Multiphysics Analysis of an Automobile Disc Brake

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Introduction: Disc brakes systems are typically used in the automotive industry for stopping or decelerating vehicles by forcing brake pads onto the surface of the brake discs. The frictional heat developed during the braking action can cause brake failure, wear and thermal cracks. This study investigates the thermal and mechanical effect of a braking action on a ventilated carbon ceramic brake disc assembly and determines the best vane profiles for maximum cooling and dissipation of heat.

Figure 1. Disc brake assembly

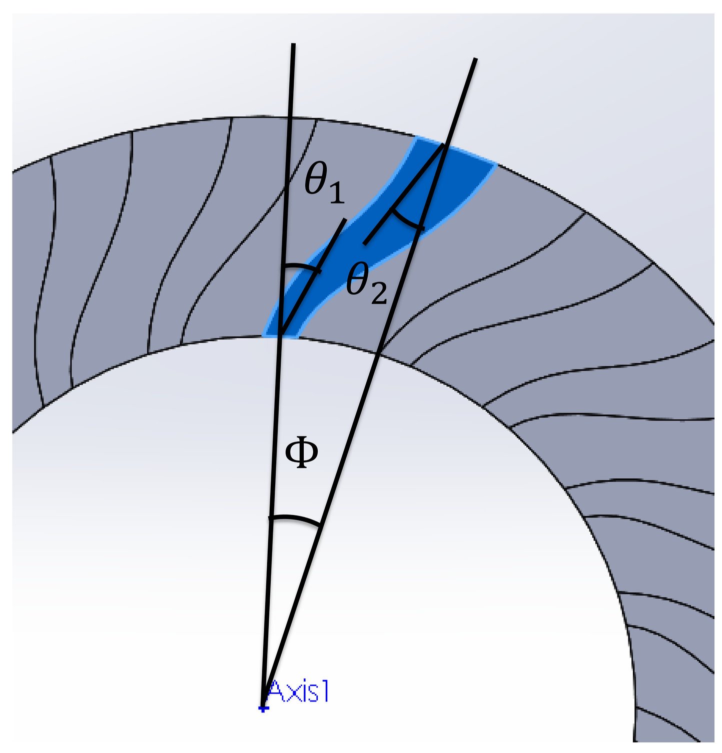


Figure 2. Parameters for vane profile

Computational Methods: Two different domains have been set up to determine the optimal curve profile of the vanes. Domain 1 represents the disc brake itself while domain 2 represents the region in which air is flowing. With a single brake, the theoretical temperature increase of the brake disk can be calculated as –

$$T_{th} = \frac{(1-\emptyset)}{2}\left[\frac{W({V_1}^2 - {V_2}^2}{2\rho_r c_r v_r}\right]$$

where ∅ is the percentage of braking , W is the vehicle weight, V_1 and V_2 are the initial and final velocity of the vehicle, c_r, v_r, and ρ_r are the specific heat capacity, volume and density of the disk brake. Newton's cooling model can be used to model cyclic loading and the maximum temperature for cyclic braking of 50s can be calculated as

$$T_{max} = T_\infty + \frac{\Delta T}{1 - \exp(-\frac{h_{CR}A}{WC_p}t_1)}$$

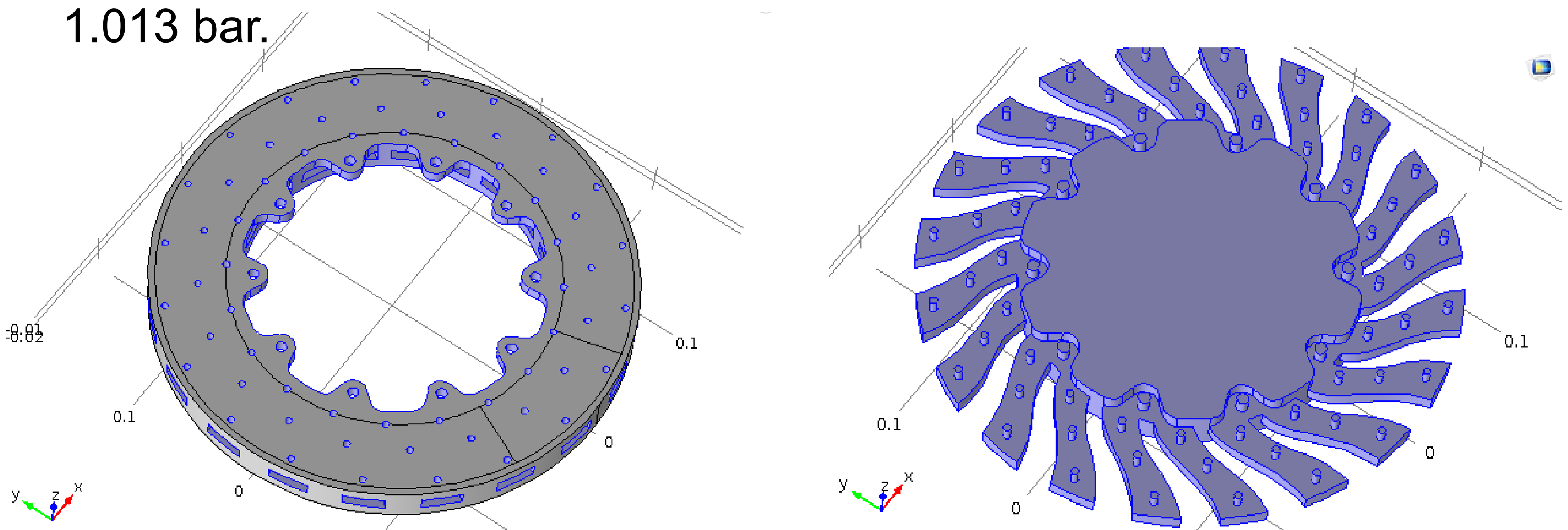
Where T_1 represents the initial temperature, T_∞ the environmental temperature and h_{CR} represents the overall coefficient of heat transfer. A and W are the lateral surface area and mass of the object.

An average temperature increase of 650K is calculated and applied on the disc brake surface. The mechanical stresses on the disc brake is due to the applied nominal clamping pressure of 2.5Mpa and tangential frictional force generated in the brake pad region given by

$$F_b = 2\mu F_N$$

F_N represents the nominal braking force which can be computed form the clamping pressure and brake pad area. μ is the coefficient of friction between the materials and is usually considered to be 0.4

The airflow through the disc brake assembly is simulated using an inlet pressure of 1.01274 bar and outlet atmospheric pressure of 1.013 bar.



(a) Domain 1 - solid disc brake (b) Domain 2 - air flow

Figure 3. Computational domains

The mid-section vane profile is optimized to have maximum air flow for better cooling effect. The curve profile is represented by 3 parameters: Φ, θ_1 and θ_2 as shown in figure 2.

Results: The multiphysics simulations clearly shows how change in vane profile can affect the temperature (Figure 4) and stress distribution (Figure 5) within the assembly. By sweeping various combinations of parameters for the vane profile an optimal design is obtained for $\Phi = 15°$, $\theta_1 = 20°$ and $\theta_2 = 10°$.

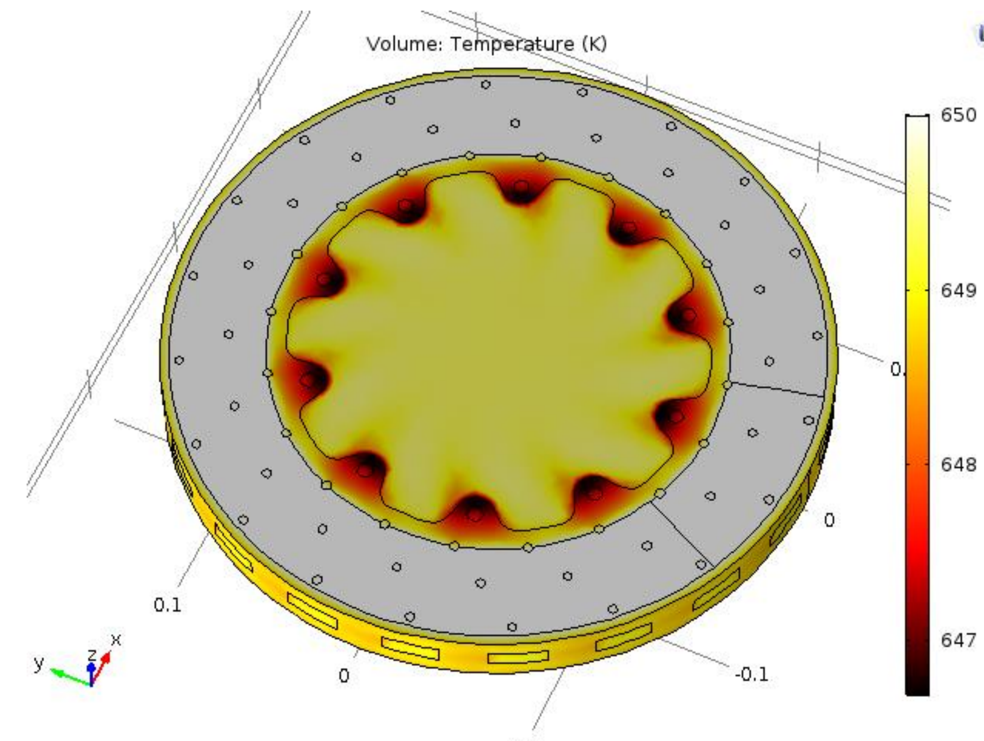


Figure 4. Temperature distribution during braking

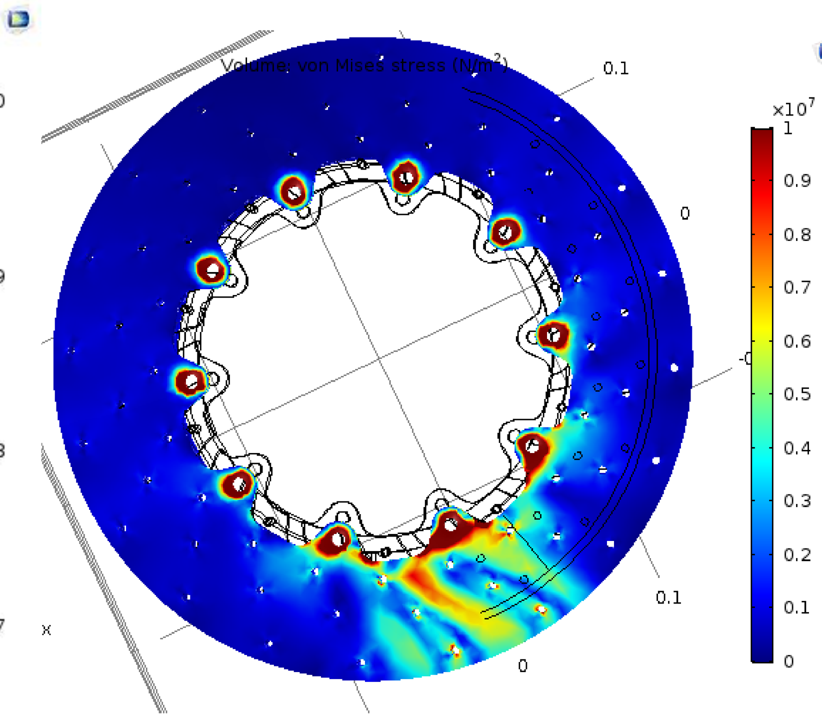


Figure 5. Von Mises stress distribution during braking

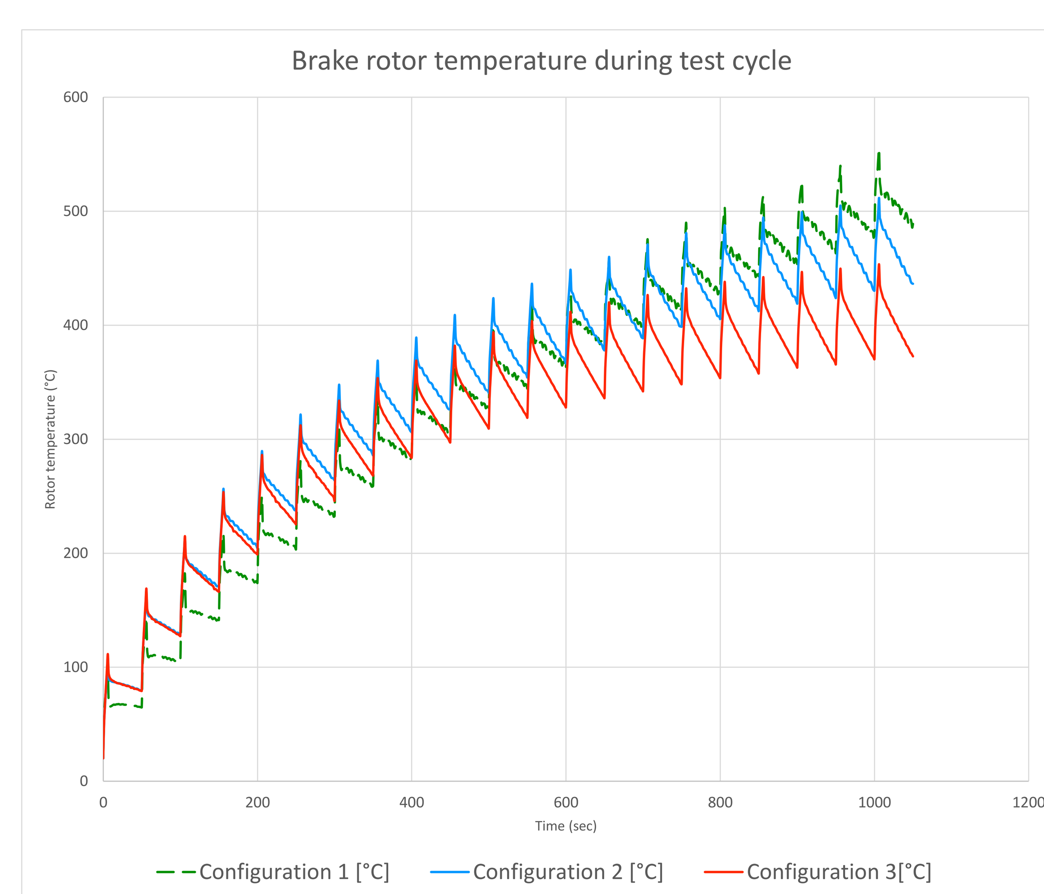


Figure 6. Temperature rise during test cycle

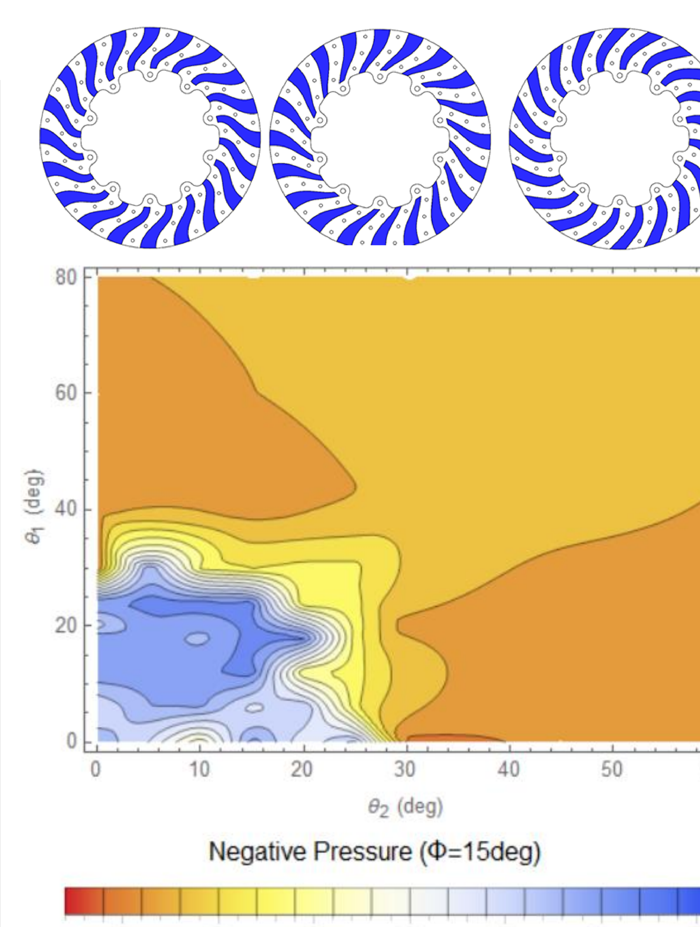


Figure 7. Pressure difference with different vane profiles

Conclusions: The COMSOL simulation study shows the effect of brake disc design on its temperature distribution. Optimizing the design by simply changing the vane profile can significantly improve the temperature dissipation within the brake disc and prevent failure of ceramic brakes at high temperatures. Certain approximations have been used in this study such as using calculated average heat and using fixed inlet and outlet pressure values for air flow. The simulation results can be dependent on the assumptions used such as heat flux and pressure inputs, physical parameters such as materials used and design of assembly and numerical parameters such as element size.

References:

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