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Theoretical and Numerical Approach in Determining the Thermal and Stress Loads in Train Disc Brakes

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This paper shows a thermal and tension analysis of a brake disc for railway vehicles. The FEM (Finite Element Method) was used to carry out the analysis. The analysis deals with one cycle of braking – braking from maximum velocity to a standstill, cooling off faze and then accelerating to maximum velocity and again braking to a standstill. This case of braking represents a part of a railway working conditions. The main boundary condition in this case was the entered heat flux on the braking surface, the centrifugal load and the force of the brake clamps. One type of disc was used - with permitted wearing.

Teorijski i numerički pristup određivanju toplinskih i mehaničkih naprezanja disk kočnica na vlakovima

Izvornoznanstveni članak

U radu je prikazana analiza toplinskih i mehaničkih naprezanja na disk kočnicama tračničkih vozila, a provedena je primjenom FEM (Finite Element Method) metode. Analizom se tretira jedan ciklus kočenja – kočenje od najveće brzine do zaustavljanja, faza hlađenja, a zatim ubrzanje do najveće brzine te ponovno kočenje do zaustavljanja. Takvim ciklusom kočenja predstavljen je jedan od radnih uvjeta. Pri takvom pristupu glavni granični uvjeti su toplinski tok na površinu kočnice, centrifugalno opterećenje i sila stezanja. Analiziran je jedan tip diskova - s dopuštenim trošenjem.

1. Introduction

The most important part of the vehicle is the braking system. Today the railway vehicles use disc brakes. Braking from 200 km/h to a standstill requires good, reliable brakes, capable of fast response times and durability. 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. The main problem of braking and stopping a heavy train system is the great input of heat flux into the disc in a very short time. Because of high temperature difference the material is exposed to high tensions. Heat shock, rapid aging and fatigues are the results. To prevent this from happening, analysis need to be made. The problem can be solved only by applying a non stationary and numerical calculation. The analysis is carried out for one model of the disc (Figure 1) and for one large load.

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 and also environmental conditions can lead to catastrophic results [2-4] and [8]. An overview of past researches 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.

Symbols/Oznake

v	- speed, m/s - brzina
t	- time, s - vrijeme
T	- temperature, °C - temperatura
M	- mass of the vehicle, kg - masa vozila
F	- force, N - sila
F_{disc}	- braking force on the disc, N - sila kočenja na disku
v_{disc}	- maximum velocity of the disc, m/s - maksimalna brzina diska
r_{disc}	- radius of the disc, mm - polumjer diska
r_{wheel}	- radius of the wheel, mm - polumjer kotača
a	- acceleration, m/s^2 - ubrzanje
$\dot{Q}$	- heat flux, W - toplinski tok
p	- pressure - tlak
A_c	- surface area of the brake pads, m^2 - površina kočnih pločica
m	- friction coefficient - koeficijent trenja
w	- angular velocity, s^{-1} - kružna brzina
λ	- heat conductivity, W/m·K - toplinska vodljivost
ρ	- density, kg/m^3 - gustoća
c_p	- specific heat, J/kg·K - specifična toplina
E	- module of elasticity, MPa - modul elastičnosti
v	- Poisson number - Poissonov broj

The main goal of this 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 under certain circumstances. 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, although the weight distribution ratio is not the same as in cars.

The purpose 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.

2. Numerical approach - calculation of the disc brake model

In the beginning of the analysis the brake disc is simplified. Only one section of the whole disc is used. The disc brake is symmetrical, therefore the model includes 1/12th of the whole disc brake. Only one type of disc was considered. This was the disc with permitted wearing of 7 mm on both sides. The load, used for this analysis, consists of:

- input of the heat flux when braking from the top speed to a standstill,
- a period of 60 s for cooling off,
- acceleration back to top speed,
- braking to a standstill.

All the boundary conditions are calculated for braking on a flat 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 µm. Disc brakes were machined and prepared on CNC machine tool with cutting conditions which were previously optimized by an intelligent optimization software [12]. Surface roughness and cutting forces acting on the disc during machining were kept constant by continuous adaptation of cutting parameters [13] 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.

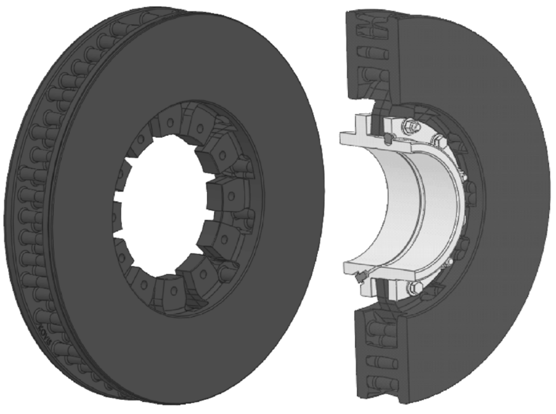

Figure 1. Brake disc

Slika 1. Kočni disk

2.1. Preparation and modelling - 3D model of the disc brake

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

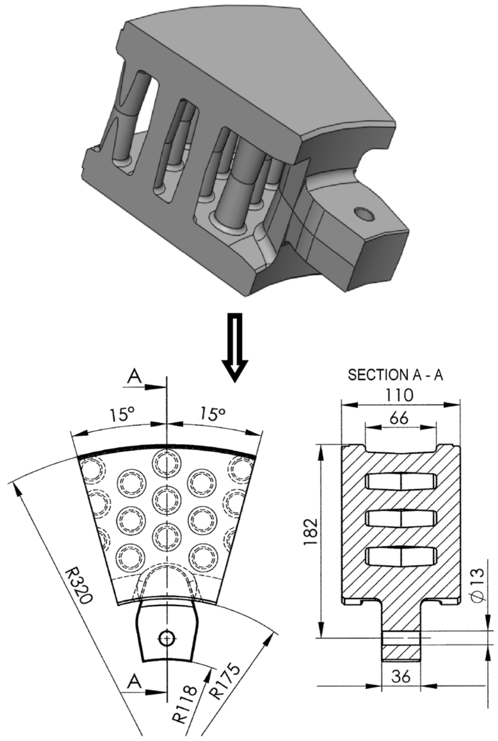

Figure 2. Part of worn disc
Slika 2. Shematski prikaz kočnog diska

2.2. Load determination

The disc was analyzed for one load case, consisting of the following steps:

- braking from the top speed of 200 km/h to a standstill,
- standing still for 60 s (simulating a train stop at a train station),
- accelerating back to top speed,
- once again braking to a stand still.

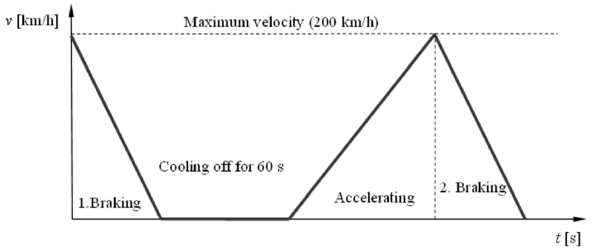

Figure 3. Braking profile of the whole cycle
Slika 3. Profil kočenja

At the beginning of the test cycle, the brake disc has the same temperature as the surrounding environment, which is 20 °C.

Because the train never stops on an inclined track, it has to stop on a horizontal track. Braking to s standstill on a horizontal track is the most challenging load case. Because of the speed reduction, the influence of the wind is smaller and therefore a smaller heat transfer coefficient (10 W/ (m^2 K)) was considered.

In the analysis the disc was checked for temperature and stress arrangement. During the whole braking cycle, the prescribed temperature of 350 °C was not exceeded. This temperature was not reached even with subsequent braking to a standstill.

The air humidity and the influence of the heat radiation were not considered in the calculation.

2.3. Determination of the physical model

All the data, used in the calculation and needed to run the analysis, are shown in Table 1.

Table 1. Material properties
Tabela 1. Materialne lastnosti

Mass of the vehicle / Masa vozila M, kg	70 000
Maximal load per axle / Maksimum opterećenja po osovini, kg	17500
Number of axles per vehicle / Broj osovina na vozilu	4
Number of discs per axle / Broj diska na osovini	2
Start speed / Početna brzina v_0, m/s	56
Deceleration / Usporavanje a, m/s^2	1,4
Braking time / Vrijeme kočenja t_s, s	34
Effective radius of the disc brake / Efektivni polumjer diska za kočenje r_{disc}, m	0,247
Radius of the wheel / Polumjer kotača r_{wheel}, m	0,460
Friction coefficient disc/pad / Koeficijent trenja disk/kočna pločica μ	0,4

Heat flux

A physical model, considering the heat flux in dependence of the braking time, was used to determine the influence of braking. Data and 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, the whole weight is distributed in a ratio of 50/50 [5]. Because of that assumption, all discs are treated equal.

In consideration of weight distribution and the fact that the bogie consists of two axles, one brake disc from the bogie carries 12,5 % of the whole braking force (Figure 4).

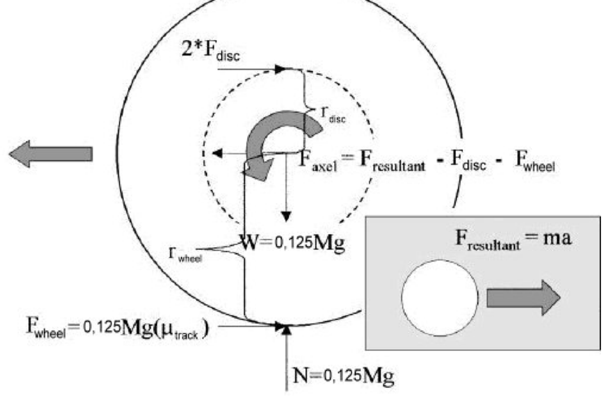

Figure 4. Representation of forces acting on wheel and disc brake, where M represents the mass of rail car and m represents mass of disc and wheel value 0,125 stands for 12,5 % of weight distribution

Slika 4. Reprezentacija svih sila koje utječu na disk za kočenje. M prestavlja masa vozila, m predstavlja masa diska i 0,125 predstavlja 12,5 % distribucije težine

The kinetic energy for one wheel (disc brake) is equivalent to the energy balance [5]:

$$0{,}125 \cdot \frac{1}{2} \cdot M \cdot v_0^2 = 2 \cdot F_{\text{disc}} \int_0^{t_s} v_{\text{disc}}(t)\text{d}t. \tag{1}$$

The energy change in the disc at the moment is equal to the heat flux on the surface of the disc. The Equation 1 is valid in the case of a constant braking deceleration. The braking force on the disc is equal to Equation 2 [5]:

$$F_{\text{disc}} = \frac{0{,}125 \cdot \frac{1}{2} \cdot M \cdot v_0^2}{2 \cdot \frac{r_{\text{disc}}}{r_{\text{wheel}}} \cdot \left(v_0 \cdot t_s - \frac{1}{2} \cdot a \cdot t_s^2\right)} \text{ ,N}. \tag{2}$$

The heat flux at the moment, which affects one half of the disc, is calculated according to the Equation 3:

$$\dot{Q}(t) = F_{\text{disc}} \cdot v_{\text{disc}}(t) = F_{\text{disc}} \cdot \frac{r_{\text{disc}}}{r_{\text{wheel}}} \cdot (v_0 - a \cdot t), \text{W}. \tag{3}$$

The whole cycle lasted for 325 steps. Each step is 1 s long.

Determining the pressure of the brake clamps

The surface pressure between the disc brake and the brake pads was determined on the basis of the calculated braking force and because the brakes work on the brake disc by means of pneumatic system, the pressure was:

$$p = \frac{F_{\text{disc}}}{A_{\text{c}} \cdot \mu} \text{ , MPa}. \tag{4}$$

In our cycle, the heat flux power for one disc was considered. The surface pressure was 1.14 MPa. This boundary condition was also considered in the calculation although because of its low value, it could be disregarded.

Angular velocity

Stresses are also caused by high angular velocity. This velocity was part of the boundary conditions and was calculated according to Equation 5:

$$\omega = \frac{v}{r} = 151, \ \text{s}^{-1}. \tag{5}$$

2.4. Determination of loads, fixing, mesh and material properties

Loads: the symmetrical boundary condition, which is located on the edge of the selected section, was modelled with slide supports in radial direction. The heat flux did not flow thru those supports. But id did flow thru both sides of the disc.

Fixing: fixing was made in the points, where the disc was mounted (screwed) onto the hub (Figure 5).

Mesh: the creation of a mesh volume was conducted automatically by the software package Abaqus CAE 6.7.1. The mesh consists of 86713 tetrahedral elements (element code C3D4AT – allows linear thermo – deformational analysis). The average size of the elements is 6mm and the number of nodes is 18135.

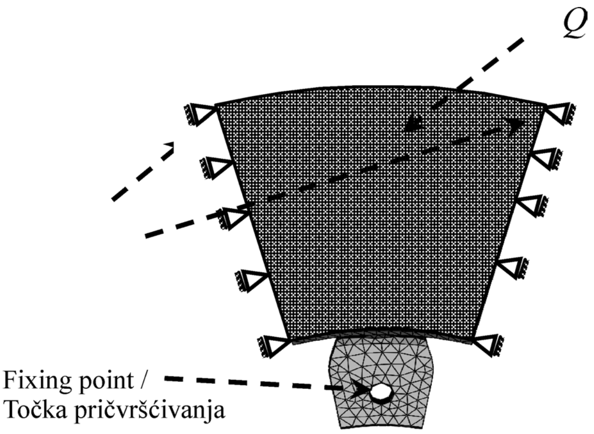

Figure 5. Load, fixing and mesh of the selected section

Slika 5. Opterećenja, oslanjanje i mreža konačnih elemenata odabranog modela

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

Table 2. Material properties

Tabela 2. Materijalna svojstva

Heat conductivity / Toplinska vodljivost λ, W/mK	35,2
Density / Gustoća ρ, kg/m^3	7100
Specific heat / Specifična toplina c_p, J/kgK	515
Module of elasticity / Modul elastičnosti E, MPa	169000
Poisson number / Poissonov broj ν	0,275

3. The results

Analysis of Thermal and Stress Load

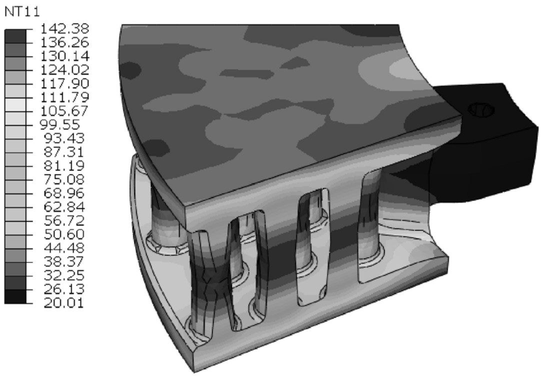

Figure 6. Maximum temperatures after the first braking

Slika 6. Maksimalne temperature nakon prvog kočenja

The figure (Figure 6) represents the maximum temperature after the first braking. The temperature amounted to 143 °C.

After a time period of 40 s (the train has come to a standstill), the temperature fell to 129 °C (Figure 7). At that point, the stress is equal to 109 MPa (Figure8).

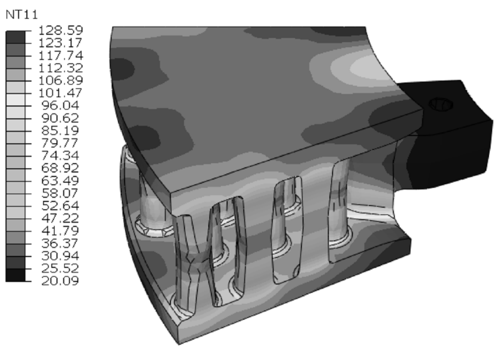

Figure 7. Temperature field of the disc after a time period of 40 s

Slika 7. Temperature diska nakon 40 sekunda

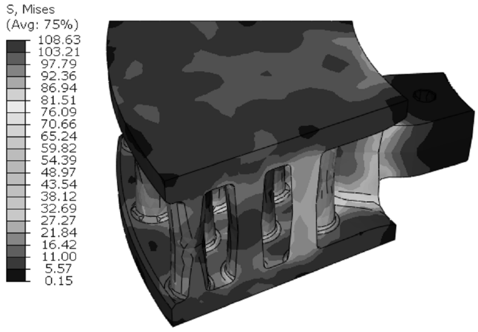

Figure 8. Stress field after 40 s

Slika 8. Naprezanja u disku nakon 40 s

After a time period of 60 s and cooling of, the temperature fell to 93 °C (Figure 9). The accumulated time was 100 s from the start of the cycle. Also the value for stress fell to 65 MPa (Figure 10)

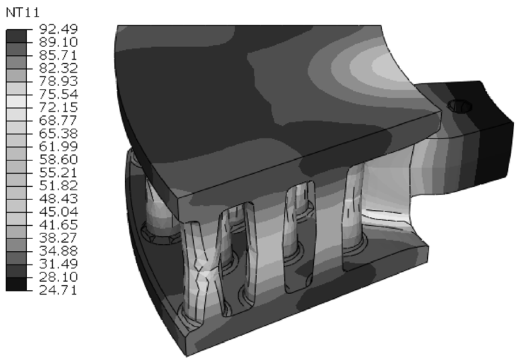

Figure 9. Temperature field of the disc after 100 s

Slika 9. Temperature diska nakon 100 s

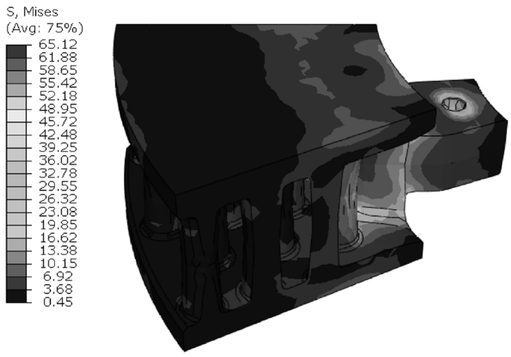

Figure 10. Stress field after 100 s

Slika 10. Naprezanje u disku nakon 100 s

After the time period of 100 s, the program simulated acceleration back to top speed (200 km/h). This acceleration took another 184 s.

After this last time period, before the braking starts, the temperatures fell to 59 °C (Figure 11) and the stress amounted to 210 MPa (Figure 12). The value for stress is high because of the centrifugal influence on the disc.

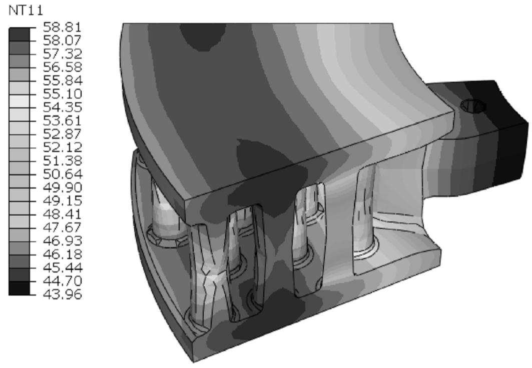

Figure 11. Temperature field of the disc after a time period of 284 s – just before the second braking

Slika 11. Temperature diska nakon 284 s, to je točno prije drugog kočenja

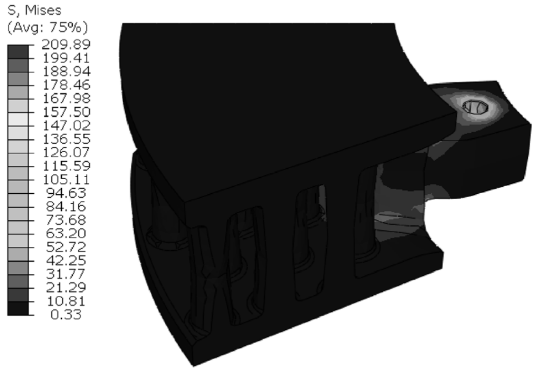

Figure 12. Stress field after 284 s

Slika 12. Naprezanja u disku nakon 284s

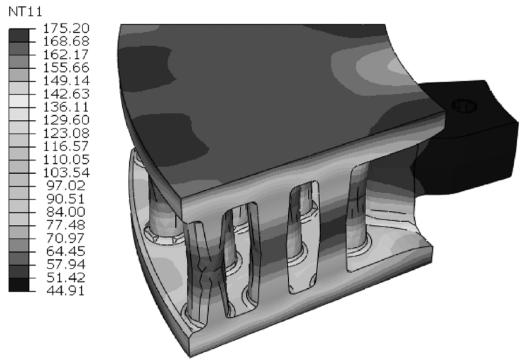

Figure 13. Maximum temperature after the second braking

Slika 13. Maksimalne temperature nakon drugog kočenja

After a time period of 284 s, the second braking of this cycle starts. The temperatures amount from 59 °C to a maximum of 175 °C (Figure 13).

At the end of the cycle, when the vehicle stopped for the second time, the temperature amounted to 161 °C (Figure 14) and the stress values amounted to 200 MPa (Figure 15).

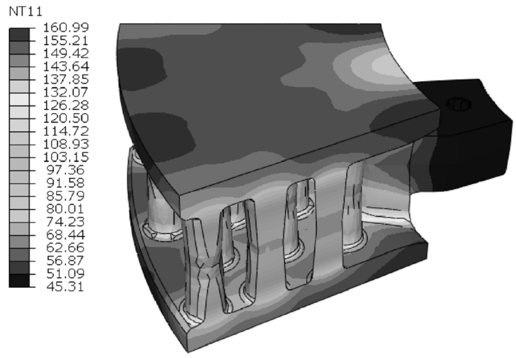

Figure 14. Temperature fields at the end of the cycle

Slika 14. Temperature diska na kraju analize

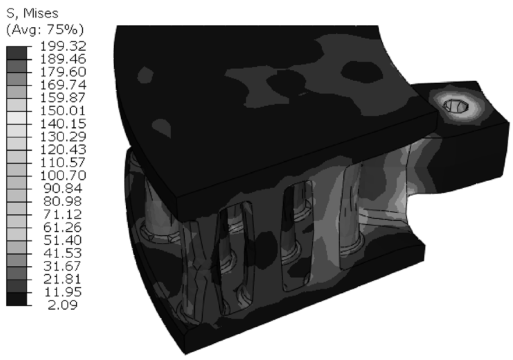

Figure 15. Stress field at the end of the cycle

Slika 15. Naprezanja u disku na kraju analize

The maximum values for stress amounted for the first braking after a time period of 7 s. The highest value of 145 MPa appeared between the disc rim and the fixing point (Figure 16). For the second braking, the value amounted to 211 MPa in a time period of 285 s after the whole cycle began (Figure 17). Here, the highest stress concentration transferred more to the fixing point of the disc.

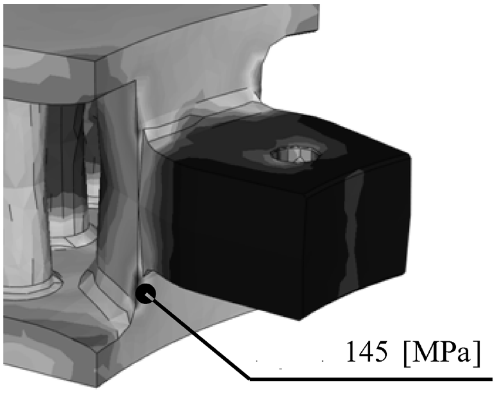

Figure 16. Maximum values for stress during the first braking

Slika 16. Najveća naprezanja tijekom prvog kočenja

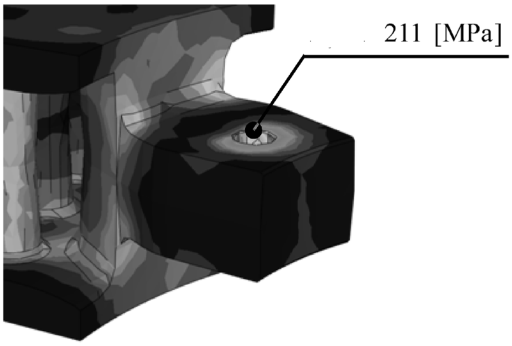

Figure 17. Maximum values for stress during the second braking

Slika 17. Najveća naprezanja tijekom drugog kočenja

This analysis of the disc was implemented for one example; braking on a horizontal track to a standstill, cooling off faze, accelerating back to top speed and again braking to a full stop. Following temperatures were taken into account; the initial temperature of the disc, disc fixture and the ambient temperature, all equal to 20 °C.

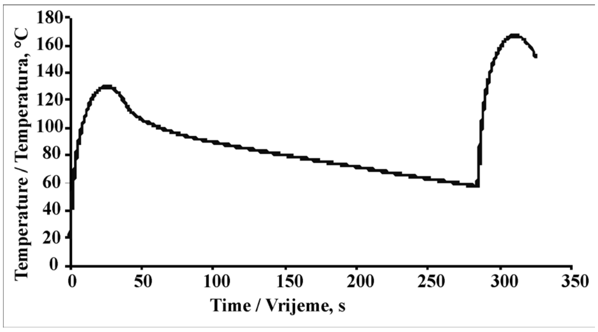

Figure 18. Curve, representing the changing temperature in a single node

Slika 18. Tijek temperature u odabranom čvoru mreže

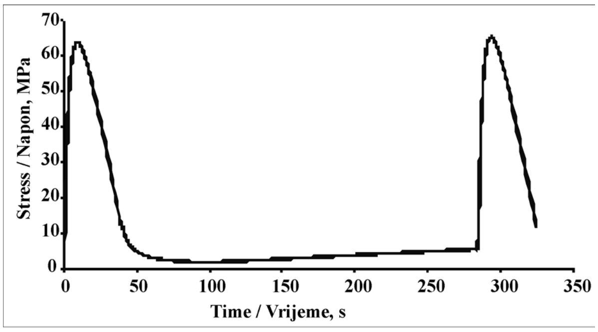

Figure 19. Curve, representing the changing values for stress in a single node (node number 5453)

Slika 19. Tijek naprezanja u odabranom čvoru mreže (čvor broj 5453)

4. Discusion of results

The braking cycle could be larger, with more time steps and with more braking and cooling off fazes. Because of the limited CPU capabilities, the analysis focused on just one cycle. The results are shown in the following tables.

Table 3. shows the temperatures at a different time step of the braking cycle.

Tabela 3. Temperature diska u različitim vremenskim razmacima

Stage/time step / Faza/vremenski korak s	Temperature / Temperatura, °C
40	129
100	93
284	59
325	161

At time step 40, the train stops for the first time. The temperatures rose from 20 °C to 129 °C. From time step 40 to time step 100, the train was stationary and did not move. The temperature fell to 93 °C. After the time step 100, the vehicle began to accelerate to maximum velocity of 200 km/h. This acceleration took a time period of 184 s. During that time, the disc cooled off to a temperature of 59 °C. When the top speed was achieved, the vehicle began to brake. In the next 40 s the temperature rose from 59 °C to 161 °C.

The maximum temperatures, achieved during the first and the second braking, occurred at the time step 28 and the time step 312. The values for both temperatures are in table 4.

Table 4. Maximum temperatures

Tabela 4. Maksimalne temperature

Stage/time step / Faza/vremenski korak s	Temperature / Temperatura, °C
28 – 1 braking / kočenje	143
312 – 2 braking / kočenje	175

In the whole time of the braking cycle, the temperatures did not reached 350 °C which is the upper allowable temperature of the disc.

The stresses are high, mostly because of the high travelling speed of the vehicle. Table 5 shows values for stresses in different time steps.

Table 5. Stress per time steps

Tabela 5. Naprezanje u disku u različitim vremenskim razmacima

Stage/time step / Faza/vremenski korak s	σ_{max}, MPa
40	109
100	65
284	210
325	200

The maximum stresses, achieved during the first and the second braking, occurred at the time step 7 and the time step 285 and are a direct consequence of the high angular velocity - centrifugal load. The results are in table 6.

Table 6. Maximum strees

Tabela 6. Maksimalna naprezanja

Stage/time step / Faza/ vremenski korak s	σ_{max}, MPa
7 – 1 braking / kočenje	145,2
285 – 2 braking / kočenje	210,5

The stresses are high, but in comparison with permissible stresses amounting to 210 MPa and by considering a safety factor of 1,5; the analyzed disc is properly dimensioned.

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

- selection of another, better material with improved mechanical properties,
- modification of transition and radii between the disc rim and the fixing point, where stress concentrations occur.

To verify the results of numerical analyses also an experiment has to be run thru. Only then, if the results are the same, we can proceed with further numerical analyses, which is at the same time a recommendation for further work.

5. Conclusion

The disc heats up to high temperatures. The highest temperature is 175 °C. Considering the material, the design of the brake disc and the outcome of the results, we can conclude, that the model of the disc is adequate. Also the set target of the analysis meets the makers standard. The analysis should also be run with different boundary conditions to see, how a higher starting temperature of the brake disc affects the end results. In addition, other boundary conditions should be taken into account as shear stresses, residual stresses and the effect of service life with cyclic load, which the disc material must daily support without failing or breaking down. In this analysis, those boundary conditions were not included.

A new trend in railway vehicles shows how the old grey cast iron brakes are being replaced with new, quieter and more reliable composite material brake discs. Disc brakes from composite materials have several advantages to cast iron ones. They have a reduced sound barrier and a higher resistance to temperatures. For further work we would suggest a comparison test with the same boundary conditions, to see, if the values for stress are lower as well.

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