

NUMERICAL OPTIMIZATION OF BRAKE DISCS FOR RAILWAY VEHICLES

NUMERIČNA OPTIMIZACIJA ZAVORNIH DISKOV ZA TIRNA VOZILA

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Keywords: Divided brake disc, computational fluid dynamics, brake disc temperature, fluid-structure interaction, finite element analysis

Abstract

In this article, a systematic numerical optimization of brake discs for railway vehicles with computer-aided engineering tools is presented. The main design parameters of brake discs were defined. One-way fluid-structure interaction analysis was performed. Computational fluid dynamics software tools were used to calculate heat transfer coefficients, which were transferred into thermal transient finite element method analysis to predict brake disc temperatures during different braking scenarios. Systematic analysis of defined design parameters was made. With the help of computer-aided engineering and numerical parametric design, it was possible to design and analyse several innovative designs for brake discs, which enables energy and material savings.

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Povzetek

V članku je prikazana sistematična numerična optimizacija zavornega diska za tirna vozila s pomočjo orodij za računalniško podprt inženiring. Definirani so glavni parametri za načrtovanje zavornega diska. Izvedena je bila analiza enosmerne interakcije stena-tekočina. Orodja za numerično dinamiko tekočin so bila uporabljena za določitev toplotne prestopnosti, ki je bila prenesena v termično simulacijo po metodi končnih elementov za določitev temperature zavornega diska v različnih scenarijih zaviranja. Narejena je bila sistematična analiza vpliva definiranih parametrov načrtovanja zavornega diska. S pomočjo računalniško podprtega inženiringa in parametrične numerične analize smo naredili več inovativnih oblik zavornih diskov, ki istočasno omogočajo prihranke energije in materiala.

1 INTRODUCTION

Brake discs are a critical component found in almost all moving vehicles. Safety and reliability are crucial for braking systems. Several types of brake discs are used in rail vehicles, which are mainly divided into two groups: axle-mounted brake discs and wheel-mounted brake discs. Brake discs are usually made from grey cast iron, nodular cast iron or steel, depending on the loads to which they will be subjected.

The use of brake discs prevents the surface of the rolling wheel itself from braking, which is significantly better for the wheels. This increases the lifespan and reduces stress on the wheels, which are also among the most crucial components in rail vehicles. Divided brake discs were developed to facilitate replacement of the brake discs, making the process less time consuming and less expensive.

Brake discs used for rail freight vehicles are subjected to large fluctuations in temperature and must accumulate and dissipate large amounts of energy due to high thermal and mechanical loads. During braking, lower maximum temperatures and temperature gradients are preferred. These are the main cause of brake disc deformations and stress. The stress caused by temperature gradients is far greater than the stress caused by the braking moment as observed from structural, finite element method (FEM), simulations. At the same time, it is also preferable that ventilation losses due to drag be reduced. Ventilation losses are caused by the rotation of the brake disc in the atmosphere. Air circulation is necessary for dissipating heat into the environment, but high ventilation losses can cause high energy use only for air circulation. Unfortunately, most of these parameters are contradictory. Low ventilation losses while obtaining high thermal dissipation or low temperatures while having low mass are very hard to achieve. This is where design parameters described in this paper come into consideration. With an improved design, material and power savings are possible.

Computer-Aided Engineering (CAE), which covers the extensive use of computer software to assist in engineering tasks and analyses, is increasingly evolving in engineering practice. It includes software for the analysis of solids using the finite element method and software for analyses of computational fluid dynamics (CFD), [1, 2, 3]. Current software packages enable studies of coupled physics, known as one or two-way fluid-structure interactions (FSI). The CAE software can also include add-ons to automatically optimize the analysed products, [4, 5, 6, 7, 8]. CAE analyses are critical in the development of products for achieving an optimized design of the product in the virtual environment.

As described in [1], CFD and FEM analyses are appropriate for predicting real properties of the brake disc. CFD and FEM simulation setup and loading conditions are described in [1, 2, 3, 4, 9, 10]. All these papers describe numerical analysis and comparison with real tests. There is also a numerical comparison of full and ventilated brake discs in [2].

2 METHODOLOGY

The main subject of the presented research is an axle-mounted brake disc. There are two of these brake discs mounted per axle of a freight rail vehicle with an axle load of 25 t per axle. The goal of systematic optimization with CAE tools is an improvement of the brake discs efficiency while lowering the mass of the disc. The mass of the brake disc is critical, as a lighter brake disc has less rotational and translational inertia. Lower mass also means smaller production costs because less energy and material are needed for the production of the brake disc.

Figure 1 presents a brake disc cutaway for easier presentation of parameters described in the paper. The initial design of the brake disc before the CAE optimization process is presented in the figure. The brake disc has a width of 170 mm, an outer diameter of 590 mm and the inner diameter of 325 mm and 10 mm of wear material. These dimensions were fixed throughout the optimization. The initial weight of the brake disc assembly was 145 kg. The top speed of the analysed vehicle is 160 km/h.

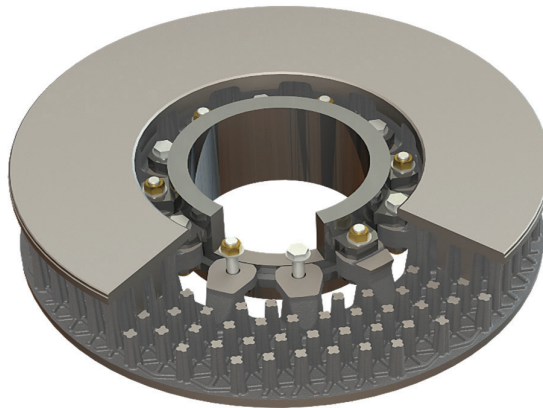


Figure 1: Rail freight vehicle brake disc (initial design)

2.1 Governing equations

The CFD model is solved using the finite volume method with the Reynolds-averaged Navier-Stokes equations (RANS) modelling approach. The computational domain is divided into small volumes where the conservation equations for each volume are integrated. A system of discrete algebraic equations is iteratively solved. The mathematical model used to describe a given physical problem is a set of integral-differential equations and constitutive relations and initial and boundary conditions. Conservation of mass in the general form of the mass conservation equation could be written as [11]:

$$\frac{\partial \rho}{\partial t} + \nabla \cdot (\rho \vec{v}) = S_m \quad (2.1)$$

Conservation of momentum is defined with [11]:

$$\frac{\partial}{\partial t}(\rho \vec{v}) + \nabla \cdot (\rho \vec{v} \vec{v}) = -\nabla p + \nabla \cdot (\vec{\tau}) + \rho \vec{g} + \vec{F} \quad (2.2)$$

Conservation of energy can be defined with [11]:

$$\frac{\partial}{\partial t}(\rho E) + \nabla \cdot (\vec{v}(\rho E + p)) = \nabla \cdot \left(k_{eff} \nabla T - \sum_j H_j J_j + (\vec{\tau}_{eff} \cdot \vec{v}) \right) + S_h \quad (2.3)$$

For transient FEM analysis, the linear system could be defined with [11]:

$$[C]\{\dot{T}\} + [K]\{T\} = \{Q(t)\} \quad (2.4)$$

3 DESIGN PARAMETERS AND NUMERICAL MODEL

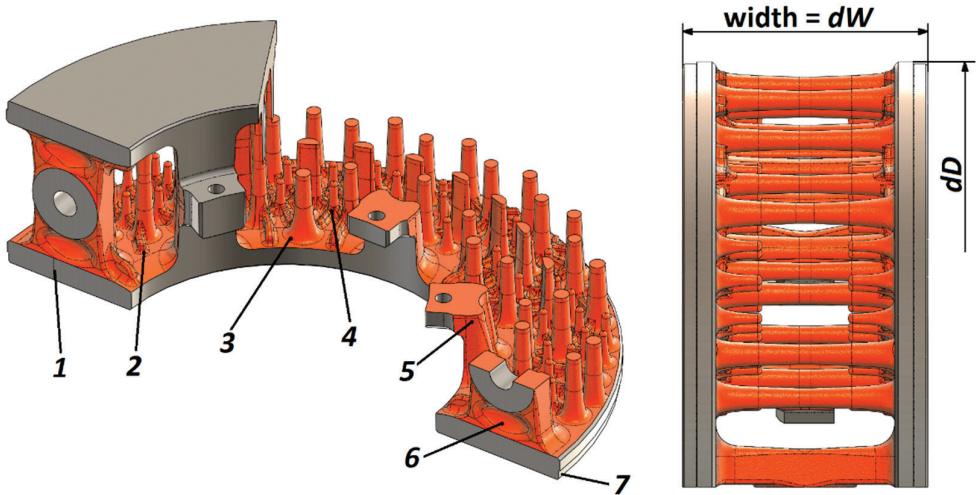


Figure 2: Presentation of design parameters

Figure 2 presents the main design parameters of the brake disc, which were varied and studied in the presented research. Some of these parameters, such as dW and dD , are usually fixed by the end-user. Other parameters (1 to 7) could be changed by the designer of the brake disc to achieve the required performance of the brake disc. Parameters 1 to 7, represented in Figure 2 are:

- 1 Friction plate thickness: it can affect temperatures as heat input is located on it. With increased thickness, lower temperatures are observed as a result of increased thermal capacity. Although high thermal capacity is good, higher plate thickness causes higher mass of the braking plate.
- 2 Additional net system cooling ribs: it was observed that thermal heat conduction is increased with increasing height and width of these ribs.
- 3 Radius in the root of the main cooling ribs: it can increase the heat flux from the friction plate towards the centre of the brake disc. The number of ribs itself is a design parameter, but the number must be sufficient to carry the structural loads.
- 4 Small cooling ribs: they were developed using parametric design optimization. Dimension parameters of cooling ribs are tuned by the use of transient thermal FEM simulation. These small ribs increase the thermal capacity, and they were patented during the design process, because of the large improvement in efficiency. They combine low mass while lowering maximum temperature on the friction surface. These can also be positioned with precision on the spots where they are needed based on transient thermal simulations.
- 5 The number of mounting brackets: it was observed that maximum stress and deflections are caused by temperature gradients and not by the actual braking torque itself. Phenomena such as thermal banding, hot-spotting and thermal cracking are thought to be the main reasons for disc failure, [6]. In fact, braking torque of around 8000 Nm per brake disc loads the brake disc only slightly. That is why the number of mounting brackets could be reduced from 12 to 6 as there were some great weight savings made by this reduction. There is another benefit of reducing the number of mounting brackets. That is the increased flow of air through the brake disc, but it caused larger ventilation losses. Some of the cooling ribs are designed for lower air resistance to obtain preferred properties.
- 6 Some additional weight savings were made by reducing the size of the bracket which connects two halves of the brake disc.
- 7 Wear limit is presented. It was fixed at 10 mm.

The ANSYS CFX commercial software package was used for CFD simulations. CFD analyses were used mainly to determine wall heat transfer coefficient values and ventilation losses. Previous research proved that prediction of ventilation losses corresponds with real measurements [1, 9]. The material of the brake disc is EN-GJL-250 grey cast iron which has a relatively high thermal conductivity of 60 W/(mK), but it has a relatively low density of 7.2 kg/dm³. There are also other material choices, such as nodular cast iron or CrMo steel, [7]. The choice of material is based on its mechanical properties, cost, and thermal diffusivity.

Numerical mesh, used for CFD simulations, represents the surrounding air and a rotating domain. Domain size was proven to be large enough not to have any effect on the results. The mesh has fine inflation layers on the walls to capture high-pressure gradients and heat transfer. The domain was decomposed into six subdomains for better control over the mesh, [1]. The innermost domain was meshed with tetrahedral and other domains with hexahedral elements. By continuous refining of the mesh, we found that the solution, independent of the computational mesh, could be calculated with the mesh with 13.8 million elements.

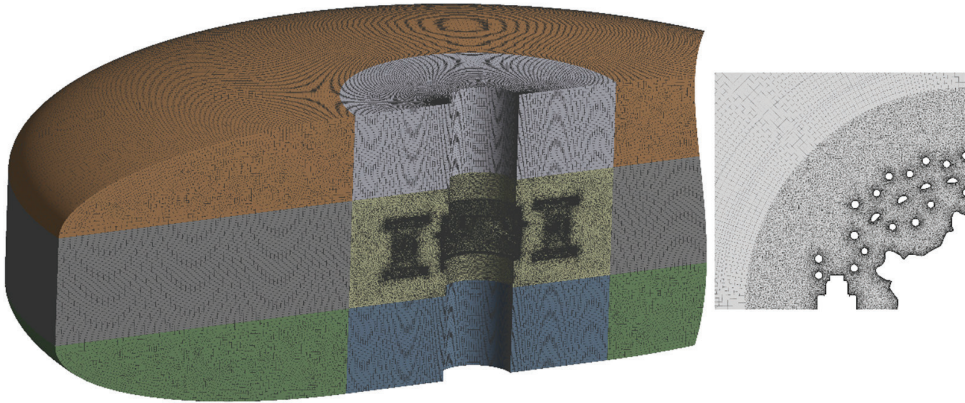


Figure 3: Numerical mesh for CFD simulation

Results from CFD simulations were then used as boundary conditions in transient thermal FEM simulations (one-way Fluid-Structure Interaction simulations). Figure 4 presents the heat transfer coefficient (HTC) contour transferred from CFD software for one of the analysed cases. The heat input of 550 kW peak at a braking time of 2 s was set on friction surface where brake pad touches the brake disc for an emergency brake scenario: 89% of the total energy was set to be absorbed by the brake disc, and 11% by the brake pad, [1]. The total braking time for the emergency brake scenario was 40 s. The drag brake scenario is a longer type of braking, which is based on constant heat inputs of 20, 30, and 40 kW. This scenario simulates a train braking downhill, maintaining a constant velocity of 70km/h. This braking scenario takes 34 minutes from start to finish in which the velocity of the disc remains constant. Both braking scenarios were simulated on new and worn-out brake discs. The worn-out brake disc model had 10 mm thinner braking plates, which resembled a brake disc that is at the end of its lifecycle and should be changed. Meshes were tested for all cases, and the results are independent of the computational mesh.

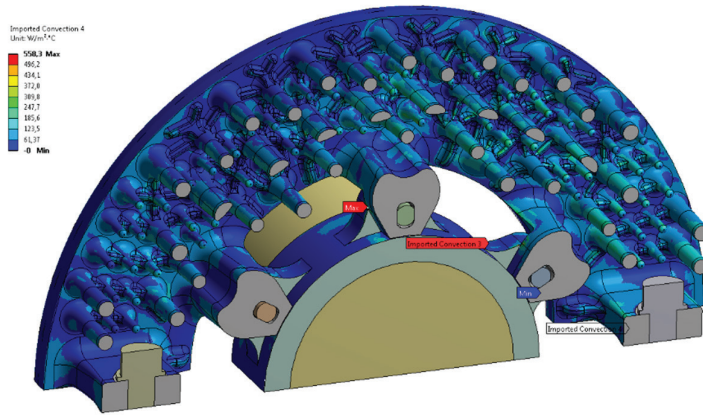


Figure 4: Model used for transient thermal analyses

4 PRELIMINARY STUDY OF PARAMETRIC DESIGN STRATEGIES

4.1 Net pattern cooling ribs

Figure 5 represents parametrized net cooling ribs. Net cooling ribs are small ribs that connect larger load-bearing ribs. These are arranged as a net pattern, which explains their name. The temperature chart represents maximum emergency brake temperatures on the friction surface. On the figure, the left-most cooling rib has no net rib. On the right-most cooling rib, the net rib is optimized based on the mass-to-heat flux ratio. The cooling rib had a fixed width, but the height and radius were variables. This simulation was done on a very small subsection, so numerous different designs were analysed.

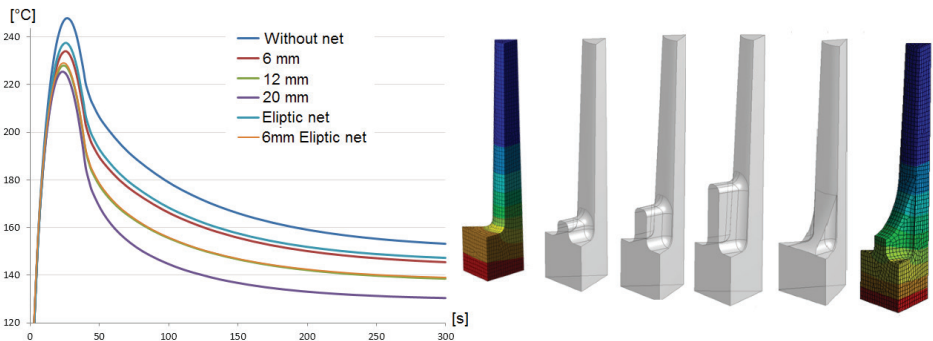


Figure 5: Net cooling rib design

It was observed that small intermediate cooling ribs like those presented in Figure 5 could increase heat flow from the friction plate towards the centre of the brake disc. This helps to lower the maximum temperatures given the same power input and material properties.

4.2 Cooling rib height parameter

Figure 6 presents the average and maximum temperatures of the cooling rib during emergency stop braking. It was observed that cooling fin length does not affect temperatures during short, powerful braking scenarios. The difference is negligible because heat does not have enough time to be conducted towards the centre of the brake disc. It is seen that a longer cooling rib has a lower average temperature and stays cooler after the braking is already over.

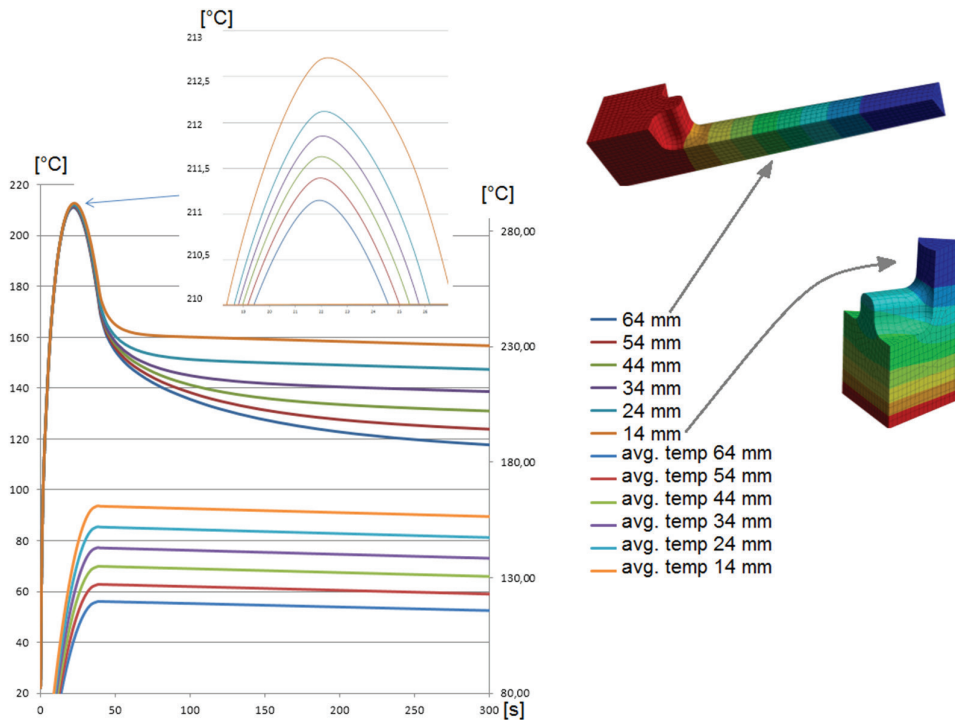


Figure 6: Height of the cooling fin as a parameter, this parameter corresponds to the width of the disc

4.3 Root filet of the cooling ribs

The radius in the root of the cooling ribs was set as a parameter. As observed in Figure 7, an elliptic radius was placed in the root of the large cooling ribs. Figure 7 represents four different designs; the left-most model has no radius, and the right-most model has a large radius in the root. All models were simulated in new and worn condition and both for emergency and for drag brake scenarios. The worn condition represents a braking plate thickness that is 10 mm worn away. It was observed that a large root radius of the cooling rib lowers the maximum temperatures of the friction surface during emergency braking. Lower temperatures are desirable, but the use of large fillets makes the weight go up so a compromise must be made.

Lower temperatures are a consequence of area A for the heat conduction increasing, which increases heat flow towards the centre of the brake disc away from the heat input surface.

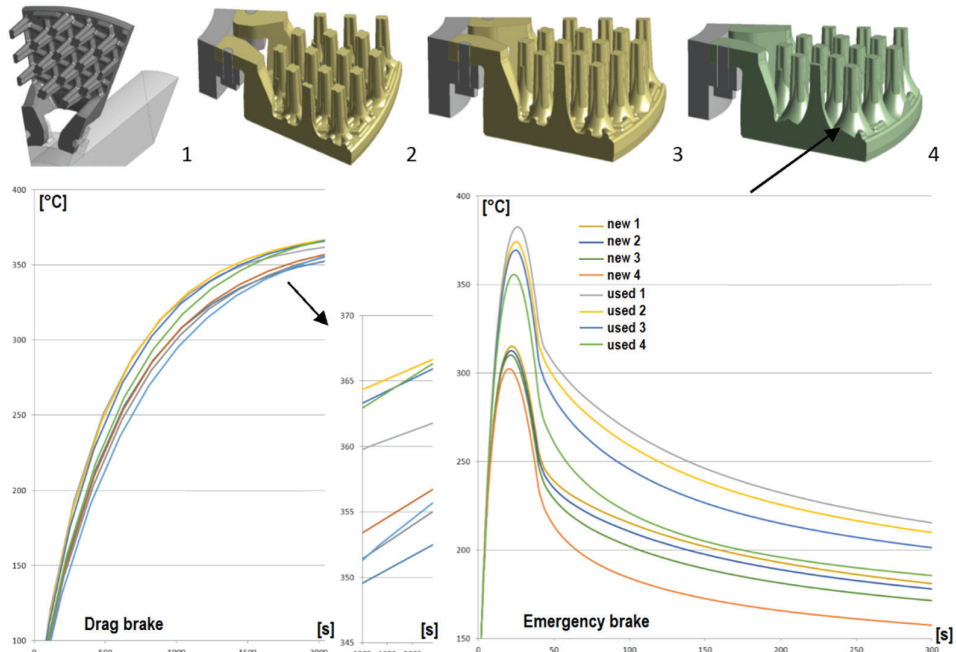


Figure 7: Root fillet parameter

4.4 Intermediate cooling ribs

It was observed from our previous parametric studies that it is preferred to have the distribution of mass more towards the brake plates where heat input is located. From the structural simulation, it was observed that the number of large load-bearing cooling ribs could be reduced and that the mass could be “moved” more towards the brake plates, so this was examined more closely. Figure 8 represents three designs, although much more designs were tested out. Parameters were, in this case, concentrated more on the intermediate cooling ribs. Effects on temperature during emergency and drag brake were examined. We can conclude that huge difference in maximum temperatures can be made during short brakings with the use of intermediate cooling ribs while reducing mass greatly. These small cooling ribs also have great thermal dissipation properties. Intermediate cooling ribs were isolated, and separate simulations were done to optimize the shape of these ribs, and a patent was made. The optimization was done with the multi-objective genetic algorithm (MOGA) which is a variant of a popular non-dominated sorted genetic algorithm-II (NSGA-II) based on controlled elitism concepts. Figure 9 presents this small rib and its effect on the temperature field. It simulates the larger one in short brakings but heats up more during long braking due to lower thermal capacity.

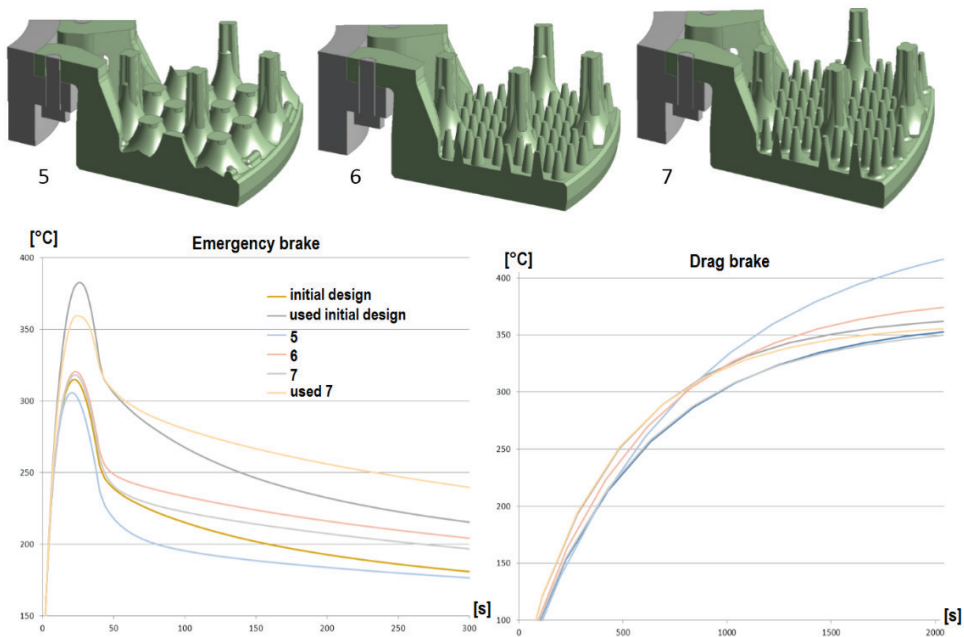


Figure 8: Number and size of intermediate cooling ribs

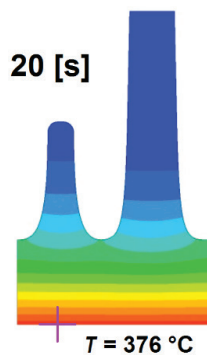


Figure 9: Optimized intermediate cooling rib during an emergency stop scenario at $t = 20$ s

4.5 Cooling rib shape parametric optimization

With the use of computational fluid dynamics software, a simulation was set up so that the cross section of the cooling rib itself was determined by seven parameters. Figure 10 presents how the shape parameters were set. With this type of settings, all the designs from Figure 11 were created automatically. Parameters were varied from a set minimum, and maximum values and a central composite design was used for the design of experiments. About 520 random shapes of cooling rib vents were simulated, with an automatic, intermediate update of geometry.

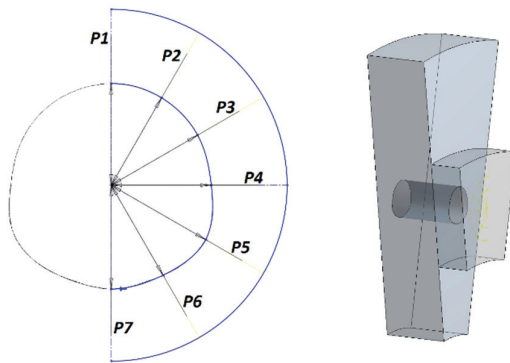


Figure 10: Cooling rib shape parameters (P1-P7) with a spline used for geometry generation

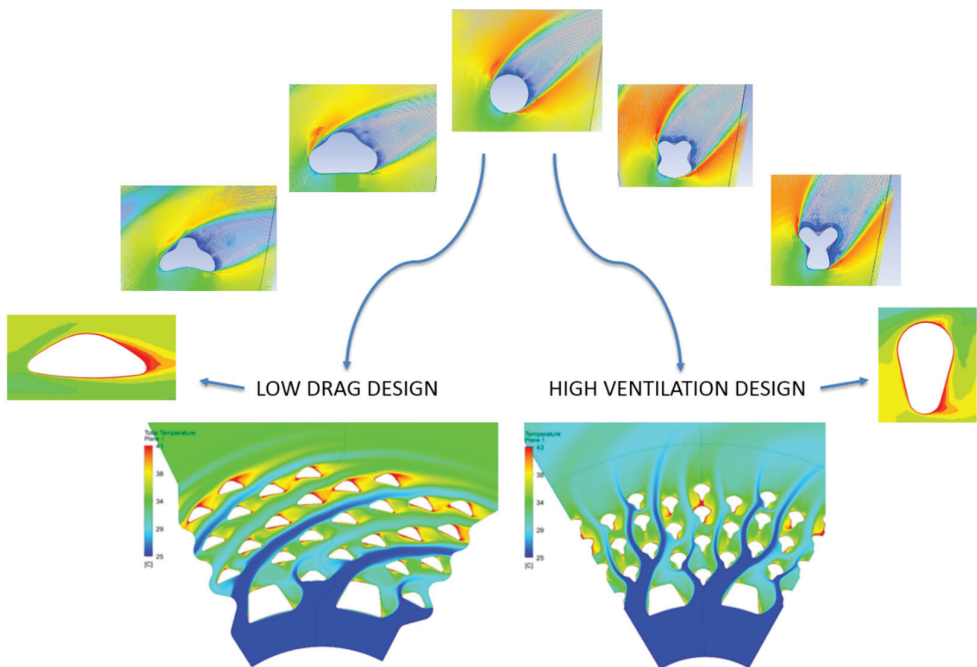


Figure 11: Parametric optimization of cooling rib

With results of the systematic disc parameters analysis, it was possible to create and analyse different brake disc designs. Throughout the process, designs were also evaluated to determine whether they were fit for metal casting, as that is the process of making brake discs.

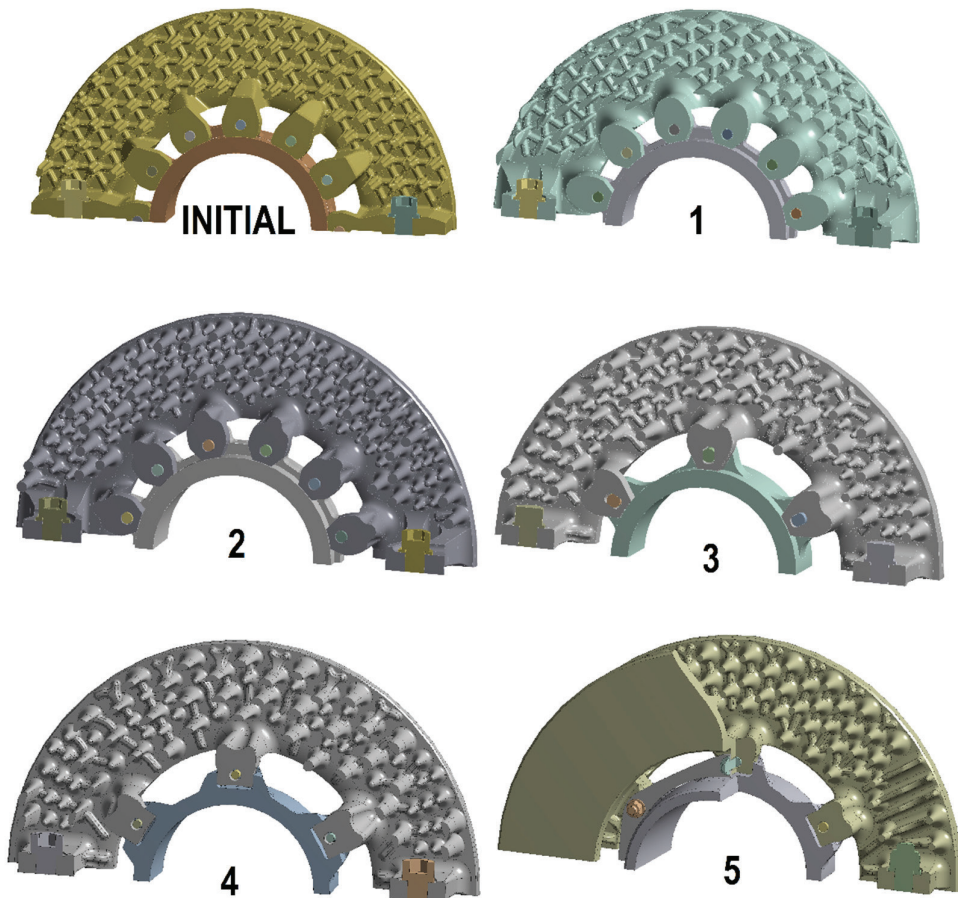


Figure 12: Rail Freight vehicle brake disc development

Figure 12 presents six designs created with the help of parametrical numerical simulations. Brake disc 1 is the initial design of the brake disc. Brake disc 2 is optimized for low ventilation losses. Brake disc 3 is optimized for heat conduction. Brake discs 4 and 6 are optimized for low mass and high thermal dissipation. Brake disc 4 is the best mix of all previous design parameters.

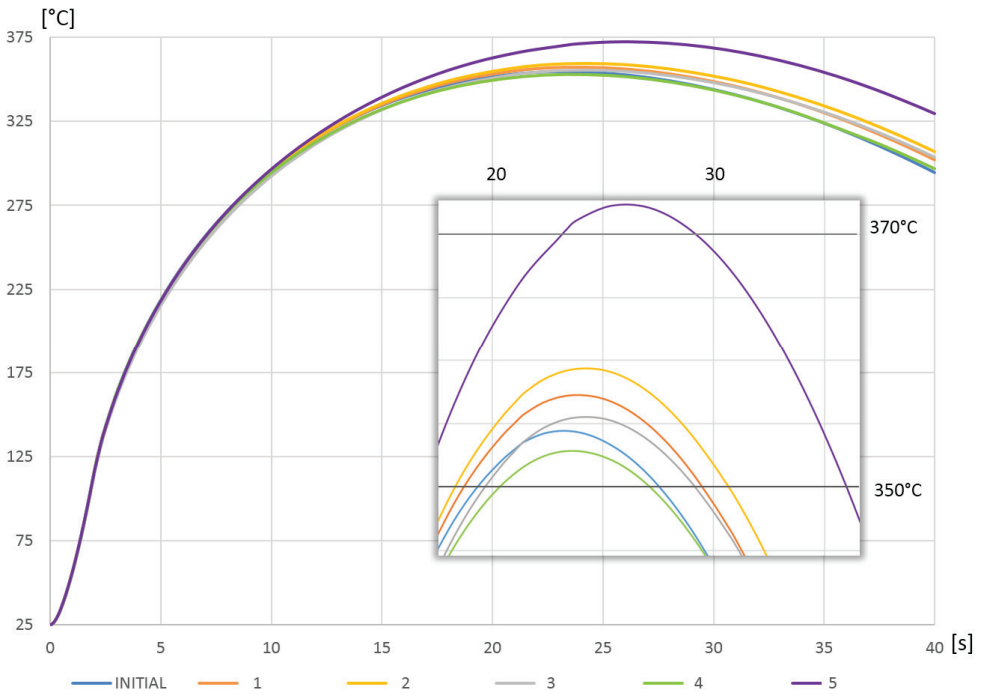


Figure 13: Emergency brake temperatures of 6 brake disc designs

Figure 13 shows emergency brake temperatures for six analysed designs of brake discs. As is observed from the chart, the lowest temperature during emergency braking was achieved with Design 4, despite it being 21.4% lighter than initial design and having 21% less ventilation loss. Design 2 has the lowest air resistance. Low ventilation losses also save energy during operation, as there can be around 180 brake discs in a train composition. In this case, if the composition is travelling at 160 km/h, it is possible to reduce the power needed to run it by 36 kW. With the optimized shape of the cooling rib, it is possible to save energy and increase the efficiency of the moving vehicle.

5 CONCLUSION

The performed parametric design analysis again confirmed that, with the use of computational fluid dynamics simulations and structural simulations, using the finite element method, CAE analyses are a very important tool in the development of products. The performed complex study of coupled physics, one-way fluid-structure interaction simulation, was done by systematic parametric optimization, which enables the study of novel ideas and optimized design of the product in the virtual environment.

Parametric design optimization was used at the beginning of the study to optimize the shape of cooling ribs. With newly defined geometry parameters we could change specific properties of brake disc parameters, which are essential during braking, which was confirmed with two types of braking scenarios simulations. It was confirmed that defined design parameters affect

maximum and average temperatures obtained during braking and that optimal brake disc design depends on the loading type.

Based on the parametric study new designs of the braking disc were studied. All new designs are significantly better than the initial design. It was observed that all novel designs achieve lower maximum and average temperatures obtained during an emergency braking scenario, but they are at the same time significantly lighter than the initial design.

It was also observed that all designs have fewer ventilation losses due to the drag of brake disc in a real environment. Low ventilation losses save energy during operation. With the optimized shape of the cooling rib, it is possible to save energy and increase the efficiency of the moving vehicle. With 180 brake discs in a train composition and speed of composition at 160 km/h, it is possible to reduce the energy use by 0.225 kWh/km. In case of electric composition and if emission for electricity production is 315 g CO₂ per kWh [12] we could expect 71 g lower emissions of CO₂ per every km, which could lead to significant reduction of emissions.

References

- [1] **U. Grivc:** *Numerical simulation of thermal and structural loads in rolling stock brake disc*, University of Maribor, Faculty of Mechanical Engineering, 2016
- [2] **A. Belhocine:** *Computational fluid dynamics modelling and computation of convective heat coefficient transfer for automotive disc brake rotors*, February 2018 Computational Thermal Sciences, vol. 10(1) pp. 1–21, 2018
- [3] **M. Reibenschuh, G. Oder, F. Čuš, I. Potrč:** *Modelling and Analysis of thermal and Stress Loads in Train Disc Brakes – braking from 250 km/h to Standstill*, Strojniški vestnik - Journal of Mechanical Engineering, vol. 55(7-8), pp. 494–502, 2009
- [4] **P. N. Gunjal:** *Design, Analysis & Optimization of Disc Brake*, International Engineering Research Journal, Special Issue 2, pp. 5010–5016, 2015
- [5] **I. Cayiroglu R. Kilic:** *Wing aerodynamic optimisation by using genetic algorithm and Ansys*, ACTA PHYSICA POLONICA, Vol 132, pp. 981–985, 2017
- [6] **A. Durgude, A. Vipradas, S. Kishore, S. Nimse:** *Design optimisation of brake disc geometry*, MAE 598-2016-11, Final Report, 2016
- [7] **P. Baskara Sethupathi, A. Muthuvel, N. Prakash, L. Stanly Wilson:** *Numerical Analysis of a Rotor Disc for Optimization of Disc materials*, Journal of Mechanical Engineering and Automation, vol. 5(3B), pp. 5–14, 2015
- [8] **E. Palmer, R. Mishra, J. Fieldhouse:** *An optimization study of a multiple-row pin-vented brake disc to promote brake cooling using computational fluid dynamics*, Proceedings of the Institution of Mechanical Engineers Part D Journal of Automobile Engineering, 223(7) pp. 865–875, 2009
- [9] **M. Pevec, I. Potrč, G. Bombek, D. Vranesevic:** *Prediction of the cooling factor of a vehicle brake disc and its influence on the results of a thermal numerical simulation*, International Journal of Automotive Technology, Vol. 13(5), pp. 725–733, 2012

- [10] **C. Jiguang, G. Fei:** *Temperature field and thermal stress analyses of high-speed train brake disc under pad variations*, The Open Mechanical Engineering Journal, vol. 9, pp. 371–378, 2015
- [11] ANSYS CFX Solver Theory Guide (2016), Release 17.2, ANSYS Inc, Canonsburg
- [12] **International Energy Agency:** *CO₂ emissions from fuel combustion*, Highlights, 2017

Funding

This work was supported by the European Union’s Horizon 2020 research and innovation programme under grant agreement No. 700985.

Nomenclature

A	Cross-section area
$[C]$	specific heat matrix
dD	disc diameter
dW	disc width
E	energy
$\vec{F}$	gravitational body force and external body forces
$\vec{g}$	standard gravity vector
J_j	diffusion flux of species j
$[K]$	thermal conductivity matrix
k_{eff}	effective conductivity
p	static pressure
$\{Q(t)\}$	time-dependent heat flow rate vector
S_h	volumetric heat sources.
S_m	mass added to the continuous phase
t	time
$\{T\}$	temperature vector
$\{\dot{T}\}$	time derivative vector of temperature
$\vec{v}$	velocity vector
ρ	density of the fluid
ρ_s	density of the solid
$\bar{\tau}_{eff}$	stress tensor