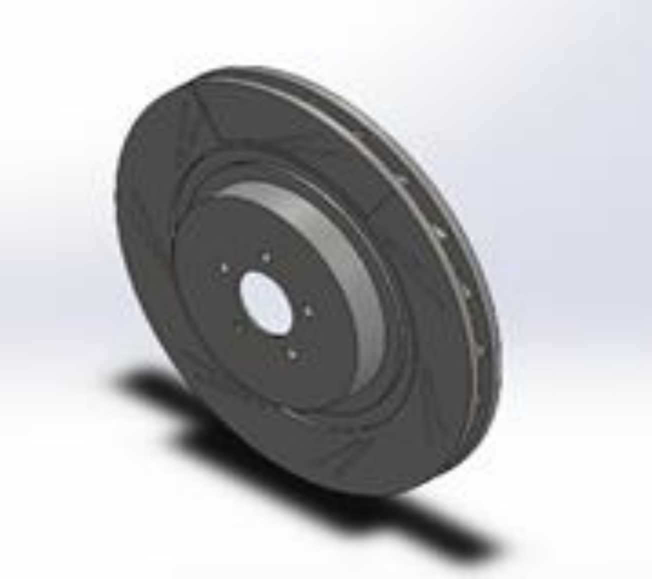
Disc Brake FEM Analysis



Under heavy duty braking conditions

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Introduction

- Conventional braking systems use 100% of the friction braking to decelerate a moving vehicle, meaning that the energy produced by the braking event is dissipated as thermal energy
- A regenerative braking system (RBS) allows a vehicle to recover significant amounts of the braking energy. A traditional brake system converts the kinetic energy from the vehicle in to mechanical energy and then releases it as heat. The RBS allows for the energy that would otherwise be converted to waste heat to be recovered as electrical energy.
- Regenerative braking holds many advantages for electric vehicles (EVs). This system can extend the brake disc’s life, minimizes the disc weight, and minimizes the brake pad wear.

Introduction (cont.)

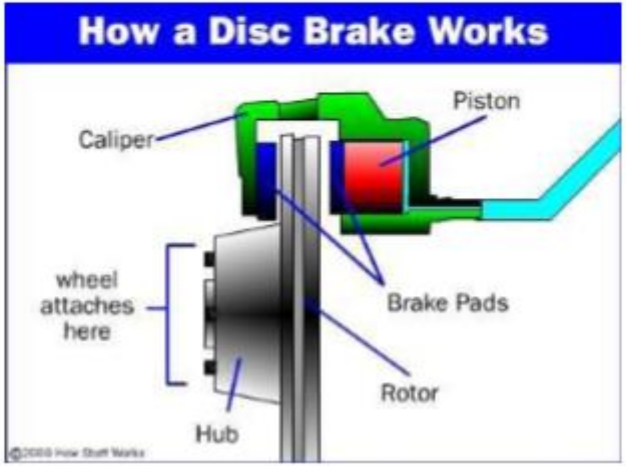
- Although there are advantages to the RBS as mentioned in the previous slide, alone it cannot function properly emergency situations. Due to these limitations the RBS has to work alongside a conventional braking system.
- An RBS lessens the burden experienced by a brake disc which could allow for a lightweight brake disc to be used alongside the RBS
- There are two different types of brake discs that are used in brake systems, solid discs and ventilated discs
  - For the purpose of this presentation, a ventilated disc will be analyzed

Problem Statement

The purpose of this analysis is to understand the behavior of the ventilated disc under heavy duty braking and to learn how to design a disc that would not fail under these conditions. As mentioned in the literature review, thermal stresses contribute greatly to failure in the disc brake. A thermal distribution and a thermal stress analysis will be conducted to simulate these conditions as well as a static analysis.

Problem Statement (cont.)

The reason for why these analyses were chosen is the brake pads exerts a pressure on a portion of the disc’s surface and that heat is generated due to the contact friction between the brake pads and the surface of the disc.



Problem Formulation

The vehicle was modeled to have mass of 1,500 kg and an initial speed of 100 km/hr with an initial disc temperature of 20^0C before applying the brakes. Based on the national average, the stopping distance was assumed to be 56 meters.

Applying the conservation of energy principle; In order for the vehicle to stop, the kinetic energy of the vehicle must be equal the thermal energy dissipated by the brakes.

$$E_{thermal} = KE_{vehicle}$$

$$KE_{vehicle} = \frac{1}{2}mv^2$$

$$E_{thermal} = F_{brakes} \times d$$

$$E_{thermal} = KE_{vehicle}$$

$$F_{brakes} = m \times a$$

$$\Delta t = \frac{v}{a}$$

$$Heat\ Power = \frac{KE_{vehicle}}{\Delta t}$$

$$Heat\ Power_{brake\ pads(1)} = \frac{(0.6)Heat\ Power}{2}$$

Calculated Parameters

KE(vehicle)	578,796 J
Friction Force	10,335 N
Acceleration	6.8 m/s^2
Time	4.03 s
Heat Power	143,621 W
Heat Power (brake pads)	43,086 W

Problem Formulation

The heat transfer coefficient was assumed to be 90 W/m^{2}K based on the experimental data conducted by papers from the literature review. The time constant β was experimentally found and used on the following equation:

$$h = \frac{mC_p}{\beta A} \ (W/m^2K)$$

Other methods of finding h, include flow analysis such as Limpert's equation:

Re<2.4 x 10^5 $$h = 0.7(k_a/D_a)Re^{0.55}$$

Re>2.4 x 10^5 $$h = 0.04(k_a/D_a)Re^{0.8}$$

Modes of Heat Transfer

- Convection heat transfer
    - Convection heat transfer can be found in the system due to the hot air from the vehicle as it is operating

$$q_{conv} = h * A_s * (T_s - T_\infty)$$

- Conduction heat transfer
    - - Conduction is occurring through the disc itself in this system

$$q_{cond} = -k * A_s * \frac{dT}{dx}$$

Formulas

For an isotropic material with no heat generation, the governing heat equation for cylindrical bodies under transient conditions is the following:

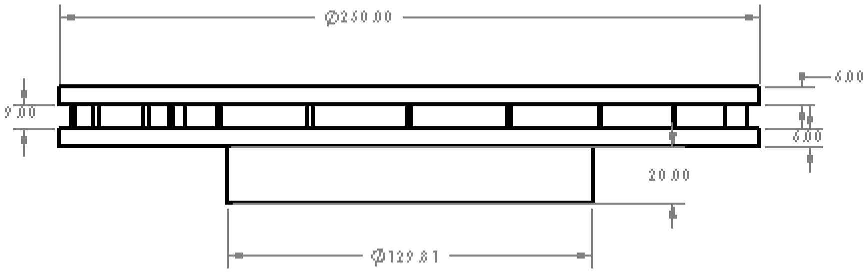
$$k_r \frac{\partial^2 T}{\partial r^2} + \frac{k_r}{r}\frac{\partial T}{\partial r} + \frac{k_\theta}{r^2}\frac{\partial^2 T}{\partial \theta^2} + k_z \frac{\partial^2 T}{\partial z^2} = \rho C_p \frac{\partial T}{\partial t}$$

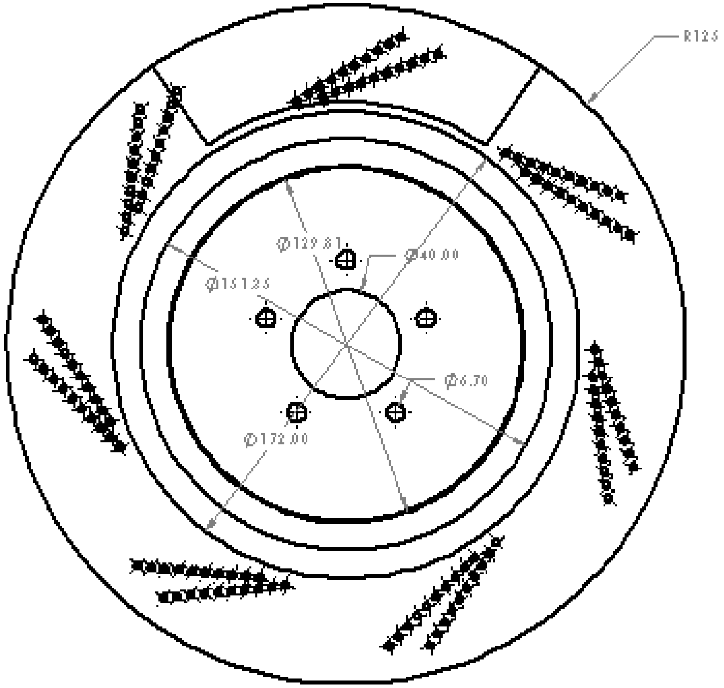
This equation can be used to analytically solve for the temperature distribution of a cylindrical body.

Model Dimensions

The dimensions for the disc brake are:

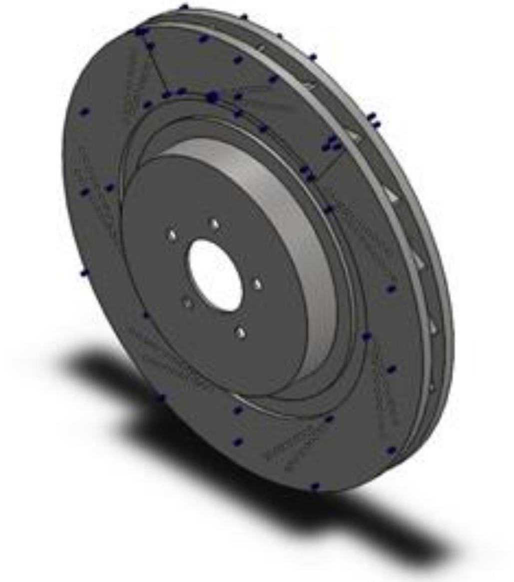
- 250 millimeter outer diameter with 172 millimeter inner diameter disc brake plates
- The face of the brake is made up of two 6 millimeter plates with 9 millimeters of





Model and Material Pro

- Cast Carbon Steel:
  - Yield Strength = 2.4817E8 Pascal
  - Elastic Modulus= 2E11 Pascals
  - Poisson’s ratio= 0.32
  - Thermal Expansion Coefficient= 1.2e-05 /Kelvin



- Model Properties:
  - Mass= 2.974 kg
  - Volume= 0.000381255 m^3
  - Density=7,800 kg/m^3

2 split lines were created to simulate the front and back brake pads.

Finite Element Modeling (Thermal)

Mesh details

- Curvature based mesh was applied to the figure
  - Maximum element sizes of 18 millimeter with mesh control applied for a minimum element size of 3.6 millimeter.

Total number of Elements: 53116

Finite Element Modeling (Static)

Mesh details

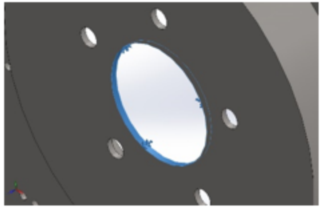
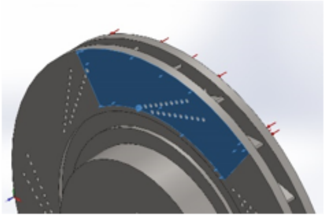
- Curvature based mesh was applied to the figure
  - Maximum element sizes of 15 millimeter with with a mesh control applied for minimum element size of 3 millimeter.

Total number of Elements: 61085

Boundary Conditions (Static)

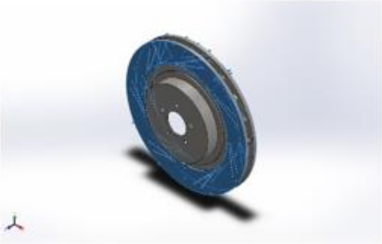
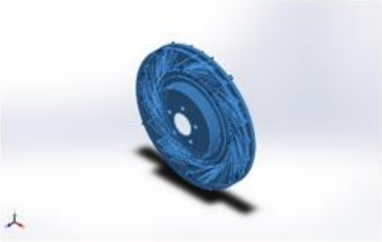
Fixtures applied:

- A fixed geometry was applied at the centermost hole similarly to how the brake disc is attached to the axle of the vehicle
- Roller slider fixture applied at the back-end split line

Fixture Image	Fixture Details
	Entities: 1 face(s) Type: Fixed Geometry
	Entities: 1 face(s) Type: Roller/Slider

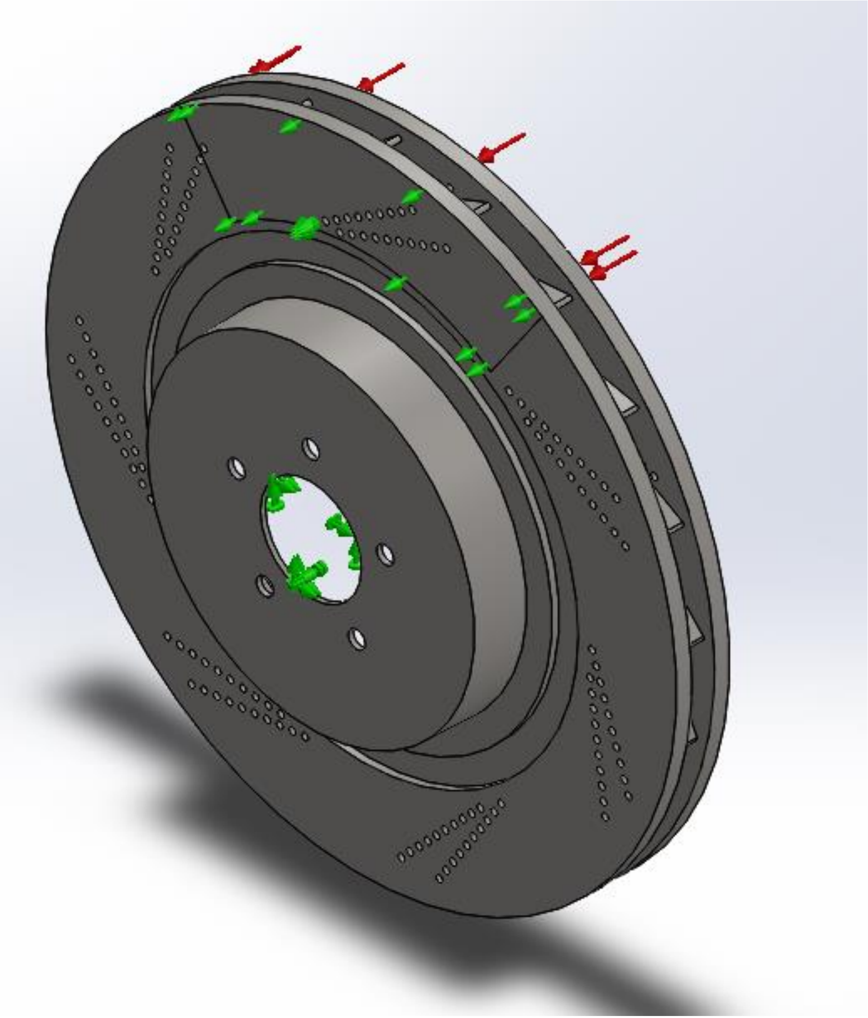
Thermal loads

- Loads
  - Convection coefficient of 90 W/m^2K
  - Heat Power of 43086 W that simulates the power exerted by one brake pad.
  - An initial temperature of 20^0C

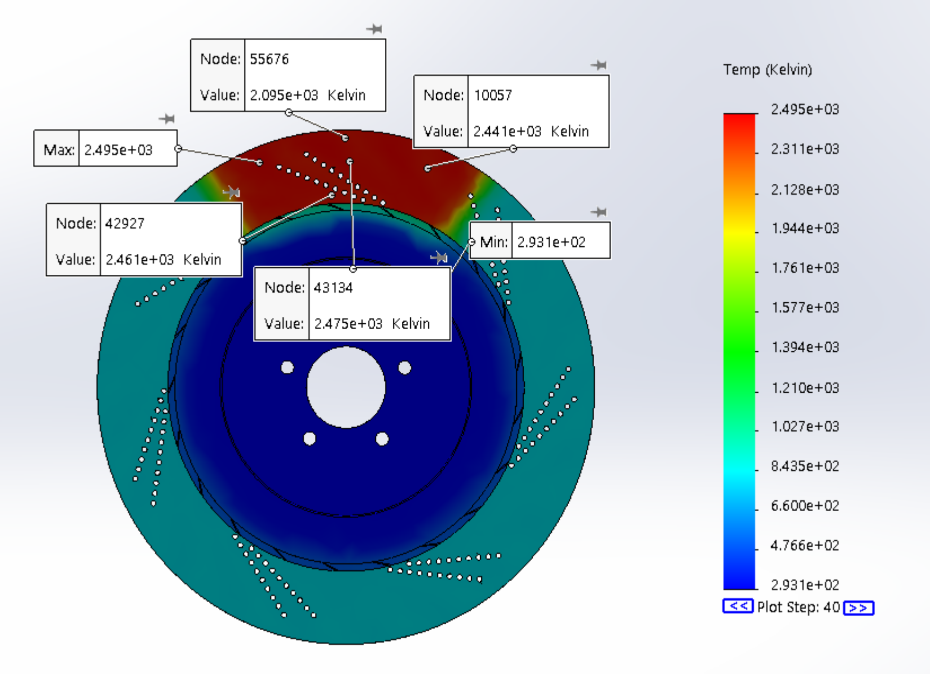
Load name	Load Image	Load Details
Convection		Entities: 389 face(s) Convection Coefficient: 90 W/(m^2.K) Time variation: On Temperature variation: Off Bulk Ambient Temperature: 293 Kelvin Time variation: Off
Heat Power		Entities: 4 face(s) Heat Power Value: 43086 W Time variation: on
Temperature (initial)		Entities: 1 Solid Body (s) Initial temperature: 20 Celsius

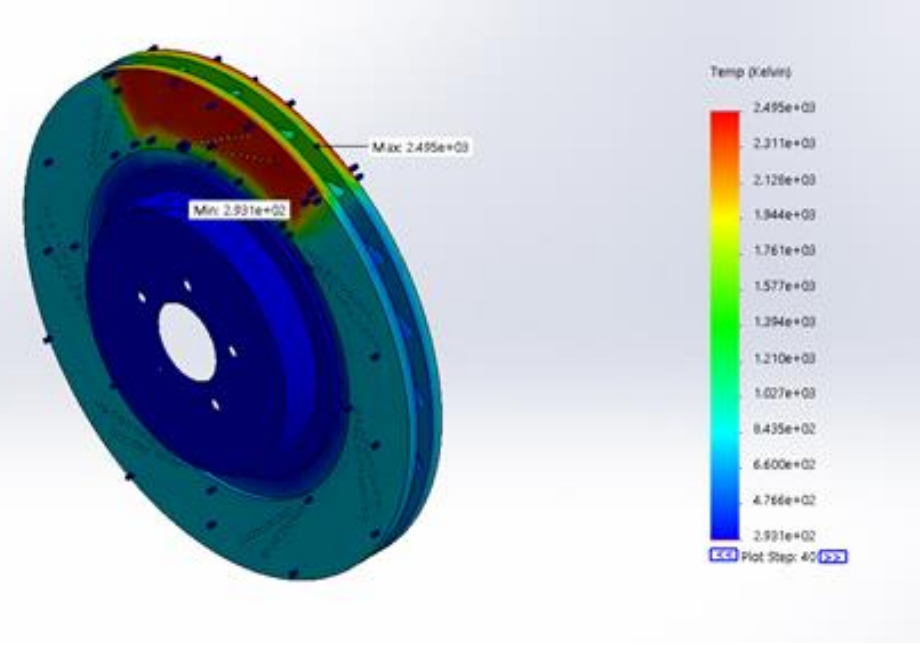
Static loads

- Loads
  - Pressure load of 4 MPa was applied to the area of the front end split line to simulate the pressure exerted by one brake pad
  - The red arrows represent the pressure load while the green markings represent the boundary conditions.
  - The temperatures of the thermal study were applied to another static study to see the thermal effects on the stress experienced by the disc.

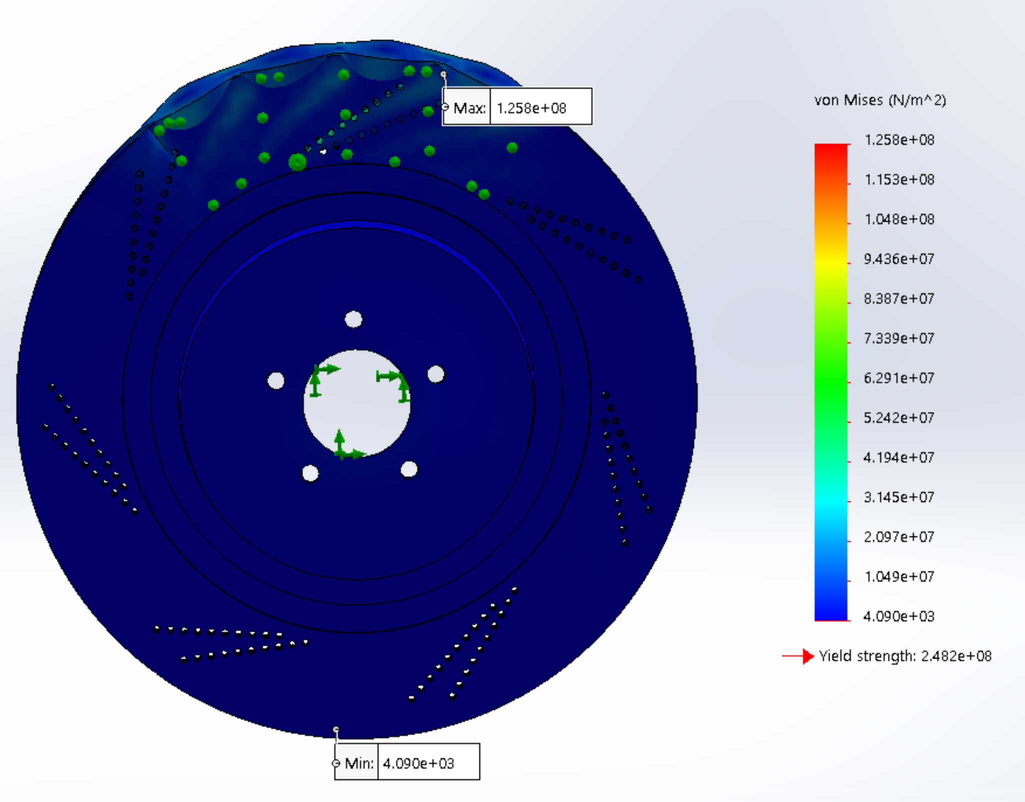


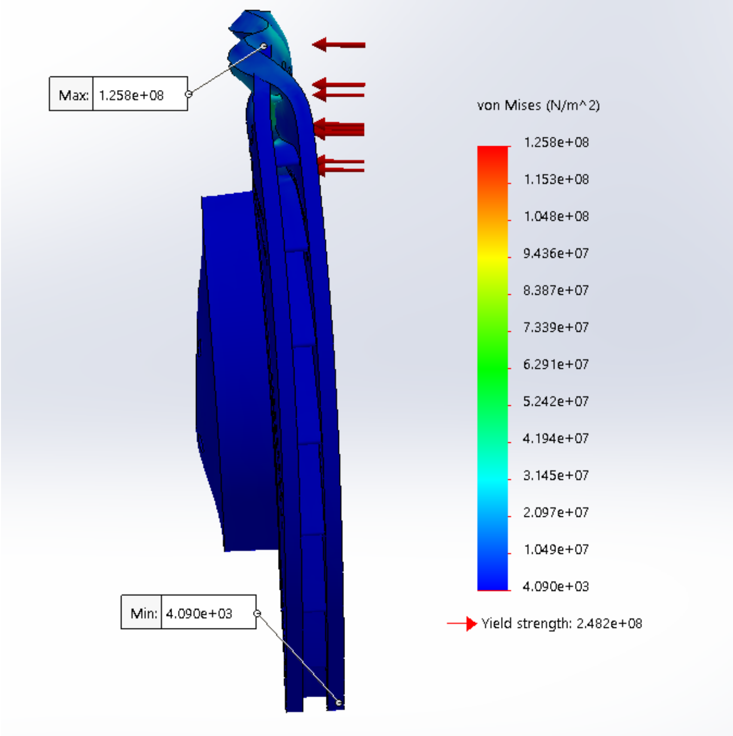
Temperature Distribution



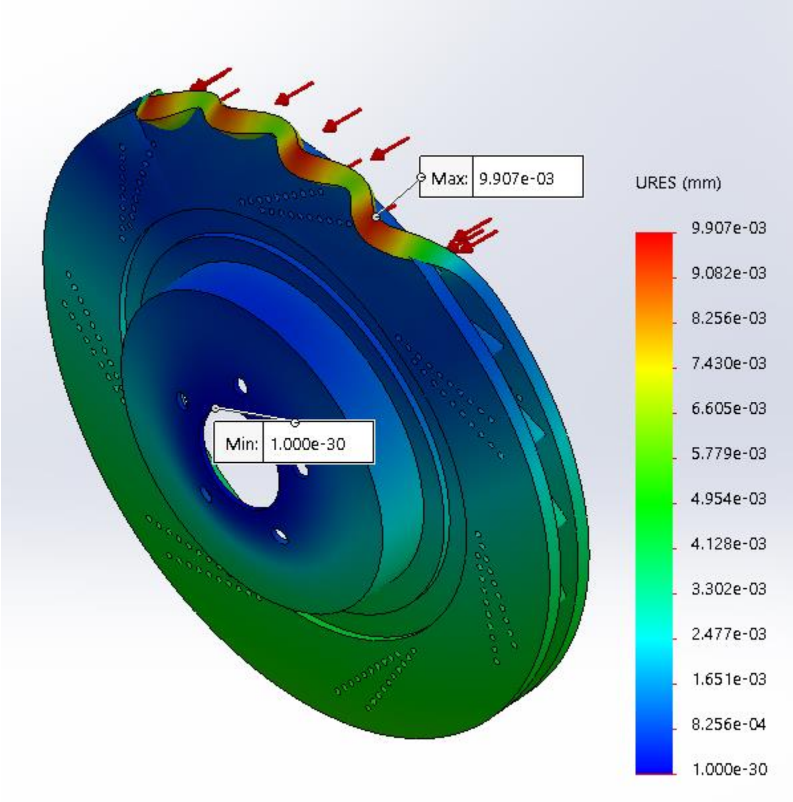


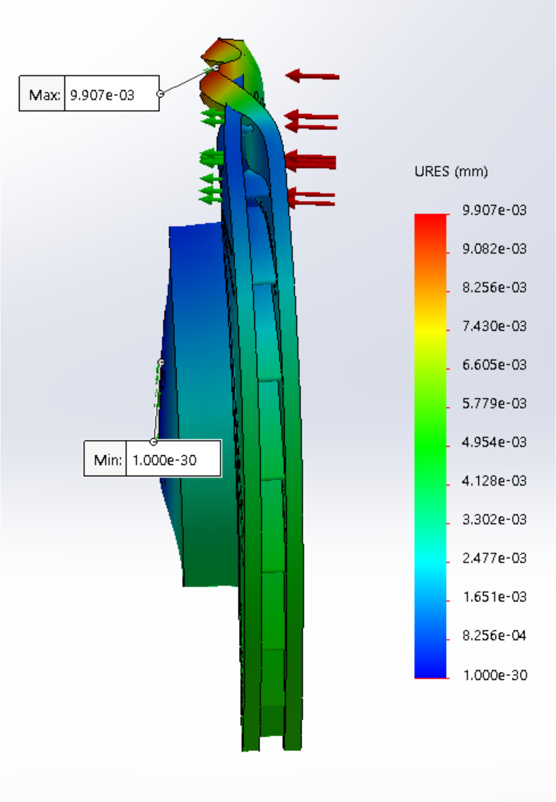
Stress Analysis (No Temperature



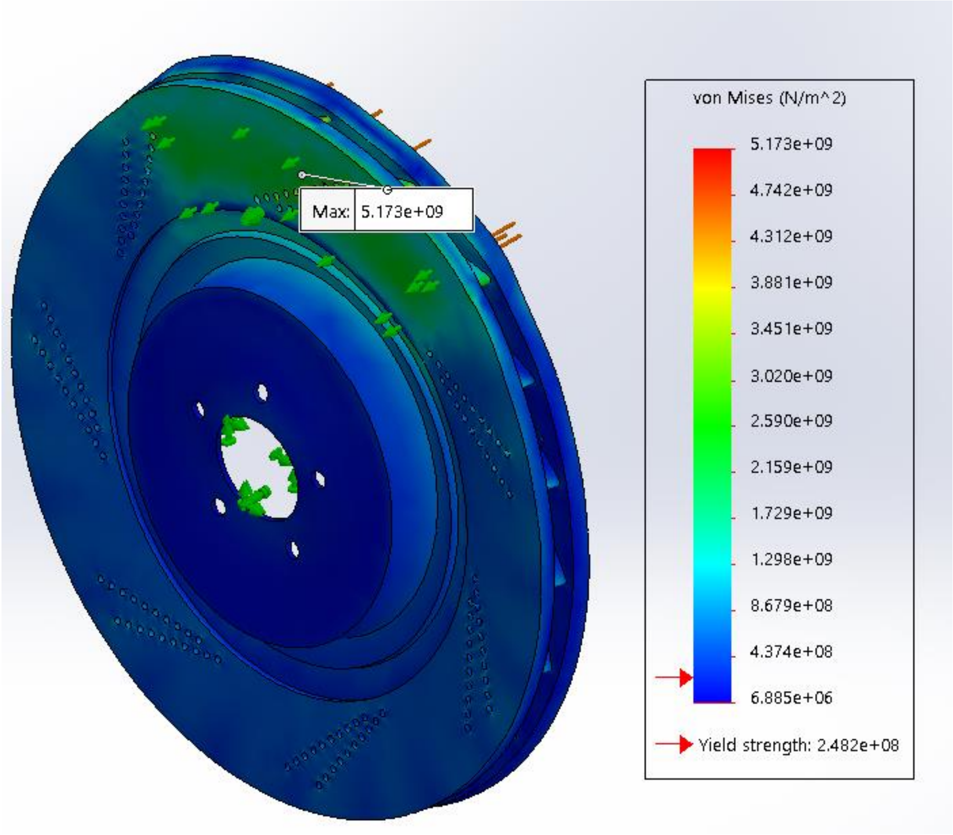


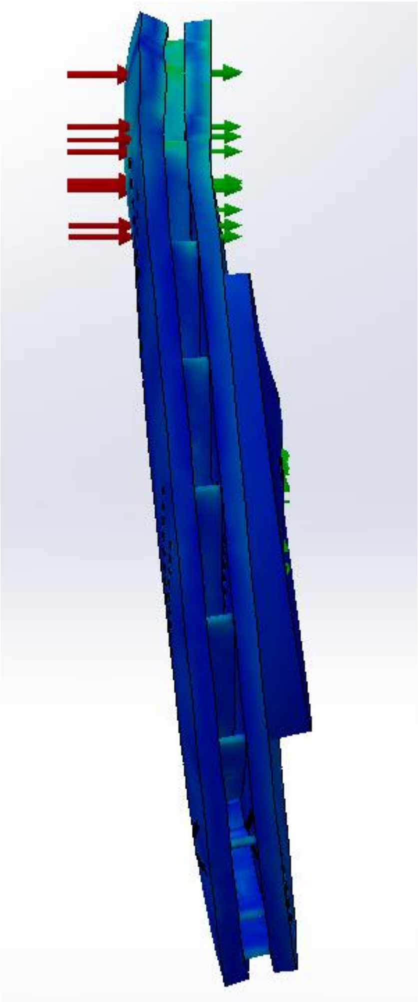
Static Deformation (No Temperature



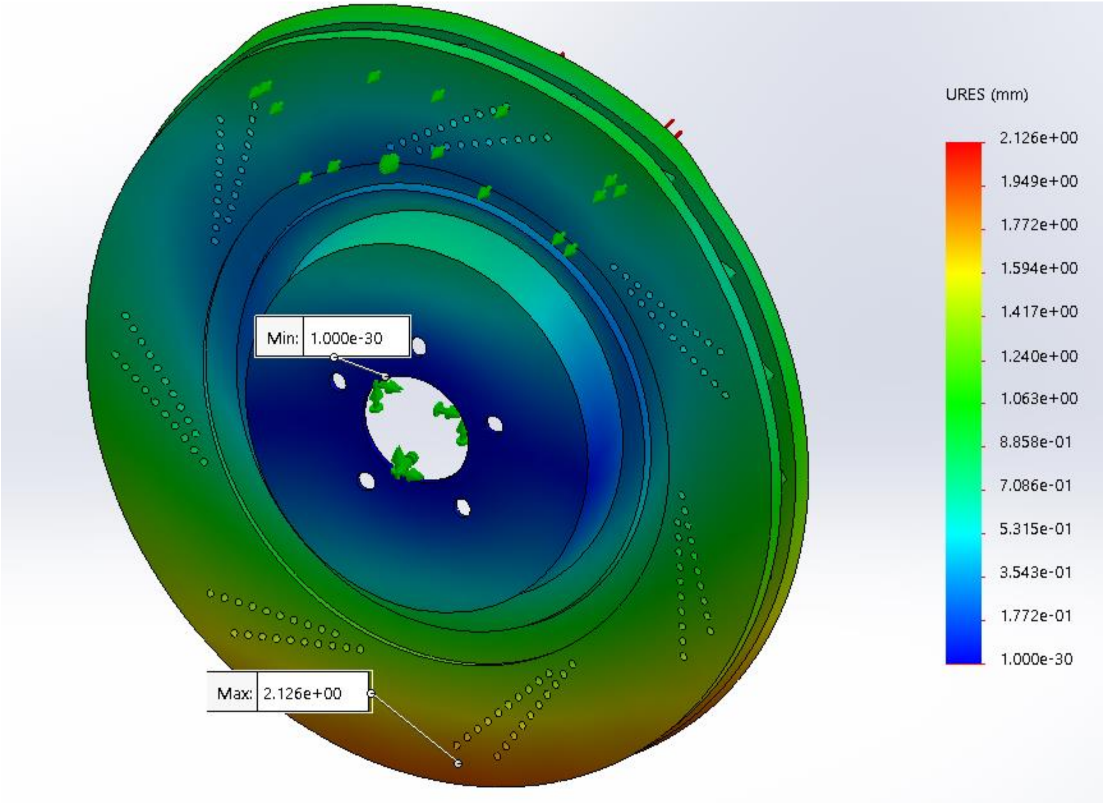


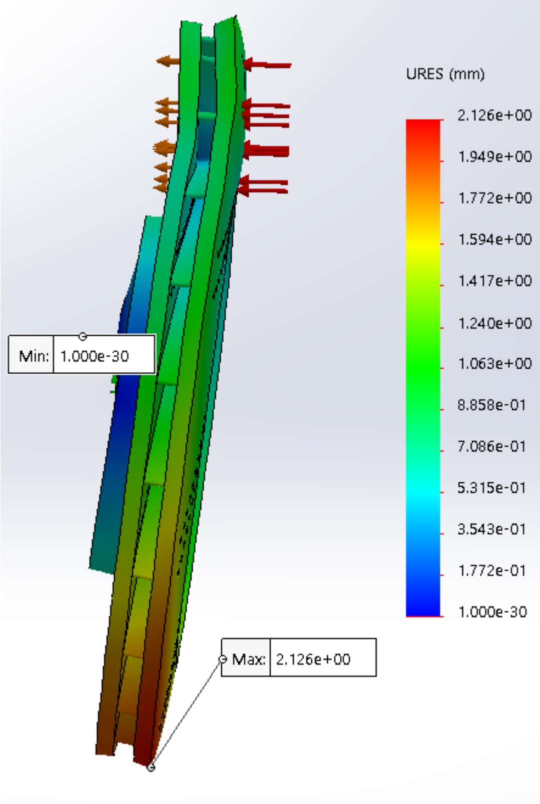
Thermal Stresses





Thermal Deformation





Sanity Check

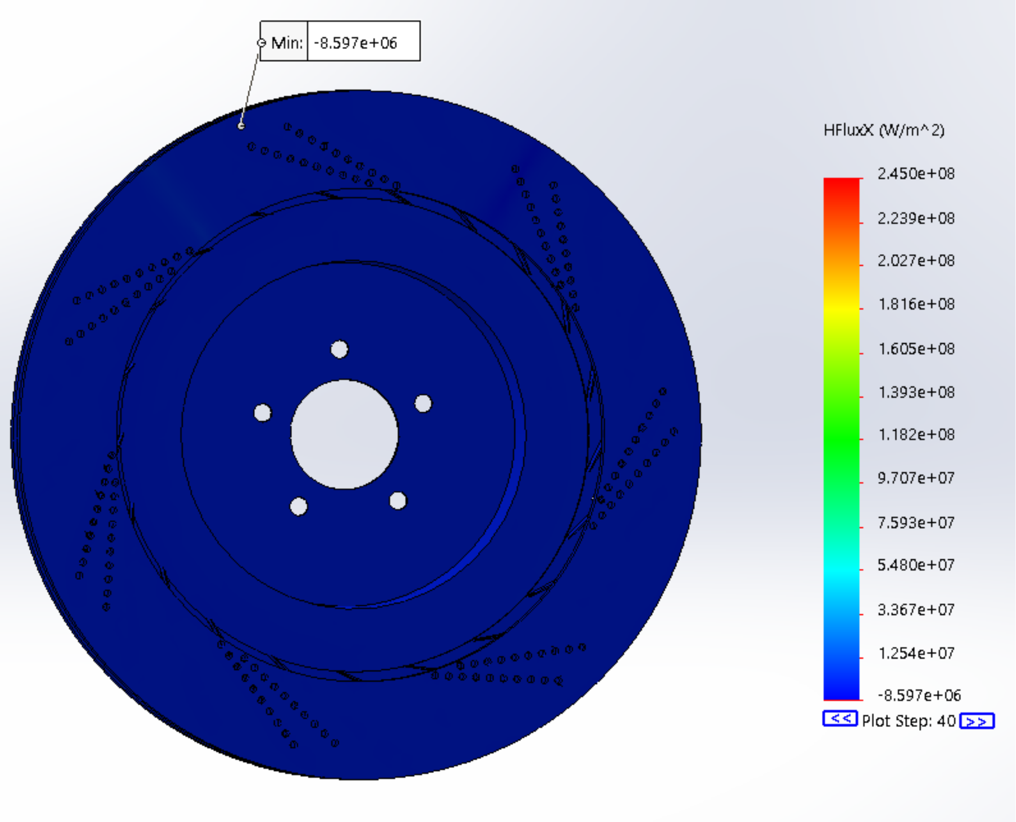
The average heat flux of the disc from the literature review was derived from the braking power over the disc frictional area.

$$q' = \frac{\mu F_{brakes} r_{eff} \omega}{A_d}$$

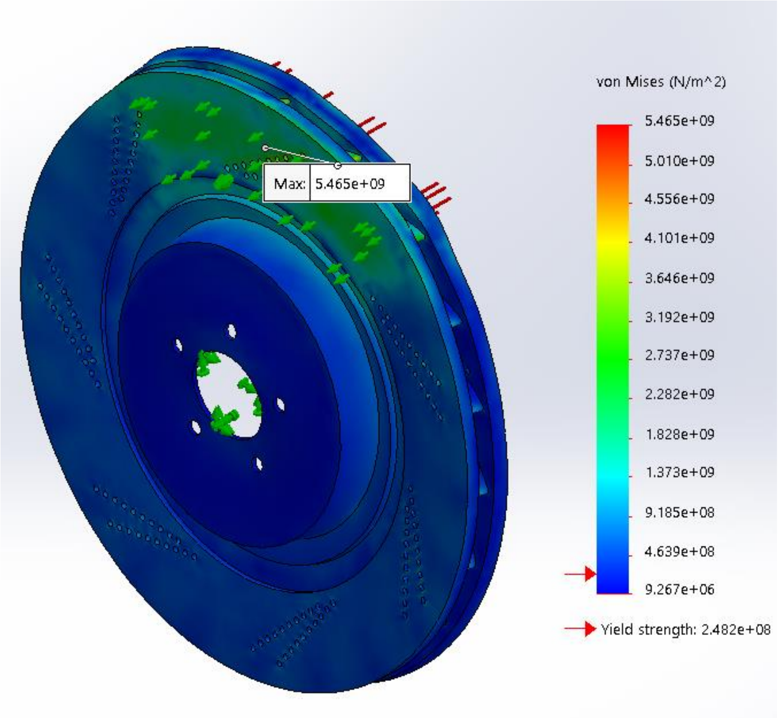
This equation can be modified to:

$$q' = \frac{Heat\ Power_{brake\ pad}}{A_d}$$

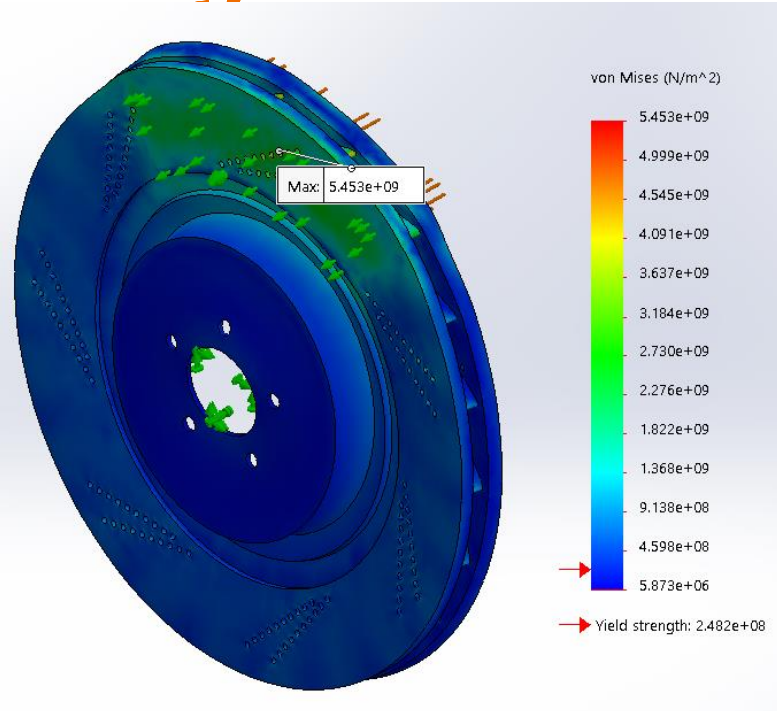
Heat flux equals 9.48E6 W/m^4



Thermal Stress Convergence

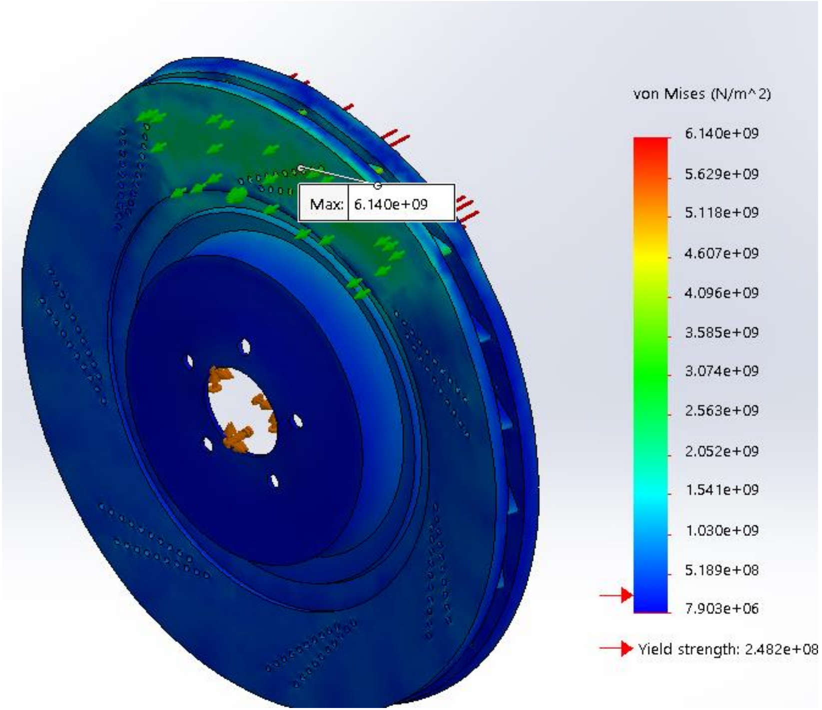


4934 Elements



75334 Elements

Thermal Stress Convergence (cont)



143712 Elements

Conclusions

Temperature distribution:

- The results indicate that the highest temperatures were found within surface area that would be in contact with the brake pads. Heat generated within the friction interface of the disc and pad caused the nodal temperatures within that area to increase.

Stress & Deformation (No thermal input):

- For the first static study, there was no thermal input. The reason for this was to see the effects the temperature distribution had on the stress. In this study the maximum Von mises stress was far below the yield strength. The maximum resultant deformation was 9.907E-3 mm.

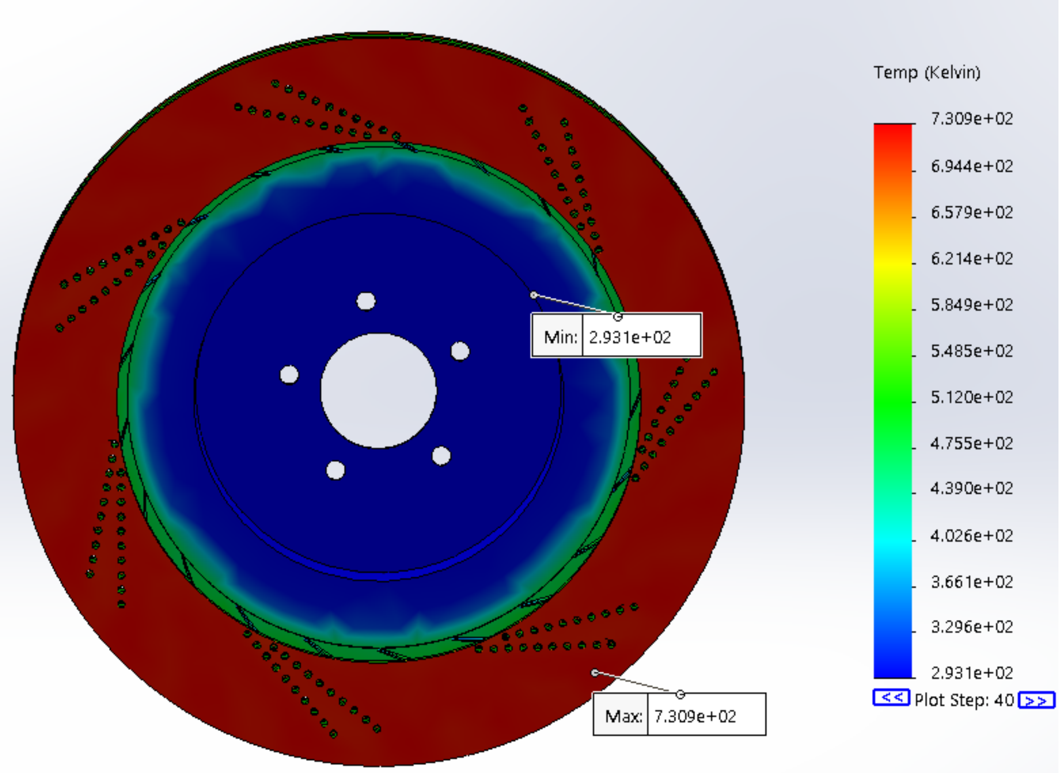
Conclusion

Thermal Stress & Deformation

- For this static study the temperature distribution of the thermal study was applied. This thermal input drastically changed the resultant stresses and deformation. The maximum Von mises stress was around 5.17E9 Pascal, a stress that is a magnitude bigger than the material yield strength. The maximum resultant deformation came up to 2.13 millimeters.
- It can be seen that the maximum deformation from thermal stress occurs around the outer edges of the disc
- Thermal stresses greatly contribute to the failure of the disc brake. Material properties to take into consideration when designing the disc should involve resistance to high temperatures.

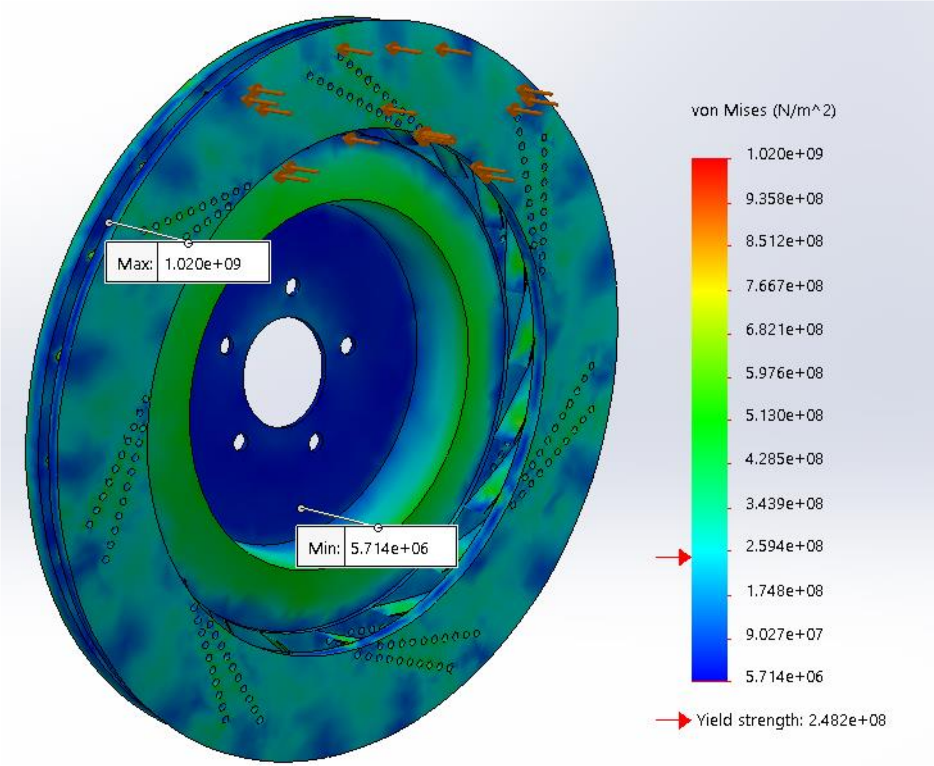
Temperature Distribution (suggested improvement)

Split lines were removed to more uniformly spread the heat power of the brake pads over the surface of the disc



Thermal Stress (New Thermal Input)

New thermal stress analysis with the new thermal distribution



References

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