Topology optimisation of an automotive disc brake rotor to improve thermal performance and minimise weight.

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ABSTRACT: Weight reduction has become a major topic in the automotive industry due to the environmental impacts of carbon emissions. When considering the thermal performance of disc brake rotors, it is increasingly important to optimise their design in a weight-efficient manner. With the advent of electric vehicles, which are heavy due to battery requirements, there is even more impetus to reduce total vehicle mass which can extend driving range. Furthermore, the brake disc constitutes part of the vehicle's unsprung mass, so minimising brake rotor weight helps to improve ride comfort and reduce damage to the road surface. Considering thermal performance, the temperature rise produced by friction at the sliding interface leads to thermo-elastic deformation within the disc. Consequently, this can change the distribution of contact pressure and cause thermal localisation 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. This paper introduces topology optimisation in the development of a ventilated brake disc used on a high performance passenger vehicle with the aim of improving thermal performance, while minimising mass. A baseline ground structure is formulated to enable new conceptual vane geometries for both improved thermal performance and lower disc mass to be derived. This approach allows novel disc designs not previously considered to evolve. Although possibly more difficult to manufacture than the conventional disc, the potential performance benefits of this more radical optimisation strategy are clearly demonstrated. Furthermore, preliminary CFD analysis is utilised to predict the air flow through the conceptual vane designs produced from the topology optimisation process. This allows for estimation of convective heat transfer coefficient for the novel design concepts and enables detailed flow patterns within the vane geometries to be predicted.

KEY WORDS: Brake disc, finite element analysis, topology optimisation, thermal performance, weight, judder, CFD.

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 localisation 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. Thermal deformation can be categorised into coning, waving, non-uniform thermal expansion, phase transformation and pad material deposition on the disc due to heating [1]. Coning is triggered by axisymmetric thermal distortion of the disc leading to its axial displacement. Coning can be controlled by careful design of the disc geometry especially in the neck region that connects the friction ring to the central hub, or utilising a pin-supported disc such as the one considered in this paper. Any non-axisymmetric thermal expansion can lead to increases in disc thickness variation, DTV, and axial run-out about the circumferential rubbing path [2]. Hot judder is usually associated with this thermal localisation phenomenon (often called hot spotting) in which several hot regions are formed on the surface of a brake disc during high energy braking events [3]. Under continuous non-uniform contact, the hotspots grow and produce hot judder vibration which makes driving difficult and dangerous [4]. 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 a higher frequency than cold judder, usually around 200Hz [5].

Moreover, 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 optimised 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 minimising this mass helps to improve ride comfort and reduce damage to the road surface.

Numerical simulations such as finite element analysis (FEA) are now routinely used in the industry to predict the thermal behaviour of disc brakes. This has helped to reduce the amount of experimental testing in developing this component, leading to decrease in development time and cost. However, as well as 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. Unfortunately, in disc brake research and development, the full potential of this optimisation technology 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 optimisation technique usually depends on problem type (e.g. linear or non-linear FEA). Parametric optimisation involves "parameterization" of already prescribed entities or parameters in a model to create design variables as previously in [6]. Examples of non-parametric optimisation techniques are topology, topography, free-shape and free-size optimisation. In this paper, the application of topology optimisation in the design of disc vane geometry is explored.

A good example of the application of non-parametric optimisation techniques in brake disc development was presented in [7]. This was applied in the design of a lightweight monobloc cast iron brake disc with a fingered hub. Although the optimisation focused only on mechanical loading with no consideration of thermal effects within the optimisation problem definition, it still demonstrated how the technique can be deployed in reducing the mass of the component. However, from a thermal FEA perspective, the application of non-parametric optimisation techniques have not yet been used in the design of vane geometry for an automotive disc brake.

Some researchers have presented the results of thermally-based topology optimisation. In [8], the application of topology design methods was investigated for thermally conductive materials. The thermal conductivity of every individual element was defined as a design variable and the prescribed evolutionary structural optimisation (ESO) was formulated to remove or deplete the conductive material of the elements with least sensitivity.

Furthermore, [9] presented how topology optimisation can be executed in thermal FEA problems. The authors used simple case studies to demonstrate the capabilities of the Optistruct software in performing optimisation of structures in coupled thermal-structural analysis. This methodology employed the use of thermal compliance rather than field temperature as the objective function of the optimisation problem. The approach was proven to be considerably more efficient in solving the problem than minimising of the peak temperature of the whole structure.

Similarly, [10] demonstrated the feasibility of implementing topology optimisation in heat conduction problems. Although this approach was based on a finite volume method (FVM) rather than the conventional finite element method of analysis, it still demonstrated an efficient strategy in dealing with thermal-based optimisation. The defined problem comprised of element area and a cost function as design variables which controlled the thermal conductivity. It also included a volume constraint with a penalty function and the final problem was solved using the method of moving asymptotes (MMA).

Researchers in [11] developed a Pareto topology optimisation considering thermal loads by using thermo-elastic equations to derive the sensitivity of the objective functions (strain energy and maximum stress) in the design domain. The authors observed that when thermal loading effect was considered, the total strain energy density function became a nonhomogeneous function of the strain. Based on the chosen objective functions, both compliance and stress minimisation problems were solved with volume fraction set as constraint.

Moreover, topology optimisation was used to stiffen a thermally restrained conceptual engine exhaust-washed structure by [12]. In this example, the widely used compliance minimisation approach did not produce favourable designs. However, the authors resolved this issue by deploying fictitious mechanical load cases which mimicked the thermo-mechanical behaviour of the part. This enabled the generation of conceptual stiffness material which prevented out-of-plane deformation.

In spite of the potential of topology optimisation, to the authors' knowledge, the technique has not yet been applied in optimising the vane layouts of a disc brake from a thermo-elastic standpoint. This is because there is currently no topology optimisation method readily available and purposely developed for such applications. This paper explores how the material layout capabilities of topology optimisation can be implemented in the design of vanes in a ventilated brake disc used on a high performance passenger vehicle with the view of improving thermal performance, especially the propensity for hot judder, while minimising mass.

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 heavy braking. Also, the aim should be to improve the durability of the disc which is affected by the generation of thermal cracks produced from the growth of locally concentrated heat zones (hot spots) on the disc surface.

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 which is 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.

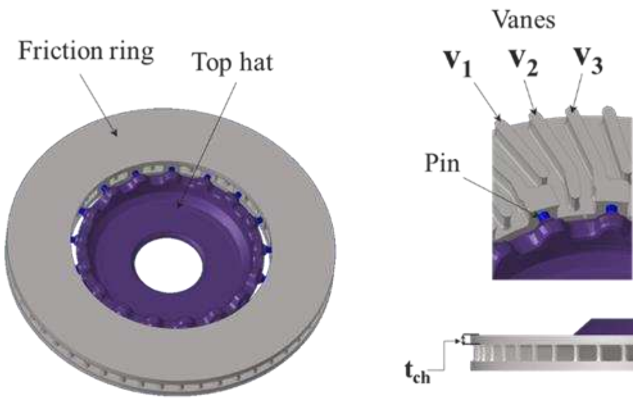


Figure 1: The ventilated brake disc

Table 1: Material properties of the disc [3]

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

The disc diameter is 380 mm and its total mass is 11.45 kg. The cheek thickness, t_{ch} = 8 mm and it has three unique vanes, namely v_1, v_2 and v_3 each with the same thickness i.e. $t_{v1} = t_{v2} = t_{v3}$ = 5.5 mm, see Figure 1. The friction ring is made of grey cast iron, the central hub 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.

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) the coefficient of heat transfer is constant during the braking process [13]; (v) the pin-disc contact interaction is not altered during service.

Furthermore, the initial temperature of the disc was set at 60°C and the convective heat transfer coefficient on the vent surfaces and other free surfaces were set to be 100 and 70 W/m^{2}K respectively [3]. 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 disc components were meshed with Altair Hypermesh software (Version 2019) using temperature displacement elements; ABAQUS eight-node trilinear elements type C3D8T were used to model the disc surface and four-node linear elements C3D4T were used for the vents, top-hat and pins.

Two scenarios were considered for the disc-pin connections in the simulations: (i.) Sliding pins - connections modelled such that the disc is allowed to move radially, thus permitting it to freely expand circumferentially. The coefficient 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 [14]. (ii.) Fixed Pins – This is the worst case scenario which could occur after several braking cycles when relative motion of the connection is minimised. This was modelled using ABAQUS "TIE" conditions. Although this formulation is too extreme as the true behaviour of the connections will be between sliding and tied.

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 (Version 14).

Table 2: Braking parameters for the disc [3]

Braking parameters	
Brake disc speed, ω (rev/min)	976
Vehicle speed, *V* (km/h)	150
Pressure, *P* (bar)	25.5
Braking torque, *T* (Nm)	499.3
Brake duration, *t* (s)	24
Brake pad inner radius, r_i (mm)	118
Brake pad outer radius, r_o (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 which is defined by $(1+\sqrt{\{[k_p\rho_p c_p]/[k_d\rho_d c_d]\}})^{-1}$, where 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. On the basis of these equations, 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 central hub is fixed in all translation directions only.

2.5 Analysis results – *Sliding pins*

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 4(a) 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.

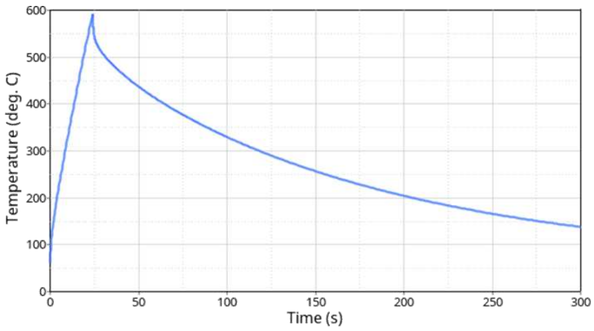
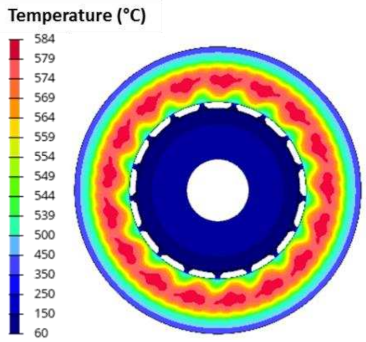
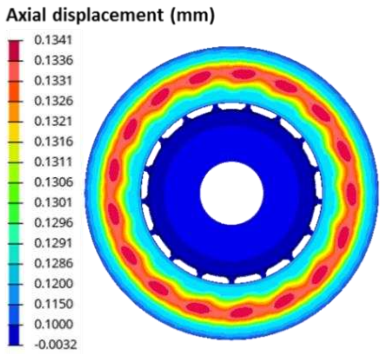


Figure 3: Maximum temperature response of disc outboard surface

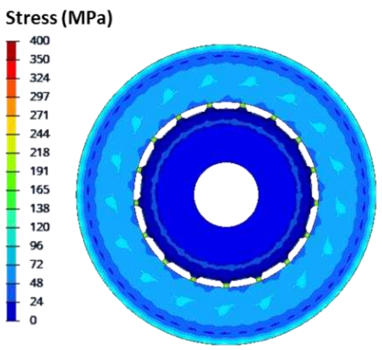
The thermal deformation of the disc is high between the vanes, Figure 4(b). Maximum deformation is 0.134 mm and DTV due to thermal deformation is 0.0012 mm (or 1.2 µm). Figure 4(c) shows that maximum stress in the disc (103 MPa) is less than yield stress of the cast iron (138 MPa).



(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

2.6 Analysis results – *Fixed pins*

As seen in Figure 5, when the pin-disc connection was assumed to be fixed, a different kind of disc shape deformation called *coning* is observed, with axial displacement being 0.601mm (or 601µm). However, the temperature remains largely unchanged at the end of braking. Another obvious difference is the high stresses seen in the disc, with maximum von Mises stress equal to 542Mpa.

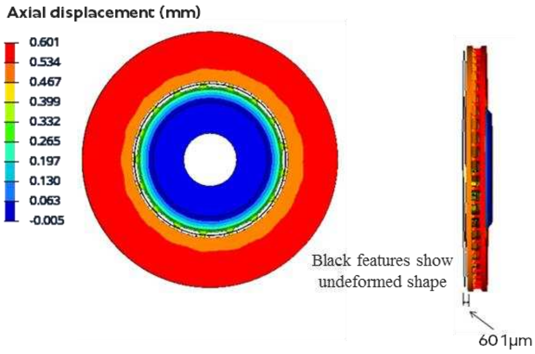


Figure 5: Axial displacement on the outboard surface (scaled by factor of 10)

2.7 Effects of change in heat transfer coefficient

In this section, the effect of the change in convective heat transfer coefficient is analysed by varying the average value on the vane surfaces by ±50%. From figure 6, it can be seen that change in HTC does not have a significant effect during the heating phase of the disc, with maximum temperature on the disc surface only increasing by 11°C when HTC is decreased by 50% and decreasing by 10°C for 50% increase in HTC. However, in the cooling phase, the effect of change in HTC value becomes more apparent. For instance, at time, t =250 s (i.e. 226 s after the end of braking), the disc would have cooled down to 110°C with increased HTC, compared to 140°C when the HTC was unchanged. This will be particularly important for intermittent braking applications such as in fast autobahn driving.

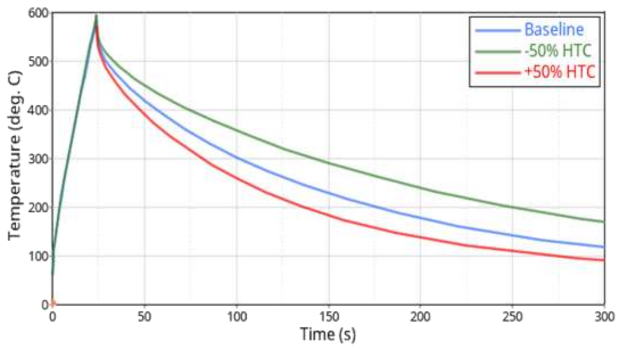


Figure 6: Effects of change in vent heat transfer co-efficient on maximum temperature response of disc outboard surface.

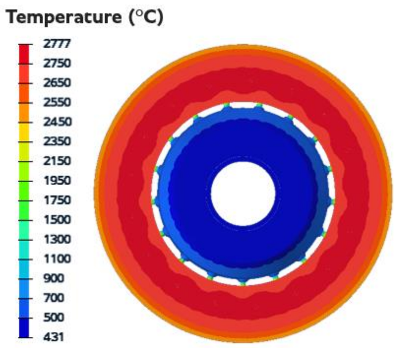
3. Topology optimisation

Topology optimisation generates an optimum shape and material distribution for a structure with a given package volume by discretising the domain into a finite element mesh and then calculating optimum material properties for each element [15]. This is a very efficient method to evaluate the optimum load-path within a structure. An extensive background on this methodology is given by Sigmund and Maute [16].

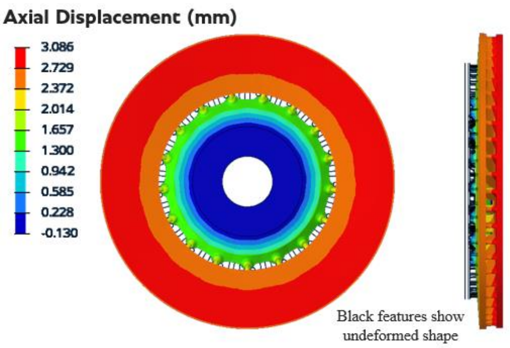
A local approximation method deployed within Optistruct was used to conduct the topology optimisation of the disc by first solving the finite element problem, then carrying out convergence analysis. This is followed by the screening of responses (constraints and objectives) and performing of design sensitivity analysis. The information derived from the aforementioned stages are then used to formulate an approximation problem which is then optimised. This process continues until the objective of the optimisation is met within the set constraints. A comprehensive description of this technique can be found in [15].

3.1 Steady state heat transfer analysis

With topology optimisation being the main focus of this work, a steady state heat transfer analysis is first conducted. In this analysis, it is assumed that there is no change in temperature with respect to time. More so, the response depends largely on the conductivity and geometry of the body being considered. The heat input and boundary conditions are the same as those used for previous transient analyses in section 2. The analysis is still coupled, but sequential, i.e. thermal analysis is first performed to determine the temperature field of the structure, then the temperature field is used as temperature load for subsequent structural analysis [14]. Also, the pin-disc connections are assumed fixed. The results are shown in Figure 7.



(a) Temperature distribution on the outboard surface



(b) Axial displacement on the outboard surface

Figure 7: Finite element results (Steady state assumption) for the baseline design

Obviously, the temperature (2777°C) and axial displacement (3.09 mm) for the steady state assumption are significantly higher than for the transient analysis shown in Figure 5. This is due to the fact that the braking is assumed to have taken place over a very long period. In this sense, the analysis is not realistic but the temperature distribution is suitable to allow the topology optimisation to proceed. Ultimately, the final optimised design must be assessed for its thermo-mechanical performance under more realistic braking conditions.

3.2 Optimisation set up

In this paper, the objective of the optimisation is set to minimise thermal compliance, ϕ_{th}, with the view of obtaining vane geometry with maximum conduction. Thermal compliance is analogous to structural compliance (or strain energy) and it is the product of surface heat flux and temperature distribution in the body. Provided the heat flux is constant, when thermal compliance is minimised, the maximum temperature of the considered structure is also minimised [17]. Furthermore, in order to limit the thermal expansion, the resulting axial deformation of the disc was defined as a constraint. Generally, a structural component that is susceptible to thermal loadings should be designed in such a manner that thermal deformation is minimised in order to prevent the detrimental effects of thermal stresses [12]. For an automotive disc brake, it is also essential to keep the temperature below the maximum operating temperature (MOT) of the material. Another constraint implemented in the optimisation set up is the mass of the disc

The optimisation is based on steady-state assumptions (section 3.1). The mathematical formulations for thermal topology optimisation are well described in literature, such as [8], [9] and [12].

Moreover, within the optimisation set up itself, no HTC was defined for the design areas, but was considered for the rest of the disc. This is because HTC could not be redefined on the free vane faces during the iterations of the optimisation. In order to overcome this issue, it was assumed that heat is lost from the vanes by conduction only. Thus, a new baseline (with maximum temperature of 7368°C and maximum axial displacement of 8.7mm) is derived. Again, these values are excessively exaggerated due to steady state assumptions, but will only be utilised for the purpose of optimisation.

Although not the focus of this research, mechanical load cases were also introduced to ensure the disc possess sufficient stiffness to resist lateral and torsional loadings during brake application. So, lateral and torsional stiffness targets were also defined.

The existing vanes were deleted and replaced with a solid region known as the design variable. The cheek thickness and the total disc thickness are unchanged and together with the rest of the disc are defined as non-design regions. In the design region, no HTC is defined in the optimisation, but defined for the rest of the disc. It is assumed that heat is lost in the vanes by conduction only.

Since there are 17 pins connecting the hat to the disc, the package volume is divided into 17 equal segments such that there were the same number of package volumes. All of them are then linked together so as to produce similar topological layouts. This was implemented using Optistruct's pattern repetition method. In order to do this, one of the segments was defined as the Master and the rest as Slaves. The Master and Slaves are linked to one another through local coordinate systems [15]. Two shape concepts were considered: straight and curved vanes. So, two package volumes were created for each concept (see Figures 8a and b).

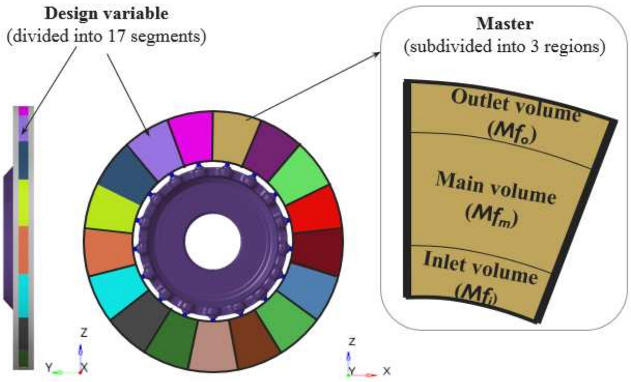


Figure 8a: Design variable (package volumes) for straight vanes

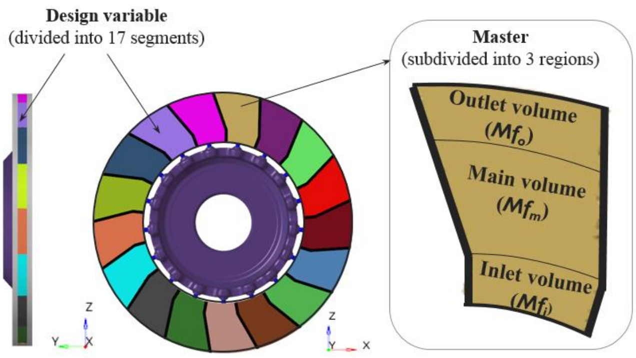


Figure 8b: Design variable (package volumes) for curved vanes

Furthermore, for both concepts, the Master is subdivided into three regions: inlet, main and outlet volume. Mass fractions were then assigned for each region: Mf_i, Mf_m and Mf_o, where subscripts i, m and o represent inlet, main and outlet respectively. The inlet and outlet mass fractions were constrained such that the optimisation only utilises a small proportion of the materials to fulfil its objective. This is mainly to ensure free movement of airflow is not prevented in and out of the disc.

The optimisation problem is set up as follows:

Minimise ϕ_{ch}

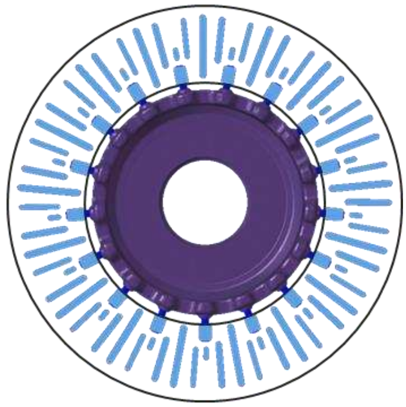
Subject to: $U_y \leq 8.7$mm, $M_{total} \leq 11.45$kg, $Mf_i \leq 0.3$, $Mf_o \leq 0.3$

Finally, the manufacturing constraint, DRAW (Split), was imposed on the Master segment to ensure materials were removed in: (i) a straight and (ii) a curved manner. Hence, two different vane concepts were investigated.

3.3 Optimisation Results

The results of the topology optimisation can be seen in Figure 9 and 10. For the straight vanes concept, it can be seen that there are four main load-paths produced. The conceptual vanes are nicely spread out which should aid airflow through the vents. The design which weighs 11.05 kg (0.4 kg less than the original design) was easily translated into a CAD solution without having to make too many changes to the original topology result (see Figure 9). However, for the curved vanes concept, the optimisation load paths in Figure 10(a) had to be modified significantly (with some having to be totally removed) in order to produce a ventilation-friendly vane design. The final modified curved vane design, shown in Figure 10(b), weighs 11.02kg, about the same as the straight vane design.

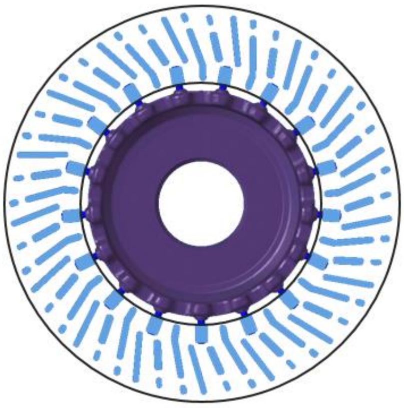
(a) Topology optimisation result for straight vanes



(b) Interpretation of topology optimisation result for straight vanes

Figure 9: Topology optimisation result and after interpretation for straight vane concept

(a) Topology optimisation result for curved vanes



(b) Interpretation of topology optimisation result for curved vanes

Figure 10: Topology optimisation result and after interpretation for curved vane concept

4. CFD Analysis

A simplified two-dimensional, Computational Fluid Dynamics (CFD) model has been developed with the ultimate goal of comparing the flow patterns of the proposed conceptual disc vane geometries and evaluating cooling performance from a fluid dynamics perspective. In this section, preliminary results from the baseline disc design are presented. Ultimately, more realistic HTC values can be derived from this modelling process. The simulation approach has been developed using the commercial CFD package, ANSYS Fluent (Version 2020R2).

4.1 Boundary conditions

Figure 11 depicts the boundary conditions prescribed in the CFD model. Due to periodic rotational symmetry and the expected repetition in flow patterns (per segment) only one segment of the disc is considered. The vanes are prescribed wall type boundary conditions with the no-slip condition and an average heat flux of 380 kW/m^2 applied. The flow is driven radially by a pressure inlet and a pressure outlet, both at atmospheric pressure and temperature. The pressure inlet is positioned at an imaginary boundary, towards the hub centre, whereas the outlet is positioned at a radial distance matching the approximate location of the inner face of the wheel. This strategy allows air to enter the vane region and flow towards the outlet boundary, depending on the speed of rotation. What drives the flow through the geometry is the centrifugal pumping effect which is achieved by separating the fluid region into three distinct regions, as shown in Figure 11. The inlet and outlet fluids are at rest but the intermediate vent fluid region is prescribed a rotational velocity to match the wheel rotational speed. As there is no pressure difference applied across the inlet and outlet boundaries, flow through the fluid region is dependent on wheel rotation only, as would be the case in reality. The rotational speed for the simulation was set to 976 rpm.

Since only one of the 17 repetitive segments (21.18°) of the disc was modelled, periodic rotational symmetry conditions were applied to the two sides of the fluid region. ANSYS Fluent employs the rotating reference frame technique in order to model a body in rotation. This method is based on a modified version of the momentum and conservation equations [18]. A summary of the boundary conditions can be seen in Table 3.

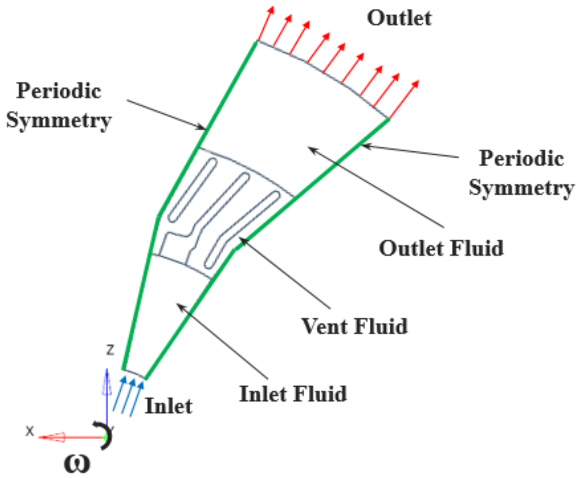


Figure 11: CFD boundary conditions (baseline design).

An average heat flux, extracted from the FEA, was applied onto the vane walls and the aim of the analysis was to derive the resulting HTC and the volumetric flow rate through the domain. Although this model is a simplification of a true braking scenario, it is sufficiently representative for the purpose of comparison [19].

Table 3: CFD boundary conditions

	Boundary condition	Parameters
Inlet	Pressure inlet	Atmospheric pressure and temperature
Outlet	Pressure outlet	
Vane surfaces	Wall	Avg. heat flux – 380kW/m^2

The standard *k-ε* turbulence model with enhanced wall-treatment was employed in conjunction with second order discretisation for all flow equations. A mesh convergence analysis was carried out on the model to ensure that the results are independent of the final mesh density, which comprises of just over 400,000 elements.

4.2 CFD results

The radial velocity results from the CFD simulation of the baseline disc are shown in Figure 12. The average HTC and mass flow rate are 83.6W/m^{2}K and 0.26kg/s (per segment) respectively. The flow pattern shows the differences in radial velocity which are particularly obvious in the vane region. This preliminary result highlights the complexity of airflow behaviour in what is a relatively simple geometric shape.

In the next stage of the work, the baseline design as well as the optimisation concepts will be re-analysed using the newly derived HTCs from the CFD simulations. Fully coupled temperature-displacement transient analyses will be deployed to compare the relative performance of the topology-optimised designs. The temperature response and axial displacements will be compared with those of the conventional design to evaluate improvements, especially in relation to hot judder performance.

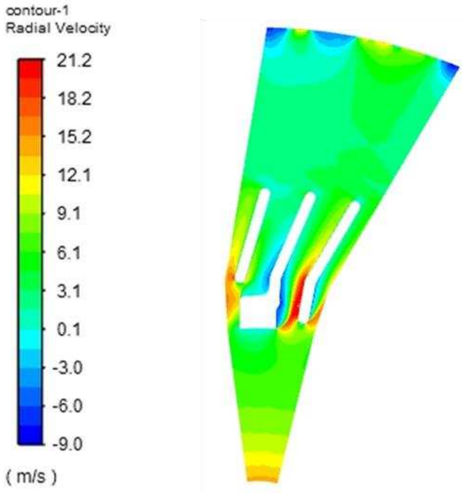


Figure 12: Radial velocity (baseline design).

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 optimisation as it exhibited typical thermo-mechanical behaviour of a large ventilated disc brake. In reality, the behaviour of the pin-disc connection will lie somewhere between free-sliding and fixed, hence the actual deformation seen in the disc could be a combination of circumferential variations in axial displacement (DTV) and coning.

The FEA-led topology optimisation demonstrates that it is possible to generate alternative conceptual vane shapes from scratch. Although the mass savings realised are not huge (up to 0.86kg per vehicle), this methodology could help to decrease development time and lead to more novel and refined disc designs.

References

[1] Jacobsson, H., 2003. Aspects of disc brake judder. Proceedings of the Institution of Mechanical Engineers, Part D: Journal of Automobile Engineering, 217(6), pp.419-430.

[2] Fieldhouse, J.D., 2013. Brake noise and vibration, Braking of Road Vehicles. University of Bradford, 2013.

[3] Tang J, Bryant D and Qi H (2016) A 3D Finite Element Simulation of Ventilated Brake Disc Hot Spotting. In: Proceedings of the EuroBrake 2016 Conference. 13-15 June 2016, Milan, Italy, 13-15 June 2016.

[4] Cho, H. and Cho, C., 2008. Prediction of hot spots by correlating finite element analysis and measurement for an automotive disk brake. Tribology Transactions, 51(5), pp.609-620.

[5] Day, A.J., 2014. Braking of road vehicles. Butterworth-Heinemann.

[6] Oshinibosi A, Barton D and Brooks P (2019) Optimization of Thermal Performance and Weight of an Automotive Disc Brake for a High Performance Passenger Car. In: Proceedings of the EuroBrake 2019 Conference. 13-15 June 2016, Milan, Italy, 13-15 June 2019.

[7] Topouris, S. and Tirovic, M., 2019. Design synthesis and structural optimization of a lightweight, monobloc cast iron brake disc with fingered hub. Engineering Optimization, 51(10), pp.1710-1726.

[8] Li, Q., Steven, G.P., Xie, Y.M. and Querin, O.M., 2004. Evolutionary topology optimization for temperature reduction of heat conducting fields. International Journal of Heat and Mass Transfer, 47(23), pp.5071-5083.

[9] Qu, X., Pagaldipti, N., Fleury, R. and Saiki, J., 2016. Thermal Topology Optimization in OptiStruct Software. In 17th AIAA/ISSMO Multidisciplinary Analysis and Optimization Conference (p. 3829).

[10] Gersborg-Hansen, A., Bendsøe, M.P. and Sigmund, O., 2006. Topology optimization of heat conduction problems using the finite volume method. Structural and multidisciplinary optimization, 31(4), pp.251-259.

[11] Takalloozadeh, M. and Yoon, G.H., 2017. Development of Pareto topology optimization considering thermal loads. Computer Methods in Applied Mechanics and Engineering, 317, pp.554-579.

[12] Deaton, J.D. and Grandhi, R.V., 2013. Stiffening of restrained thermal structures via topology optimization. Structural and multidisciplinary optimization, 48(4), pp.731-745.

[13] Grześ, P., 2009. Finite element analysis of disc temperature during braking process. Acta mechanica et automatica, 3(4), pp.36-42.

[14] Abaqus 2014. Analysis User's Guide. Simulia Corp. Providence, RI, USA.

[15] Altair Hyperworks 2016. User's Guide. Altair Engineering Inc., Troy, Michigan.

[16] Sigmund, O. and Maute, K., 2013. Topology optimization approaches. Structural and Multidisciplinary Optimization, 48(6), pp.1031-1055.

[17] Yoon, G.H., 2010. Topological design of heat dissipating structure with forced convective heat transfer. Journal of Mechanical Science and Technology, 24(6), pp.1225-1233.

[18] Ansys Inc, 2021. ANSYS FLUENT theory guide. Canonsburg, PA, Release 2.

[19] Palmer, E., Mishra, R. and Fieldhouse, J., 2009. 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.

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