DOI: 10.21062/mft.2024.061 http://www.journalmt.com

Thermal – Static Analysis of the Brake Disc in SolidWorks

Daniela Koštialiková (0000-0003-4981-7698), Mariana Janeková (0000-0002-9380-2760), Petra Dubcová (0000-0003-3450-3290), Marek Hulc
Faculty of Industrial Technology in Puchov, Alexander Dubcek University of Trencín, I. Krasku 491/30, 020 01 Puchov, Slovakia. E-mail: daniela.kostialikova@tnuni.sk, mariana.janekova@tnuni.sk, marek.hulc@student.tnuni.sk

The work deals with the creation of 3D model of a disc brake of a personal vehicle, followed by a simulation of thermal and strengh analysis during car braking in the SolidWorks program. The disc brake is made of a gray cast iron, the most common material for this application. Cast iron discs offer excellent performance, making them suitable for various types of vehicles. The deformation of a brake disc was analyzed by using the program SolidWorks, where the first a simulation of the thermal load was performed, the aim of which was to determinate the temperature distribution on the brake disc and display the temperature rise during braking. In the next part, the simulation of strength deformation was carried out, and thus determination of the ther-mal load effect of the brake pads on the brake disc.

Keywords: Disc brake, Thermal analysis, Static analysis, Drum brakes, SolidWorks

1 Introduction

There's no safety system on a vehicle more important than the braking system. This system helps slow down the rotation of the wheels when the brake pedal is depressed, ensuring a vehicle comes to a complete stop [1- 2]. Major parts of a disc brake system include the master cylinder, caliper, rotor and pads. Friction brakes work on the prin-ciple that the friction pads are pressed against a rotating disc, and thereby produce frictional forces that slow down the rotating disc, and consequently reduces the speed of the car or completely stops it. During the braking process, the brake pads press against the brake disc, and frictional heat is generated, which is absorbed and dissipated by the brakes and its components [3]. For automobiles, the most commonly used brake disc material is grey cast iron (GCI), because it possesses excellent friction properties, has low cost, retains strength at elevated temperature, has relative ease of manufacture and is thermally stable [4]. Unfortunately, the weak corrosion resistance, heavy weight and weak wear resistance are some of the drawbacks of grey cast iron as brake disc material. The SolidWorks program enables the creation of a 3D model of a disc brake and the subsequent simulation of strength and thermal analysis during car braking [5-13].

2 Experiment

The aim of the work was to create a 3D model of a disc brake of a passenger car, then to carry out a simulation of thermal and strength analysis during braking in the SolidWorks software environment. In order perform the analy-ses, it was necessary to design and create a 3D model of the disc brake, which consisted of two parts, namely the hub and the brake disc (Fig. 1). After creating the models of the hub and brake disc, they were then combined into one assembly, by defining the contact links between the hub and the brake disc. Then the selected material was applied to the model.

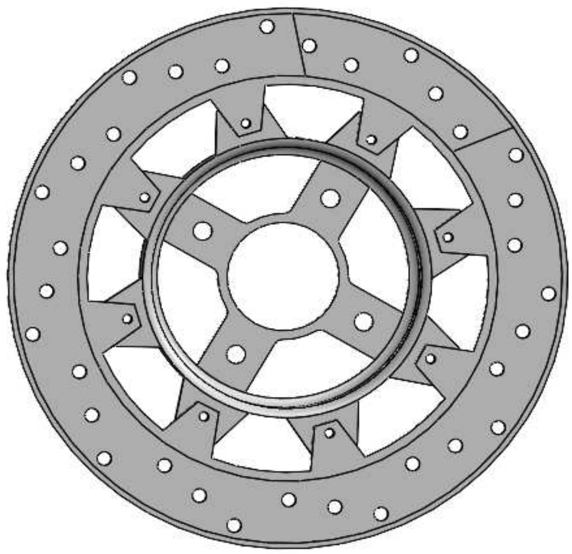

Fig. 1 *3D model of a disc brake*

After modeling of the 3D model, we were able to perform a thermal analysis simulation, where we defined the pa-rameters of the car braking: initial speed, the time needed to stop, the weight and the coefficient of friction betwe-en rubber tire and asphalt. Then we applied thermal boundary conditions: heat flow, thermal energy and ambient temperature.

3 Transient Thermal Analysis

The aim of the thermal analysis was to determine the temperature distribution on the brake disc and display the increase in temperature during braking. In the thermal analysis, we consider that the car stops from a given speed to a complete stop. The following input parameters were initial for the thermal analysis:

- Speed of the car: v = 40 $m.s^{-1}$
- Car weight: m = 1400 kg
- The frictional coefficient (asphalt – rubber): μ = 0.73.
- Dry asphalt road: coefficient of about 0.7 to 0.8, wet road: A bit slippery, with a reduced coefficient ranging from 0.4 to 0.6, icy road: significantly lower coefficient, often below 0.2.

Parameters for the analyse are the friction force and the needed time to stop the car which were calculated from the given data.

This maximum friction force is calculated as follows:

$$F_f = \mu \, . m \, . g \tag{1}$$

Calculating with the given data to get:

$$F_f = (0{,}73).(1400\,kg).(9{,}81\,m.s^{-2}) = 10\,025{,}82\,N \tag{2}$$

Where:
F_f…Friction force [N],
μ...Frictional coefficient [-],
m…Weight [kg],
g…Gravitational acceleration [$m.s^{-2}$].

Acceleration (deceleration):

$$a_x = \frac{F_f}{m} \tag{3}$$

Calculating with the given data to get:

$$a_x = \frac{10025{,}82\,N}{1400\,kg} = 7{,}16\,m.s^{-2} \tag{4}$$

Where:
a_x…Acceleration [$m.s^{-2}$].
F_f ...Friction force [N],
m…Weight [kg].

Time needed to stop:

$$t = \frac{v_x}{a_x} \tag{5}$$

Calculating with the given data to get:

$$t = \frac{40\,m.s^{-1}}{7{,}16\,m.s^{-2}} = 5{,}58\,s = 6\,s \tag{6}$$

Where:
t…Time acceleration [s],
v_x…Speed [$m.s^{-1}$],
a_x…Acceleration [$m.s^{-2}$].

Other material and environment related input parameters:

- Material: Alloyed gray cast iron,
- Convection coefficient: 90 $W.m^{-2}K^{-1}$,
- Ambient temperature: 20°C, 30°C.

Analysis procedure in the SolidWorks program:

- We selected the type of study. The value of the time needed to brake the car represents the time of the analyzed section.
- Convection coefficient – all surfaces of the brake disc were marked for which the heat flow coefficient of 90 $W.m^{-2}K^{-1}$ was defined. The ambient temperature was also specified at this step. The actual convection coefficients and ambient temperature can be calculated by performing CFD analysis or from experiments.
- Thermal energy – when braking the vehicle, the brake disc rotates and the brake pads rest on the surface of the rotor, which causes friction and creates thermal energy. A large part of the car's kinetic energy is converted into heat energy through the brake pads. Thermal energy will be generated on the brake disc in the area where the pads are in touch with the brake disc (Fig. 2). Its value can be calculated from the amount of kinetic energy given by the car's movement.

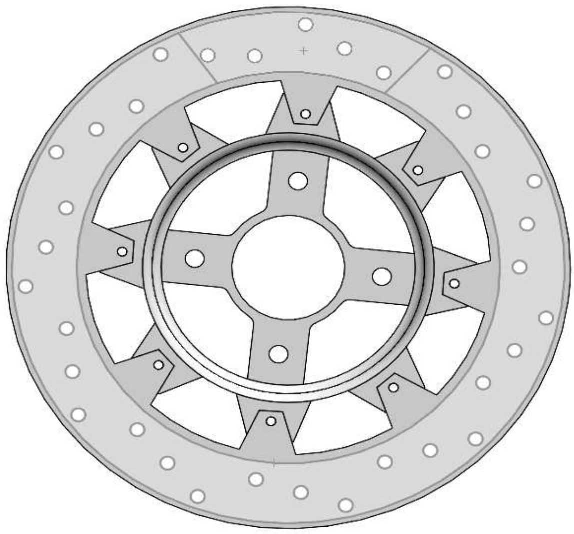

Fig. 2 *Display of area where the pads are in touch with the brake disc*

$$W_K = \frac{1}{2}\,mv^2 \tag{7}$$

$$W_K = \frac{1}{2}\,(1\,400\,kg)(40\,m.s^{-1})^{\,2} = 1\,120\,kJ \qquad (8)$$

Where:
W_K…Kinetic energy [kJ],
m ... Weight [kg],
v… Speed [m.s-1].

$$P = \frac{W_K}{\Delta t} \qquad (9)$$

$$P = \frac{1\,120\,kJ}{5{,}58\,s} = 200{,}71\,kW \qquad (10)$$

Where:
P…Heat power [kW],
W_K…Kinetic energy [kJ],
t…Time [s].

Since the analyze of the brake disc will be carried out only on one wheel and approximately 60 % of the vehicle's weight is in the front, the power value will be reduced.

$$P = \frac{200{,}71kW.(0{,}60)}{2} = 60{,}21\,kW \qquad (11)$$

3.1 Analysis results

The result of the thermal analysis was the temperature distribution during braking to a complete stop as it is shown on the color map (Fig. 3). The areas with the highest temperature are at the points of contact of the brake disc with the brake pad. The highest temperature was 123 °C - ambient temperature: 20 °C and 133 °C - ambient temperature: 30 °C (Fig. 4). The graph shows the increase in temperature during the period of car braking.

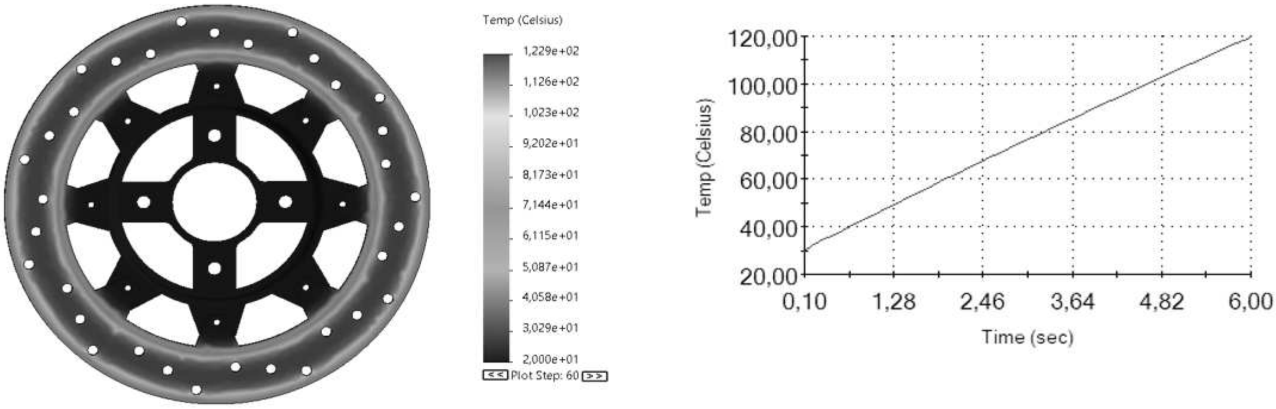

Fig. 3 *Display of analysis results - Ambient temperature: 20 °C*

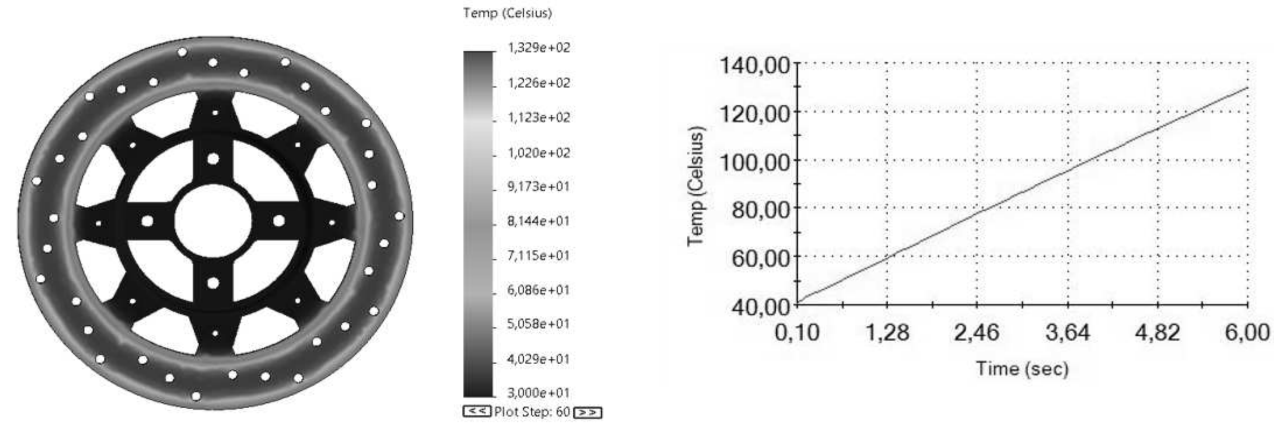

Fig. 4 *Display of analysis results - Ambient temperature: 30 °C*

4 Static study

The goal of the static study was to determine the deformation of the brake disc when the brake pad is pressed against the brake disc.

The following types of fixtures were used:

- Fixed geometry – the face on the inner diameter of the rotor hub where it would be mounted to a shaft was selected.

- Roller/Slider – the face on one side of the rotor where the fixed brake pad will rub the rotor was selected.

The following input parameters were initial for the static study:

- The pressure of the brake pads on the brake disc: 1.10^6 N.m^2 – the area where the brake pad will act against the brake disc was marked. This value can be known from experiment [7].
- The friction coefficient (brake pads – brake disc): μ = 0.4. (Normally, the friction coefficient of ordinary brake pads is about 0.3 to 0.4, while the friction coefficient of high-performance brake pads is about 0.4 to 0.5. With a higher friction coefficient, you can generate more braking force with less pedaling force and achieve better braking effect. But if the friction coefficient is too high, it will stop suddenly without cushioning when you step on the brakes, which is also not a good state).
- The frictional force – two split faces on the rotor where the brake pads make contact was chosen.

The frictional force is calculated using the area of the pad application.

$$F_f = \mu \, . \, F_N \tag{12}$$

$$F_f = (0{,}4).\left(1\,.\,10^6 \frac{N}{m^2} \cdot 1{,}2714\,.\,10^{-4} m^2\right) = 50{,}8\,N \tag{13}$$

Where:
F_f…Frictional force [N],
μ…Friction coefficient [-],
F_N…Braking force [N].

Thermal loading – the temperature distribution, which was obtained from the results of the thermal analysis, will be further transferred to the static study.

Plot stress:

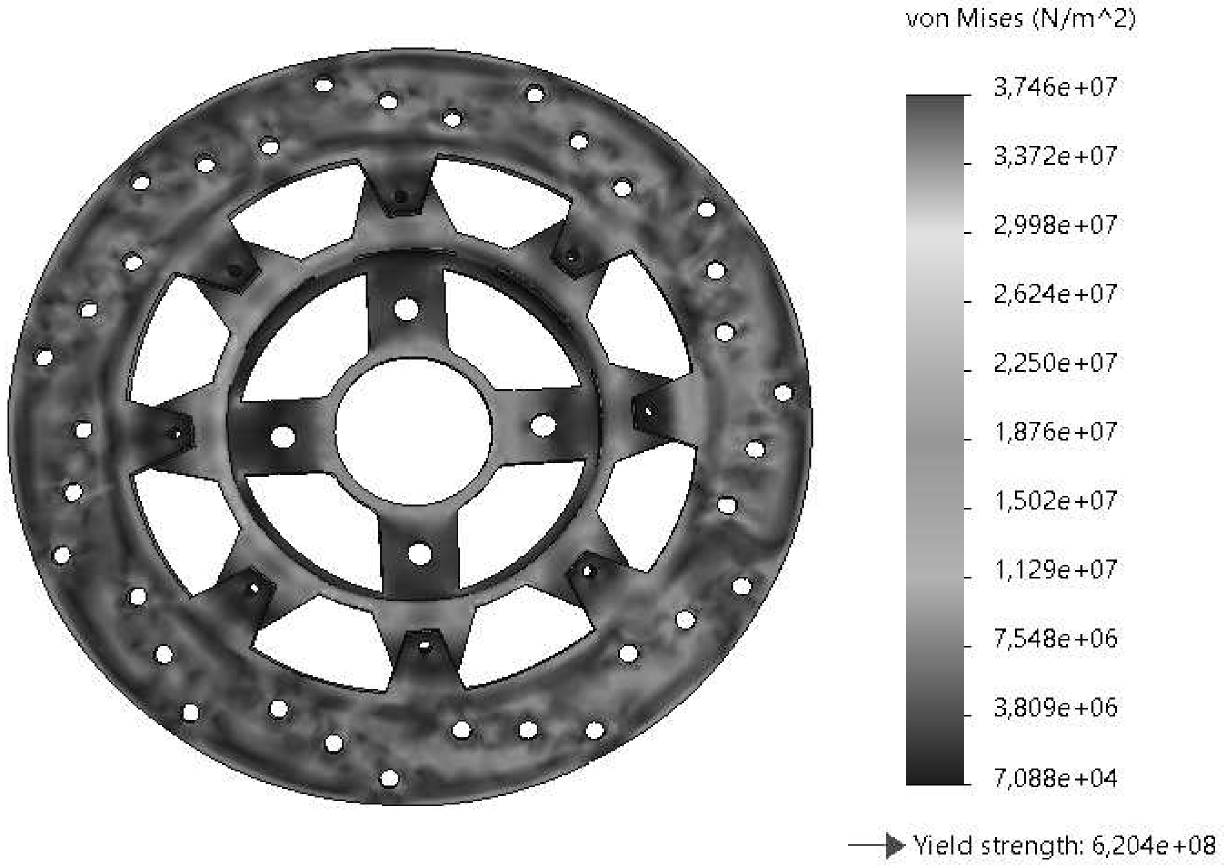

Fig. 5 *Plot of the von Mises stress*

The results of the analysis are shown with a color map (Fig. 5). The largest displacement was 0.15 mm (Fig. 6).

In Fig. 6 is shown plot displacement.

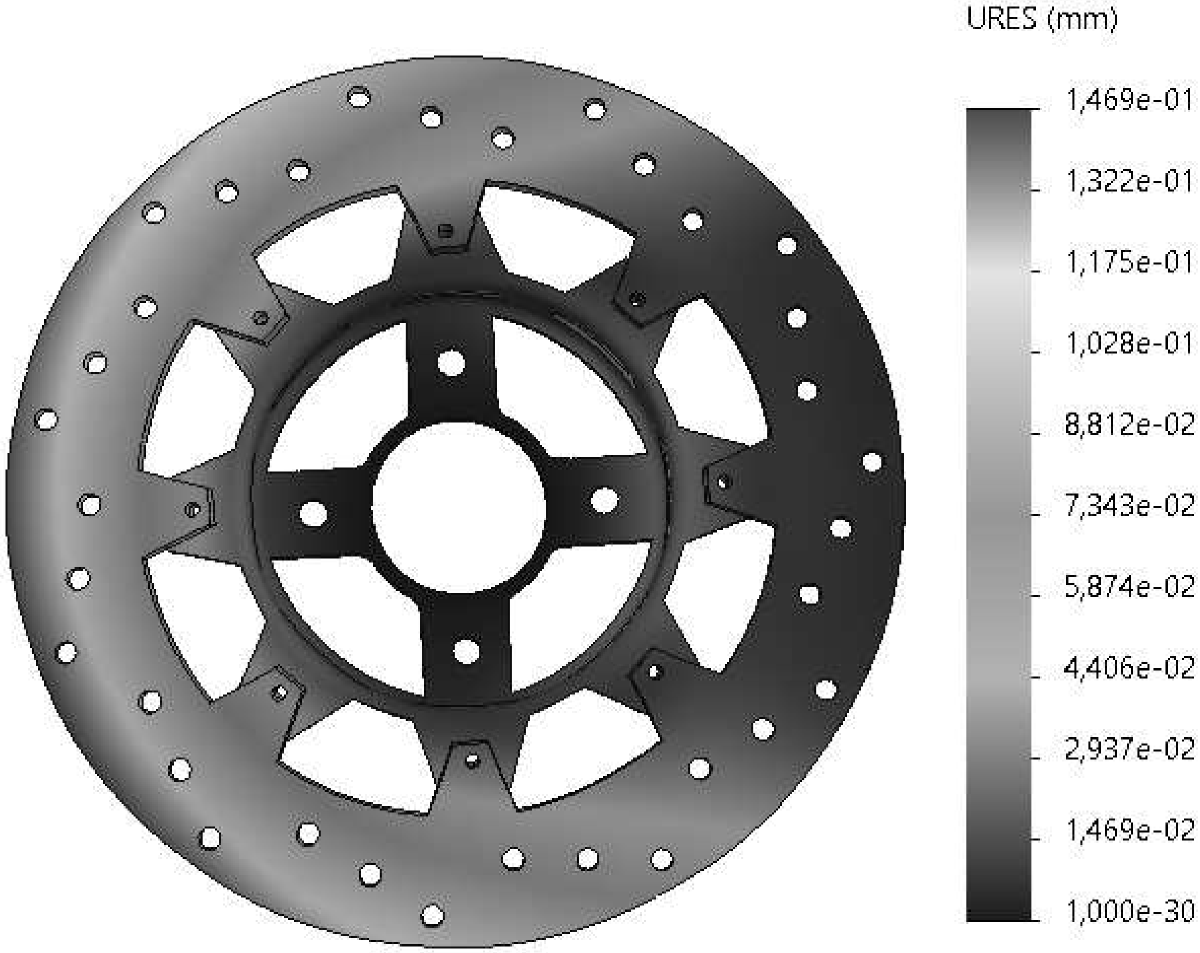

Fig. 6 *Plot displacement*

5 Conclusion

In order to carry out the analyses, it was necessary to design and create a 3D model of the disc brake, which con-sisted of two parts, namely the hub and the brake disc. After creating the models of the hub and brake disc, we then combined the models into one assembly, by defining the contact links between the jaw and the brake disc. Next, the selected materiál was defined. After modeling the 3D model, a thermal analysis simulation was performer. Af-ter starting the thermal study, the speed of the car was defined, the time needed to brake the car, the weight of the car and the coefficient of friction between rubber and asphalt, then the thermal boundary conditions were applied. Heat flow, thermal energy and ambient temperature. In both types of analysis, as a final step, a software-generated mesh of triangle-shaped polygons was created. After the thermal analysis simulation, the temperature distribution on the disc brake during braking until the car comes to a complete stop can be identified. The highest temperature was on the contact surface of the brake disc with the brake pad and reached 123 °C - ambient temperature: 20 °C and 133 °C - ambient temperature: 30 °C. To start the strength analysis, the coefficient of friction and the pressure of the brake pads on the brake disc, clamping of the disc geometry, rotational-sliding connection, reference temperature at zero stress have to be defined. After finishing the strength analysis simulation, the sulting stress and displacement can be identified. The largest displacement was 0.15 mm.

For better results, further studies on the effects of thermomechanical contact between the brake disc and brake pads can be carried out in the future, such as an experimental study to verify accuracy, triboogy and a study of dry contact sliding under a macroscopic aspect.

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