161 MPa for the new disc and 115 MPa for the worn disc.

### 3 DISCUSSION OF RESULTS

The stresses themselves are high, but in comparison with permissible stresses amounting to 210 MPa by considering the 1.5 safety factor the analyzed disc is adequately dimensioned. Table 3 shows the effect of centrifugal loads – the analysis was carried out for the most unfavourable operating case, i.e., the highest speed of the railway vehicle equals 250 km/h.

Table 3. Highest values of result for analyses of centrifugal loads

Numerical analyses		$\sigma_{max}$ [MPa]	Displacement [mm]
Disc brake	Worn	77	0.0119
	New	76	0.0126

Due to the brake heating up during braking, it is obvious that the effect of the heat flux is outstanding, since the highest values of temperature fields occurred in the worn disc during braking downhill with later stopping on flat surface. They reached 298 °C within the time of simulated braking 136 s. The highest allowable temperatures in the brake pad and, consequently, in the disc were 350 °C (longstanding). The results of stress analyses show that by considering the centrifugal loads, the stress increased by 10 to 20 MPa. The highest stress value was 201 MPa and was also smaller than the permissible stress for the material of the disc. In all cases the effect of centrifugal loads on the disc was small in comparison to thermal loads. In Table 4 the highest values of analyses are summarized.

Table 4. Highest values of results for analyses of combined loads

Numerical analyses		T [°C]	$\sigma_{therm,med}$ [MPa]
Flat surface	New	174	185
	Worn	211	174
Downhill + Flat surface	New	251	201
	Worn	298	198

The highest values of stresses of concerned stress fields occurred during braking downhill with stopping on a flat surface. This is due to the fact that in this case the temperature difference between the warmest and coldest part is much higher than in case of braking on a flat surface only.

In order to reduce stresses and to improve the construction of the disc brake, two improvements are recommended:

- selection of another material with better mechanical properties,
- modification of transition and radii, where stress concentrations occur.

Whoever order the research will compare the results of numerical analyses also with experimental results, which is at the same time a recommendation for further work.

#### 4 CONCLUSION

The temperatures, to which the disc heats up and the stresses, are high. Concerning the given design of the disc, the material and the results of analyses of thermal and centrifugal loads, it can be claimed that the selected model of the disc brake is adequate and that the set target of the analysis meets the requirements of the disc brake makers internal standards. However, the analysis has not considered the effects of shear stresses, residual stresses and the effect of service life with cyclic load, which the disc material must daily support without breaking down. The new trend of equipping railway vehicles is orientated towards replacing the grey cast iron brake discs by brake discs from composite materials. Reduced sound barriers during braking and higher resistance to temperatures are two major advantages of composite materials.

#### 5 REFERENCES

- [1] Ren, Z., Ulbin M., FEM practical course for Abaqus: Faculty of mechanical engineering Maribor, 2002. Acquired 13.04.2009 at [http://lace.uni-mb.si/Num\\_meth\\_konst\\_ABAQUS\\_6.7.1](http://lace.uni-mb.si/Num_meth_konst_ABAQUS_6.7.1)
- [2] ECONOMIC expert, List of rail accidents. Acquired 14.05.2009 at <http://www.economicexpert.com/2a/List:of:railway:disasters.htm>
- [3] Mackin, T.J. (2002) Thermal cracking in disc brakes. *Engineering Failure Analysis*, vol. 9, p. 63-76.
- [4] Panier, S., Dufrenoy P., Weichert D. (2004) An experimental investigation of hot spots in railway disc brakes, *Wear*, vol. 256, no. 7-8, p. 764-773.
- [5] SIST EN 1563:1998. Founding – Spheroidal graphite cast irons. SIST, Ljubljana 1998.
- [6] Weitz&Luxenberg P.C., Recent Train Accidents. Acquired 14.05.2009, at [http://www.weitzlux.com/trainaccident\\_4533.html](http://www.weitzlux.com/trainaccident_4533.html)
- [7] Yevtushenko, A.A., Kuciej, M. (2009) Frictional heating during braking in a three - element tribosystem, *International Journal of Heat and Mass Transfer*, vol. 52, no. 13-14, p. 2942-2948.
- [8] Zhen-cai Z., Yu-xing P., Zhi-yuan S., Guo-an C. (2009) Three-dimensional transient temperature field of brake shoe during hoist's emergency braking, *Applied Thermal Engineering*, vol. 29, no. 5-6, p. 932-937.
- [9] Bagnoli, F., Dolce, F., Bernabei, M. (2009) Thermal fatigue cracks of fire fighting vehicles gray iron brake discs, *Engineering Failure Analysis*, vol. 16, Iss. 1, p. 152-163.
- [10] Župerl, U., Čuš, F. (2008) Machining process optimization by colony based cooperative search technique. *Strojniški vestnik - Journal of Mechanical Engineering*, vol. 54, no. 11, p. 751-758.
- [11] Čuš, F., Župerl, U., Kiker, E. (2007) A model - based system for the dynamic adjustment of cutting parameters during a milling process, *Strojniški vestnik - Journal of Mechanical Engineering*, vol. 53, no. 9, p. 524-540.
- [12] Potrč, I., Lerher, T., Šraml, M., Šamec, B., Oder, G., Expert advice on the levels of tensions in A, B, C, D and M versions of brake discs, Maribor 2008.