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Optimization of thermal performance and weight of an automotive disc brake for a high performance passenger car.

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ABSTRACT: Brake judder is a major problem in the automotive industry due to high cost of rectifying it. This cost is a substantial fraction of the yearly warranty costs to vehicle manufacturers. Also, the expectations of customers with regards to comfort level have increased, especially for luxury vehicles. This NVH phenomenon can be classified into cold and hot judder. The latter, which is the focus of this research, accounts for almost a quarter of all brake judder concerns. In this research, a coupled thermal-structural analysis was first developed to predict the thermo-mechanical behaviour of an existing disc for a given braking condition. Thereafter, the geometry of the existing disc was parametrized into shape variables and the influence of each design variable on thermo-mechanical performance of the disc was studied using response surface methodology. Based on these surrogate models, an adaptive surface response optimization was used to derive optimum values of the chosen geometric parameters while minimising mass, thereby yielding an improved thermal performance.

KEY WORDS: Brake disc, finite element analysis, thermal performance, judder, optimization, DTV, temperature, weight.

1. Introduction

The performance of a vehicle braking system can be seriously undermined by excessive increase in temperature in the brake components, especially in the brake disc itself. This rise in temperature is produced by the heat generated from the relative sliding at the friction interface. The resulting thermo-elastic deformation within the disc can change the distribution of contact pressure and lead to thermal localization such as hot-banding and hot-spotting. The phenomenon is called thermo-elastic instability, and if severe, this can cause judder, as well as decrease in fatigue life of the disc.

One of the major problems that arises from the thermal loading of an automotive disc brake is hot judder. This phenomenon is caused by thermal deformation of the disc during braking. The thermal deformation of disc brake can be categorised into coning, waving, non-uniform thermal expansion, phase transformation and pad material deposition on the disc due to heating (Jacobsson, 2003). Coning is triggered by thermal distortion of the disc leading to its axial displacement. This then gives rise to increased disc run-out. The non-uniform expansion leads to increase in disc thickness variation, DTV about the circumferential rubbing path (Fieldhouse, 2013). Hot judder is usually associated with this thermal localisation phenomenon called hot spotting in which several hot regions are formed on the surface of a brake disc during high energy braking events (Tang et al., 2016). Under continuous non-uniform contact, the hotspots grow and produce hot judder vibration which makes driving difficult and dangerous (Cho et al., 2008). Owing to the greater circumferential irregularities of the disc surface prompted by thermo-mechanical interactions between the pad and the disc, hot judder occurs at higher frequency than cold judder, usually around 200Hz (Day, 2014).

More so, since weight reduction has become a major topic in the automotive industry due to the environmental impacts of carbon emissions, it is imperative to ensure that the thermal performance of a disc brake is optimized in a weight efficient method. With the advent of electric vehicles, there are even more reasons to reduce total vehicle mass in order to extend the range of the vehicle. Furthermore, the brake disc constitutes part of the vehicle's unsprung mass, so minimizing this mass helps to improve ride comfort and reduce damage to the road surface.

Undoubtedly, numerical simulations like finite element analysis (FEA) are now commonly used in the industry to predict the thermal behaviour of disc brakes. This has helped to reduce the number of experimental testing in developing this component, leading to decrease in development time and cost. However, apart from using this technology as merely a design verification tool, it can also be integrated with optimisation technology to even further reduce development time, weight and cost significantly. Unfortunately, in disc brake research and development, the full potential of this know-how has not been leveraged for enhancing thermal performance and reducing weight.

Generally, structural optimisation in FEA can be classified into two main categories, parametric and non-parametric. The choice of optimization technique usually depends on problem type (e.g. linear or non-linear FEA). Parametric optimization involves "parameterization" of already prescribed entities or parameters in a model to create design variables. Examples of non-parametric optimization techniques are topology, topography, free-shape and free-size optimization.

From a thermal perspective, few researchers have adopted the parametric approach for optimising disc brakes. However, Le Gigan (2017) improved the design of a brake disc for heavy vehicle application through parametric evaluation. The reference case was a standard commercial brake disc made of grey cast iron comprising of straight vanes. A design of experiments method was introduced to study different geometry variations with regards to various performance indices such as temperatures, displacements and plastic strains. Response surfaces were then constructed based on the parametric studies and optimization was set up to minimise the mass of the disc whilst making sure that the fatigue is below that of the reference disc. In this same research, an additional optimization strategy was built to minimise the fatigue load while setting the mass of the reference disc as constraint. The results showed that it was possible to achieve a mass saving of up to 13% and up to 50% improvement in durability by the integration of different pillar arrangements rather than straight vanes.

A shape optimization of a ventilated disc was carried out by Jung et al. (2012). In this study, a FE model was first developed to simulate the thermo-mechanical behaviour of the disc using a mathematically derived heat flux as an input. An optimization strategy was then implemented to minimise temperature rise and thermal deformation by determining the optimum shape of the disc cross-section. This is also a response surface method based on design of experiments. Likewise, a kriging surrogate model was developed by Song et al. (2009) in order to evaluate the best geometry for a circumferential friction disc brake considering thermo-elastic instability. Although the conceptual new design produced resulted in a slight increase in weight, a significant improvement in thermal performance was achieved.

Another good application of structural optimization in brake disc development was presented by Wagner et al. (2014). This work was aimed primarily at preventing squeal, which is not within the scope of this research. However the very structured approach involving the use of Sequential Quadratic Programming (SQP) and Genetic Algorithm (GA) to separate eigen-frequencies within a given frequency range can be tailored to solve other kinds of parametric optimization problems relating to disc brakes.

Alnaqi et al. (2014) also produced a lightweight brake rotor. This also involved the use of General Algorithms in meeting both the cost and performance requirements of an automobile braking system. Before the optimization was deployed, a sensitivity analysis was first set up using Taguchi techniques to understand the influence of various design parameters on the thermal behaviour of the brake rotor. The analysis focused on the following parameters relating to both the disc and coating: thickness, density, thermal conductivity and specific heat capacity). It was observed that the performance of the disc could be improved by coating the surface to enhance the dissipation of heat by convection.

Furthermore, a FE model of a disc-type magnetorheological brake was developed by Assadsangabi et al. (2011) which also integrated the use of Genetic Algorithms to derive optimal design parameters for automotive applications. The goal was to improve torque capacity of the brake whilst maintaining mass. This is a unique strategy owing to the inclusion of a cost function which helped to minimise the cost of the final design solution. More so, Tirovic et al. (2004) carried out a design optimization process for wheel hub mounted railway disc brakes considering both bulk and macro thermal effects. The objective was to reduce axial deformation of the disc and to minimise stress level in the hub. This method was successfully implemented in proposing new disc geometry which did not only satisfy the aforementioned targets, but also offered considerable reduction in cost and development risks. An optimization technique for a brake disc comprising of multiple rows and vented pins was studied by Palmer et al. (2009). Although this was based on a CFD analysis, the optimization strategy provided good insight into how the rate of cooling can be maximised in a brake disc through the evaluation of best vane geometry.

A sensitivity analysis based on hot judder simulation of a ventilated disc was also carried out by Park et al. (2014). Having employed the use of commercial FE program SAMCEF to develop a robust thermo-mechanically coupled system to predict hot spots on the disc surface, they proceeded to develop a Design of Experiments (DoE) approach using a two-level factor, Plackett-Burman design. Thereafter, a response equation of the disc thickness variation was produced. This led to the proposal of an improved disc design with reduced brake torque variation (BTV). Similarly, Grieve et al. (1998) deployed Taguchi method to carry out a parametric sensitivity analysis of a medium-sized passenger disc brake. The design challenge was to propose a light weight design using aluminium metal matrix composite. Initially, a FE model of an existing concept was used in calculating maximum temperatures for critical brake tests. Thereafter, the parametric study was set up to understand the influence of cheek thickness, vent width, thermal conductivity and rate of cooling. This then enabled optimum values of these parameters to be evaluated in order to minimise rotor temperatures.

In order to reduce the occurrence of hot judder, a disc brake should be designed with rapid cooling performance and minimised thermal deformation in the event of braking. Also, the durability of the disc which is affected by the generation of thermal cracks produced from the growth of locally concentrated heat zones on the disc surface (called hot spots) must be maximised. This paper describes how structural optimization techniques have been implemented in the redesign of a ventilated brake disc used on a high performance passenger vehicle with the view of improving thermal performance, while minimizing mass.

2. Thermal analysis

2.1 The two-piece ventilated brake disc

The disc under consideration is shown in Figure 1. This is a ventilated disc brake used in a high performance luxury sport utility vehicle and known to exhibit hot judder. It is a two-piece disc design comprising of a friction ring (with vents) connected by 17 pins to a top hat structure. The intention of this concept is to allow the friction ring to freely expand circumferentially such that thermal coning of the disc is minimised.

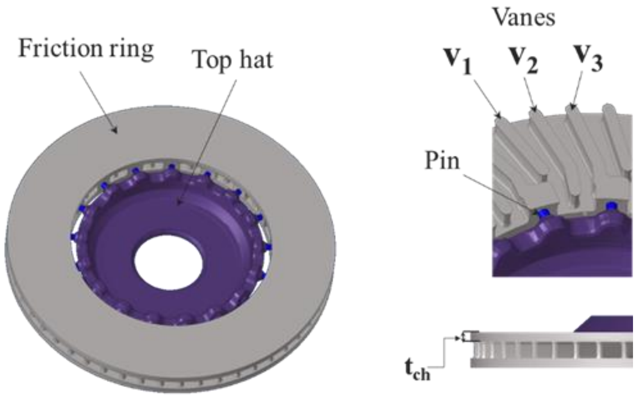


Figure 1: The ventilated brake disc

The disc's diameter is 380 mm and its total mass is 11.45 kg. It has three unique vanes with thickness: t_{v1}, t_{v2} & t_{v3} = 5.5 mm and cheek thickness, t_{ch} = 8 mm. More so, the disc is made of grey cast iron, the top hat is aluminium alloy, while the pins and pads are made from steel and proprietary friction material respectively. The assumed mechanical and thermal properties of each component are shown in Table 1.

Table 1: Material properties of the disc (Tang et. al, 2016)

Material properties	**Disc**	**Top Hat**	**Pins**	**Pads**
	Grey cast iron	Aluminium alloy	Steel	Friction material
Thermal conductivity, *K* (W/m K)	48	113	17	0.5
Density, ρ (kg/m^3)	7200	2680	7800	1250
Elastic modulus, *E* (GPa)	100	71	210	0.7
Poisson's ratio, υ	0.25	0.33	0.4	0.25
Co-efficient of thermal expansion, α (10^{-6}/K)	10	21	11	11
Specific heat capacity, *c* (J/kg K)	480	880	500	1000

2.2. Assumptions

In order to analyse the disc, the following assumptions have been made with the aim of simplifying the problem: (i) no consideration for change in material properties with temperature or the effects of thermo-plasticity; (ii) no wear effects; (iii) pressure is uniformly distributed throughout the friction surfaces; (iv) co-efficient of heat transfer is constant during braking process (Grzes, 2009); (v) the pin-disc contact interaction is not altered during service.

More so, the initial temperature of the disc was set at 60°C and the convective heat transfer co-efficient on the vent surfaces and other free surfaces were set to be 100 and 70 W/m^{2}K respectively (Tang et al., 2016). However heat loss due to radiation was assumed to be small and ignored.

2.3 Finite element modelling

The FE model for the analysis did not include the pads and the back plates. The components were meshed with Altair Hypermesh software using temperature displacement elements; ABAQUS eight-node trilinear element type C3D8T was used to model the disc surface and four-node linear elements C3D4T were used for the vents, top-hat and pins.

The disc-pin connections were modelled such that the disc is allowed to move radially, thus permitting it to freely expand circumferentially. The co-efficient of friction between the disc and pad was assumed to be 0.4 and a large gap conductance, 1 MW/m^2/°K, was also defined at this interface. In addition, a standard ABAQUS surface behaviour known as "SMALL SLIDING" was defined. With this formulation, the contacting surfaces of the pins and disc are allowed to go through only small scale sliding relative to each other; however rotation of the pad contact surface is allowed (Abaqus, 2014).

2.4 Thermal loading and boundary conditions

A fully coupled temperature-displacement transient analysis was deployed in simulating the thermo-mechanical behaviour of the two-piece ventilated disc using ABAQUS.

Table 2: Braking parameters for the disc (Tang et. al, 2016)

Braking parameters	
Brake disc speed (rev/min)	976
Vehicle speed (km/h)	150
Pressure (bar)	25.5
Braking torque (Nm)	499.3
Brake duration (s)	24
Brake pad inner radius (mm)	118
Brake pad outer radius (mm)	184

The disc is subjected to drag braking for which the assumed parameters are shown in Table 2. The maximum heat flux is calculated by dividing the braking power (product of braking torque and angular velocity) by the area swept by the pad. The actual heat flux applied to the disc in the analysis is evaluated by multiplying the maximum value by the heat partition factor γ of the disc. This is calculated as shown below:

Braking power, P = Braking torque (T) x disc rotational velocity (ω) = 499.3 x 102.2 = 50928.6 W

Surface area, A swept by pad = $\pi\,(0.1842^2\text{-}0.1182^2) = 0.03216\ \text{m}^2$

Heat flux = Braking power/Area = 820 kW/m^2

Also, it is assumed that $\gamma = (1+\sqrt{\{[k_p\rho_p c_p]/[k_d\rho_d c_d]\}})^{-1}$

Here k, ρ and c are thermal conductivity, density and specific heat capacity respectively. The subscripts p and d denote pad and disc respectively. Referring to Table 1, γ is estimated to be 0.94. Therefore, the heat flux transmitted to the disc surface is 770 kW/m^2.

The heat input is smeared out over the entire rubbing surface of the disc. All other surfaces are assumed to be subject to convective heat transfer. Heat flux is applied for 24 s and the disc is then allowed to cool down by convection for the rest of the analysis.

With regards to the boundary condition for the structural analysis, the outer face of the top hat structure is fixed in all translation directions only.

2.5 Analysis results

The maximum temperature in the disc at the end of the brake application (24 s) is 584°C (see Figure 3). It can be seen in Figure 4a that hot spots have formed on the disc surface and the number of hot spots corresponds to the number of pins in the disc, which is typical for this type of ventilated disc. The thermal deformation of the disc is high between the vanes (Figure 4b). Maximum deformation is 0.134mm and DTV due to thermal deformations is 0.0012mm (or 1.2μm). Figure 4c shows that maximum stress in the disc (103 MPa) is less than yield stress of the cast iron (138 MPa).

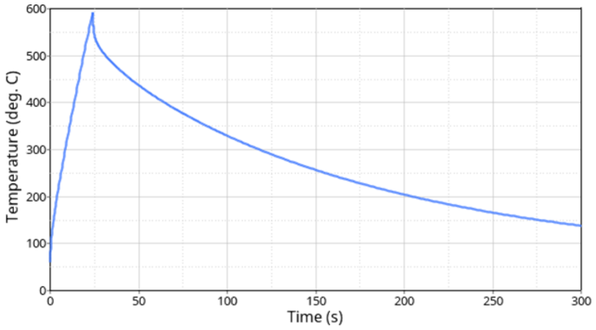
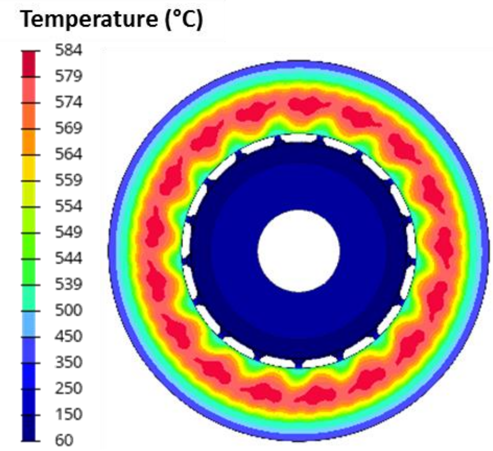
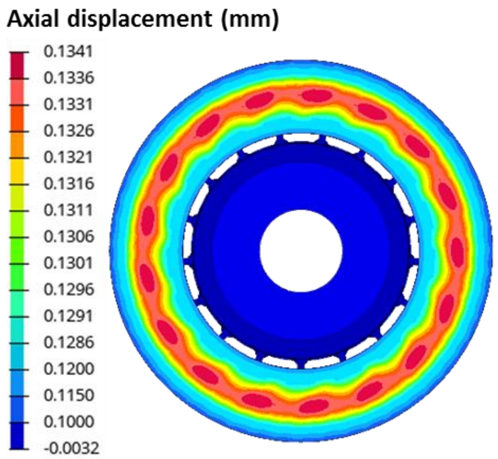


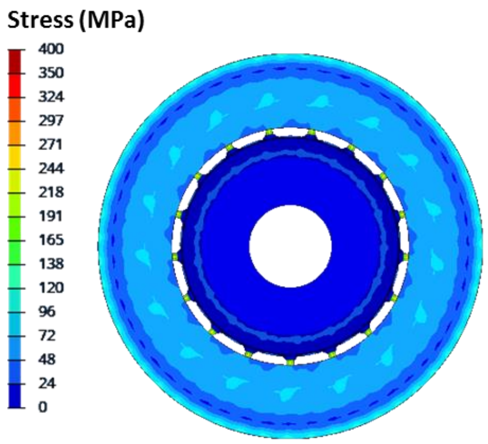
Figure 3: Maximum temperature response of disc outboard surface



(a) Temperature distribution on the outboard surface



(b) Axial displacement on the outboard surface



(c) Von Mises stress on the outboard surface

Figure 4: Finite element results at the end of the brake application for the baseline design

3. Parametric optimization

In the parametric optimization, a Design of Experiments (DoE) approach will be utilised, followed by building a mathematical approximation of the structured run matrix relating the variables to the outputs (often referred to as a response surface). A response surface methodology is then used to evaluate the optimum combination of the chosen design variables. The overall objective was to investigate how the disc can be modified in such a way that temperature distribution is more uniform and DTV is decreased. The design challenge is how to do this in a weight efficient method.

The chosen design variables are the cheek thickness (t_{ch}) and vane thicknesses (t_{v1}, t_{v2} and t_{v3}) of the three different types of vane, making a "shape" optimization. The cheek thickness is varied such that the overall thickness of the disc remains constant.

Also, the performance indices to be evaluated in this study are maximum disc temperature (T_{max}), maximum axial displacement (Dy_{max}), circumferential difference in disc temperature (ΔT), disc thickness variation (DTV), von-Mises stress (σ) and disc mass (M_d).

Referring to Figure 1, it is apparent that any change to the chosen design variables will only affect the mass of the friction ring (no effect on top hat and pins); therefore in this study, M_d refers only to the mass of the friction ring.

The three stages of the parametric optimization were carried out using Altair Hyperstudy software.

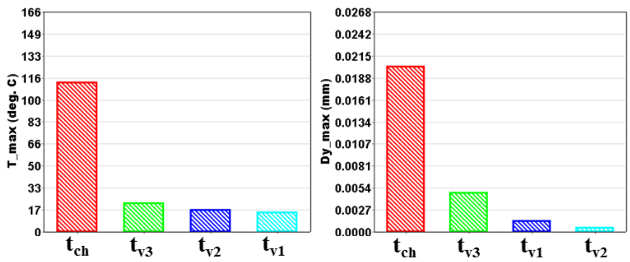
3.1. Design of Experiments

In this research, the Latin Hypercube DoE method is utilized to understand the global behaviour of the chosen disc geometrical parameters on the thermo-mechanical performance. This method has been selected is order to minimize the numbers of runs required to build the matrix by only choosing a fraction of the combinations needed for a full factorial DoE. Generally, choosing a different DoE type other than full factorial comes at the expense of not being able to completely resolve all the main effects and interactions (Altair, 2016), but the benefit of reduced computational costs is highly important and makes Latin Hypercube a better option.

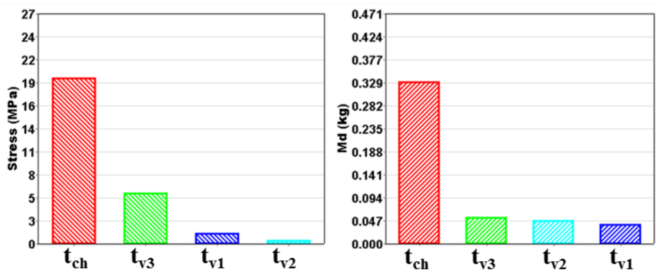
Table 3: Initial, lower and upper bounds selected for DoE study

Design variables		Initial values	Lower bound	Upper bound
Cheek thickness	t_{ch}	8.0	6.0	10.0
Vane thickness	t_{v1}	5.5	3.5	7.5
	t_{v2}	5.5	3.5	7.5
	t_{v3}	5.5	3.5	7.5

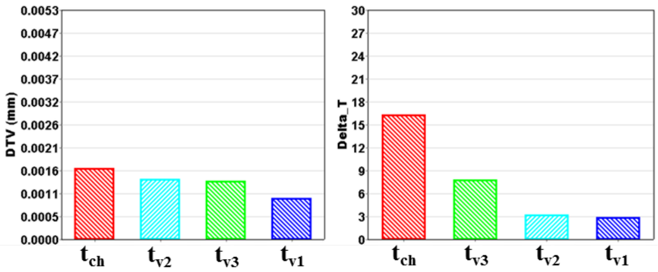
As shown in Table 3, all the geometric parameters were varied by ±2mm. Figure 5a-5c show Pareto plots which rank the effects of the disc geometrical parameters on its thermo-mechanical performance in hierarchical order. Clearly, it can be observed that the cheek thickness has the most influence. Increasing t_{ch} alone will decrease temperature, but will also significantly increase mass. However, when it comes to DTV, the effects of vane thicknesses are more visible.



(a) Maximum temperature (left) and maximum axial displacement (right)



(b) Von Mises stress (left) and disc mass (right)



(c) Disc thickness variation (left) and circumferential difference in disc temperature (right)

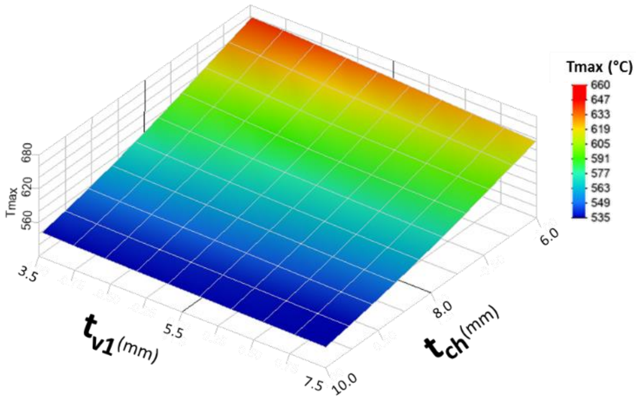
Figure 5: Pareto plots showing effect of design variables on analysis results

3.2 Response surface approximation

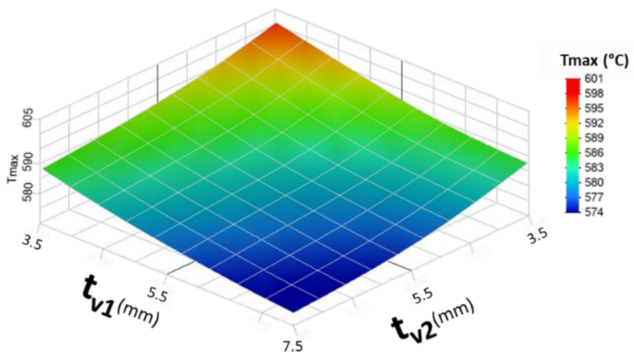
In the response surface approximation, surrogate or meta-models which relate the design variables to the output responses are produced. Here, the relationships between the design variables, (t_{ch}, t_{v1}, t_{v2} & t_{v3}), and the thermal performance indices are determined. The method utilized in this study is the "Kriging "approach which has been selected so that the approximations interpolate between the sampling points as smoothly as possible (Altair, 2016), thereby reproducing simulation results with minimal errors and producing a reasonable fit. In disc brake development, this has been tested by researchers such as (Song and Lee, 2009).

Figure 6a shows the relationship between cheek thickness, t_{ch} and one of the vane thicknesses, t_{v1} with respect to the disc maximum temperature, T_{max}. It can be seen that T_{max} is highly sensitive to change in t_{ch}. The two entities are directly and linearly proportional. Similar behaviour is observed for t_{ch} versus the other vane thicknesses, t_{v2} and t_{v3}. In Figure 6b, it can also be seen that t_{v1} and t_{v2} are proportional to T_{max}, but the potential for decreasing temperature is not as much. In addition, plotting the vane thicknesses against one another show that they are almost equally sensitive to T_{max}, but not as much as t_{ch}.

On the other hand, when it comes to DTV, the influences of the vane thicknesses become apparent (Figure 7a and 7b). It is also clear that, the relationship between t_{v1} and t_{v2} (or t_{v3}) against DTV is non-linear; hence the response surfaces are a little complex (Figure 7c-7e).

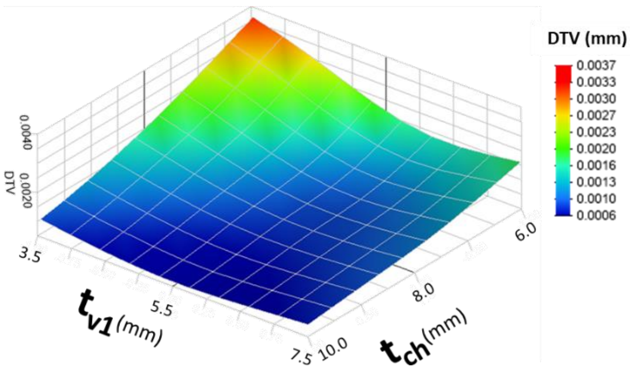


(a) Effect of vane thickness t_{v1} and cheek thickness t_{ch}.

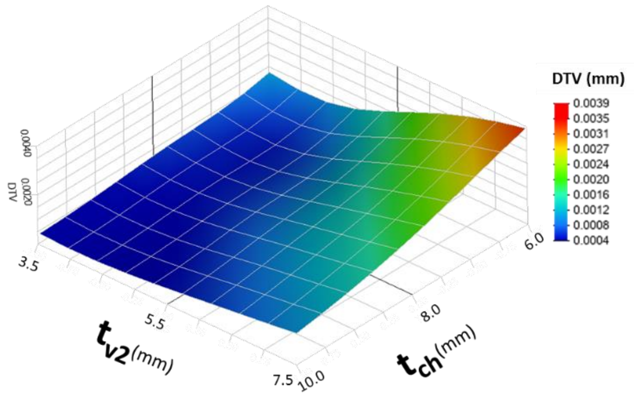


(b) Effect of vane thicknesses t_{v1} and t_{v2}.

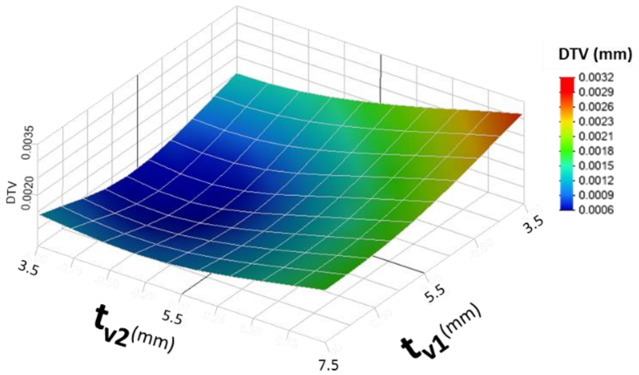
Figure 6: Response surfaces for maximum disc temperature, T_{max}



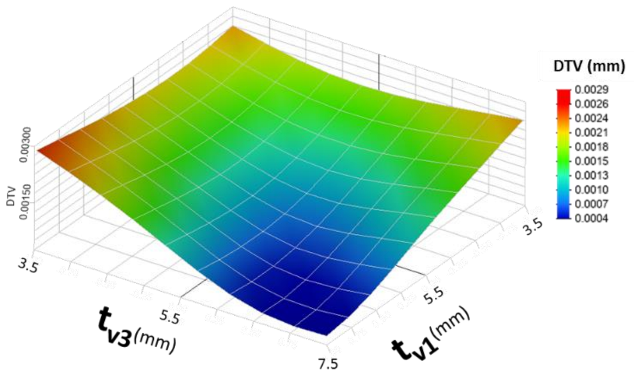
(a) Effect of vane thickness t_{v1} and cheek thickness t_{ch}.



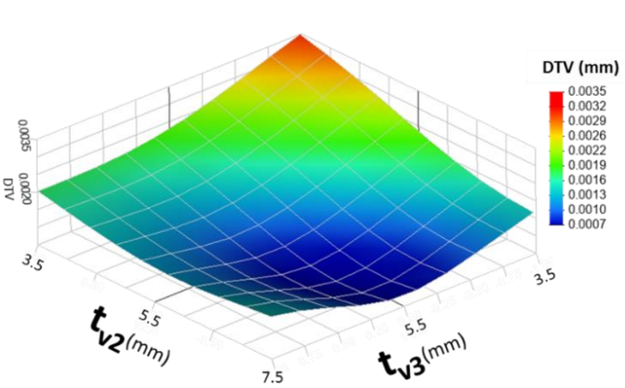
(b) Effect of vane thickness t_{v2} and cheek thickness t_{ch}.



(c) Effect of vane thicknesses t_{v1} and t_{v2}.



(d) Effect of vane thicknesses t_{v1} and t_{v3}.



(e) Effect of with vane thicknesses t_{v2} and t_{v3}.

Figure 7: Response surfaces for DTV

3.3 Optimization

In order to minimise the propensity of hot judder in a disc brake, DTV should be minimised by allowing braking energy to be transmitted into the disc in an even manner. Therefore, a target of 50% of the baseline DTV value of 1.2µm was set in the optimization. It was also important to ensure that any proposed change to the disc did not lead to increased stress levels.

The optimization technique chosen in the study is the "Adaptive Surface Response Method (ASRM)". This method is very efficient at evaluating global optima (Altair, 2016). Two optimization set-ups were defined: one where T_{max} is constrained and another where it is not constrained.

3.3.1 Optimization A (with temperature constraint)

The aim was to propose a new design with more uniform thermal deformation around the circumference of the disc. Also, a reduced operating temperature is desired in order to minimise the likelihood of hot spotting and thermal instability. Another constraint implemented is the von-Mises stress which was set to be below the yield stress of the cast iron friction ring, 138 MPa. Thus, the optimization problem is defined as follow:

Evaluate t_{ch}, t_{v1}, t_{v2}, t_{v3}

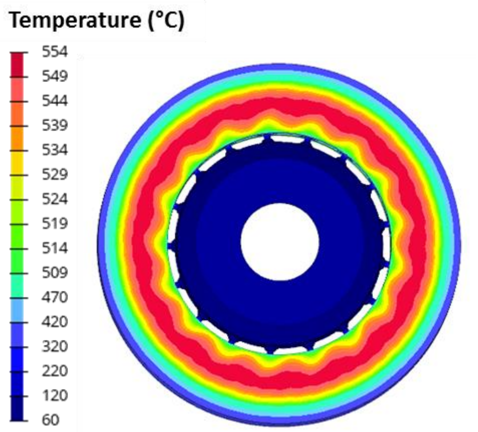
Minimize disc mass, M_d

Subject to: Maximum von-Mises stress < yield (138 MPa)

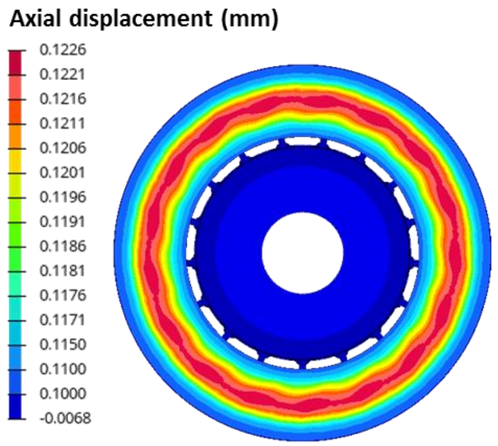
DTV ≤ 50% of baseline (1.2µm)

$T_{max} < 584$°C

The calculated optimum values for cheek and vane thicknesses are summarized in Table 4. It can be seen that t_{ch} has increased to 9.4mm. The optimization then tried to offset this added mass by reducing t_{v1} and t_{v2}, but t_{v3} remained unchanged. Notably, the thermal performance of the disc has improved. Apart from the decrease in maximum axial displacement to 0.123 mm (-9%), the DTV of the disc has also reduced to 0.56µm (-53%). Moreover a decrease in T_{max} to 554°C (-30°C) and a more uniform temperature distribution have been achieved (see Figure 8a and 8b). The decrease in temperature is as a result of the redistribution of mass in the friction ring, thereby minimising the increase in total disc mass (+ 0.85 kg).



(a) Temperature distribution on the outboard surface



(b) Axial displacement on the outboard surface

Figure 8: Finite element results at the end of the brake application for Optimization A (with temperature constraint)

3.3.2 Optimization B (without temperature constraint)

In this optimization, T_{max} is not constrained. This was in order to investigate the possibility of generating a lighter disc with decreased non-uniformity in deformation and temperature, even with a slightly higher T_{max} under the same thermal loading conditions. This new optimization problem can be stated as follow:

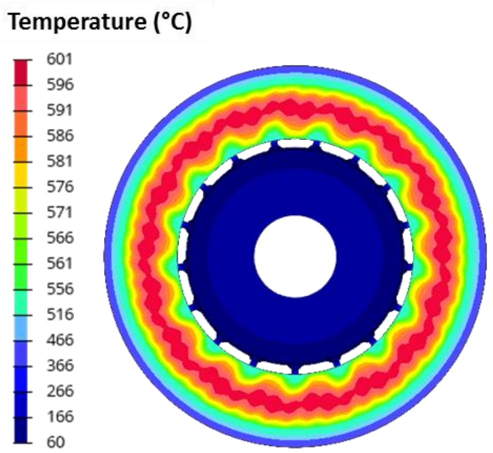
Evaluate t_{ch}, t_{v1}, t_{v2}, t_{v3}

Minimize disc mass, M_d

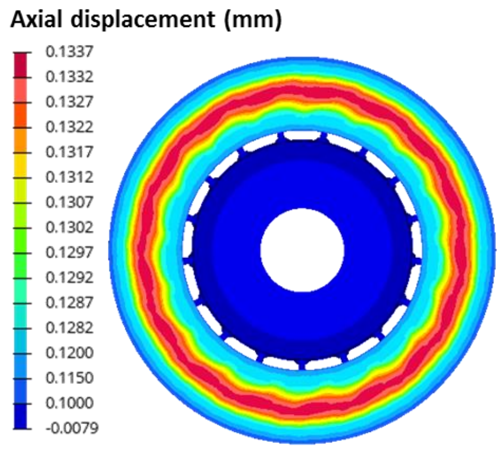
Subject to: Maximum von-Mises stress < yield (138MPa)

DTV $\leq$ 50% of baseline (1.2µm)

The new results are also shown in Table 4, t_{ch} and t_{v2} are reduced to 7.5mm and 4.5mm respectively, while t_{v1} and t_{v3} are unchanged, yielding a reduced disc mass of 10.98 kg (-0.47 kg). Owing to these changes, the temperature in the disc has now increased to 601°C (+17°C), but without increase in axial displacement and in fact, a reduced DTV of 0.68µm (-43%), a little more than 50% of the baseline value of 1.2µm. Temperature distribution and axial displacement for the new disc are shown in Figure 9a and 9b.



(a) Temperature distribution on the outboard surface



(b) Axial displacement on the outboard surface

Figure 9: Finite element results at the end of the brake application for Optimization B (without temperature constraint)

Table 4. Optimum geometric parameters from both optimizations

Design variables		**Baseline values**	**Optimization**	
			A	**B**
Cheek thickness (mm)	$\mathbf{t_{ch}}$	8.0	9.4	7.5
Vane thickness (mm)	$\mathbf{t_{v1}}$	5.5	4.8	5.5
	$\mathbf{t_{v2}}$	5.5	4.6	4.5
	$\mathbf{t_{v3}}$	5.5	5.5	5.5
Maximum temperature (°C)	$\mathbf{T_{max}}$	584	554	601
Disc thickness variation (µm)	**DTV**	1.2	0.56	0.68
Friction ring mass (kg)	$\mathbf{M_d}$	10.32	11.17	9.85
Total disc mass (kg)	$\mathbf{M_{total}}$	11.45	12.30	10.98

4. Discussion and Conclusions

Coupled finite element analysis has demonstrated the thermo-mechanical behaviour of a pin-mounted two-piece disc design. Although the analysis is yet to be validated against measured data, the model was considered good enough for the purpose of optimization as it exhibited typical thermo-mechanical behaviour of a large ventilated disc brake.

In optimization A, a reduced temperature was achieved, along with a minimized DTV, resulting in a mass increase of 0.85kg. In contrast, a mass saving of 0.47kg was achieved in optimization B, mainly due to the reduction of t_{ch}, although this came at the expense of increased T_{max}. as a result of not constraining T_{max} in the optimization set-up. However, in optimization B, it was still possible to reduce DTV by 43% and improve the uniformity of temperature distribution on the disc surface. In both optimization studies, the von-Mises stress level did not exceed yield stress of the cast iron material.

The paper demonstrates that disc geometric parameters can be tuned to improve thermo-mechanical performance and to minimize the risk of hot judder. Optimizing a disc brake should not only focus on reducing temperature and axial displacement, but also on minimizing circumferential differences in axial displacement and maximizing uniformity in temperature distribution.

Finally, it is evident that parametric optimization can be utilized as a design tool for disc brake development. The response surface methodology can be used to evaluate various design options without the need for re-modelling and re-analysis. This will help to ensure that the component is designed in a weight efficient manner and in reduced time.

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