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***Finite Element Thermal–Structural and Fatigue Assessment of Eight Ventilated Brake-Disc
Geometries under Telemetry-Derived Repeated Motorsport Braking***

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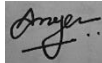
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Declaration of Authenticity

This dissertation is the result of the author's original research. It has been written by the author and has not been previously submitted for examination leading to the award of a degree.

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Date: 06/09/2026

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Abstract

Brake discs used in performance and motorsport applications are exposed to repeated high-energy braking events that generate large thermal gradients, thermo-mechanical stress and deformation. This study investigated the influence of ventilated brake-disc geometry on thermal, structural and fatigue-related behaviour using finite element analysis. Eight configurations were considered: Baseline, Dense Drill, Dense Groove, Less Drill, Less Groove, Thin, Thin Drill-Groove and Two-Piece discs.

Measured MoTeC vehicle telemetry was used to define six braking events over a representative 91.831 s lap. The sequence was repeated for five laps, giving a transient thermal simulation duration of 459.155 s. Event-specific reductions in vehicle kinetic energy were converted into heat-flux inputs and applied to the swept friction surfaces. Speed-dependent convection and thermal radiation were included before representative late-cycle temperature fields were transferred into static structural analyses. Equivalent von Mises stress, elastic strain, total deformation and fatigue-related outputs were then compared.

The Baseline Disc reached a peak temperature of 623.39°C and a final average temperature of 380.05°C. Dense Drill achieved the lowest peak temperature, 568.37°C, representing an 8.83% reduction, whereas Thin Drill-Groove produced the lowest final average temperature, 284.97°C, a reduction of 25.02%. These thermally strong designs introduced structural penalties. Dense Groove reached a peak temperature of 574.74°C and a final average temperature of 329.72°C while slightly reducing maximum equivalent stress from 255.54 MPa to 252.66 MPa.

The results show that improved cooling does not automatically produce improved structural performance. Within the assumptions and boundary conditions of the numerical model, Dense Groove provided the most favourable overall thermal–structural compromise among the modified single-piece configurations. Experimental validation, geometry-specific CFD, temperature-dependent material data and a more realistic floating-interface model for the Two-Piece Disc are recommended for future work.

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Chapter 1 – Introduction

1.1 Background to the Study

Braking systems are critical to vehicle safety and performance because they must convert vehicle kinetic energy into heat in a controlled and repeatable manner. In a disc-brake system, hydraulic pressure forces the pads against the rotating disc, generating friction at the contact interface. A large proportion of the vehicle's kinetic energy is then converted into thermal energy and absorbed by the brake disc before being dissipated through conduction, convection and radiation. Effective thermal management is therefore essential because excessive heat can alter friction behaviour, increase thermal distortion and contribute to structural degradation (Vdovin and Le Gigan, 2020; Li *et al.*, 2020; Park *et al.*, 2022).

The thermal demand becomes more severe during repeated braking. A disc may not cool back to its initial temperature before the next braking event, so residual heat is carried into subsequent events, and the bulk temperature can rise progressively. Grzes (2019) showed that repeated brake applications produce cumulative thermal behaviour that differs from a single braking event. This is especially relevant to motorsport, where several high-energy braking events may occur within each lap and the rotor repeatedly experiences heating and cooling cycles.

Temperature distribution is as important as the absolute peak temperature. Localised heating within the swept friction region creates thermal gradients, and the hotter regions attempt to expand more than the cooler surrounding material. This uneven expansion can generate thermal stress, deformation and localised hot spots. Goo (2018) demonstrated, using infrared measurement and finite element analysis, that brake-disc thermal behaviour can be spatially non-uniform and closely linked to thermo-elastic response. Consequently, a design that reduces temperature may not necessarily provide the most favourable structural performance.

Ventilated brake discs are widely used to improve heat rejection through internal air passages between the friction plates. Additional features such as drilled holes and grooves can further alter cooling behaviour by changing exposed surface area and conduction paths. However, such modifications also remove material and can introduce stress concentrations. Park *et al.* (2022) reported that holes and slit-type features can enhance heat dissipation while also affecting structural stability. Brake-disc design therefore involves a compromise between thermal performance, structural integrity and, in performance applications, component mass.

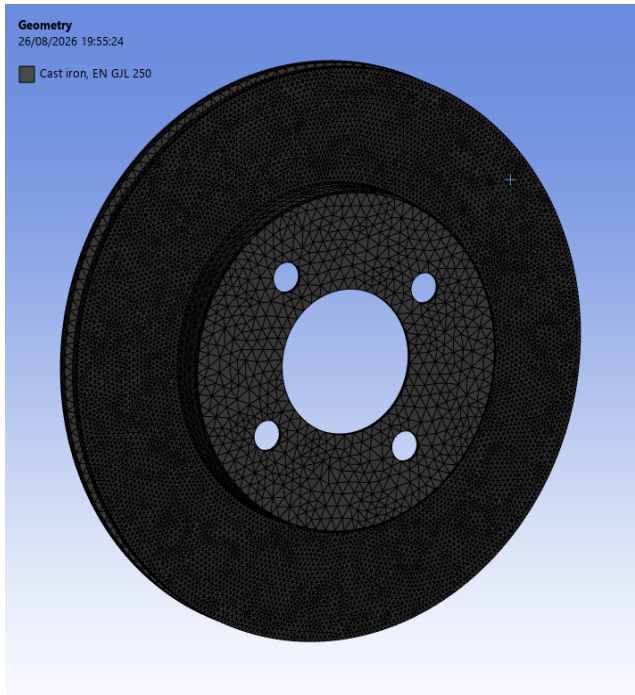


Figure 1: Baseline ventilated brake-disc geometry investigated in the present study.

1.2 Engineering Problem and Research Rationale

The engineering problem addressed in this research is how to improve brake-disc thermal performance under repeated braking without creating an unacceptable structural penalty. A brake disc must absorb significant thermal energy while remaining sufficiently stiff and strong to withstand the resulting thermal and mechanical loads. Geometric modifications intended to improve cooling may reduce peak temperature, but they can also reduce load-bearing material or increase stress concentration. Huang *et al.* (2022) and Park *et al.* (2022) demonstrate that changes in rotor geometry can influence both thermal behaviour and structural response, meaning that temperature alone is not an adequate basis for selecting an improved design.

This problem is particularly relevant to motorsport, where repeated heavy braking is combined with a requirement to minimise rotating and unsprung mass. Reduced-thickness rotors and two-piece discs can offer mass-saving potential, but a reduction in material can also alter thermal capacity and stiffness. Oshinibosi *et al.* (2021) treated brake-rotor design as a combined thermal and lightweighting problem, illustrating the need to consider competing performance requirements rather than one isolated output.

Finite element analysis provides an appropriate method for investigating these trade-offs because several geometries can be assessed under the same loading and boundary conditions. The present

study therefore compares eight brake-disc configurations: Baseline, Dense Drill, Dense Groove, Less Drill, Less Groove, Thin, Thin Drill-Groove and Two-Piece. Each design is exposed to the same telemetry-derived repeated-braking history so that relative differences in temperature, stress, strain, deformation and fatigue-related behaviour can be evaluated consistently.

1.3 Research Aim and Objectives

The aim of this study is:

To investigate the effect of ventilated brake-disc geometry on temperature distribution, thermo-mechanical response and fatigue-related performance under repeated braking conditions using finite element analysis.

To achieve this aim, the following objectives were established:

1. To critically review literature relating to brake-disc thermal behaviour, repeated braking, ventilation, drilled and grooved geometries, lightweight design and thermo-mechanical finite element analysis.
2. To develop a representative baseline ventilated brake-disc model and seven alternative geometries for comparison.
3. To derive a repeated-braking thermal load from MoTeC vehicle telemetry using vehicle speed and braking-event data.
4. To perform transient thermal analysis of all eight geometries under consistent heat-input, convection, radiation and ambient-temperature conditions.
5. To transfer representative thermal states into structural analyses and evaluate equivalent stress, elastic strain and total deformation.
6. To assess fatigue-related behaviour while recognising the limitations of highly localised numerical stress peaks and the adopted Stress-Life approach.
7. To compare the overall thermal and structural performance of the designs and identify the most favourable thermo-mechanical compromise.

1.4 Research Questions

The study addresses the following research questions:

1. How does brake-disc geometry influence temperature distribution under repeated braking conditions?
2. Which investigated geometry provides the greatest reduction in peak and accumulated disc temperature compared with the baseline design?
3. How do drilled, grooved, reduced thickness and two-piece configurations influence equivalent stress, elastic strain and total deformation?
4. What thermal–structural trade-offs are introduced by the different geometry modifications?
5. Which configuration provides the most favourable overall thermo-mechanical performance under the applied repeated-braking conditions?

These questions ensure that the designs are assessed using a combination of thermal and structural measures rather than temperature alone.

1.5 Scope of the Study

The investigation is limited to a comparative finite element assessment of the eight selected brake-disc configurations. Geometry is the primary variable, while the Two-Piece design also introduces a material change by a cast-iron friction ring, aluminium centre bell and steel fasteners. The same fundamental thermal and structural boundary conditions are retained across the models wherever possible.

Thermal loading is derived from MoTeC telemetry and represents five repeated laps, giving a total transient simulation time of approximately **459.15 s**. Heat flux is applied to the two swept friction surfaces during the identified braking periods. Cooling is represented by convection on the exposed and internal vane surfaces together with thermal radiation to an ambient environment of **22°C**. Explicit rotating pad-disc contact is not modelled; instead, frictional heat input is represented through the calculated heat-flux history.

The resulting temperature fields are transferred into static structural analyses at representative higher-temperature and cooling states. Equivalent von Mises stress, equivalent elastic strain and total deformation are then evaluated. Fatigue-related results are used as supporting indicators rather than exact service-life predictions because isolated mesh regions can produce very low minimum-life values that are not representative of most of the component.

For the Two-Piece Disc, the principal structural comparison focuses on the cast-iron friction ring and aluminium bell. Extreme local stresses occurring in bolts and nuts are considered separately because fastener and clamping behaviour is strongly dependent on the modelling assumptions used at the connections.

The study does not include dynamometer testing, thermocouple or infrared validation, detailed CFD of the cooling airflow, pad or disc wear, temperature-dependent friction, or a detailed floating-bobbin contact model. The results are therefore intended primarily for relative comparison of the investigated designs rather than exact prediction of real-world brake-disc service behaviour.

1.6 Significance and Contribution of the Study

The main contribution of this research is the direct comparison of eight substantially different brake-disc concepts within one consistent numerical framework. Existing studies commonly investigate design features or operating conditions, making direct comparison difficult when rotor dimensions, materials, loads and cooling assumptions differ. By applying the same repeated-braking history and modelling procedure to each configuration, the present work provides a clearer assessment of the influence of geometry itself.

A further contribution is the use of vehicle telemetry to define the thermal loading. Rather than applying a single arbitrary heat input, the model uses braking events derived from MoTeC data, producing repeated heating and cooling behaviour that is more representative of track operation. The study also combines thermal and structural assessment, recognising that improved cooling may be accompanied by increased stress, strain or deformation.

The inclusion of reduced-thickness and two-piece concepts extends the comparison towards lightweight brake design. Oshinibosi *et al.* (2021) highlighted the importance of balancing rotor mass reduction with thermal performance, while the present study additionally examines the associated structural response. The purpose is not to claim a universally optimum rotor, but to identify the relative advantages and disadvantages of the investigated geometries under the defined operating conditions.

1.7 Structure of the Dissertation

The remainder of the dissertation is organised into four chapters. Chapter 2 critically reviews literature concerning brake-disc thermal behaviour, geometry, materials and numerical modelling. Chapter 3 describes the geometry development, telemetry-derived loading and thermal-structural FEA methodology. Chapter 4 presents and discusses the thermal, structural and fatigue-related

results for all eight configurations. Chapter 5 summarises the principal findings, answers the research questions, identifies study limitations and provides recommendations for future work.

Chapter 2 – Literature Review

2.1 Introduction

Brake-disc design involves more than dissipating the heat generated during braking. The disc must absorb repeated thermal loads while maintaining sufficient stiffness, strength and durability. These requirements are closely linked because changes intended to improve cooling can also alter thermal capacity, stress distribution and deformation. Recent literature therefore treats brake-disc development as a combined thermal, structural and optimisation problem rather than a purely thermal one (Vdovin and Le Gigan, 2020; Park *et al.*, 2022; Xu *et al.*, 2026).

This chapter reviews research directly relevant to the present investigation. It first considers heat generation and repeated braking before examining ventilation, drilled and grooved geometries, thickness reduction, materials and two-piece construction. Thermo-mechanical stress, fatigue and finite element modelling are then evaluated. Attention is given to areas where published studies report competing thermal and structural effects, as these trade-offs form the basis of the eight-design comparison undertaken in this dissertation.

Literature synthesis: thermal-structural pathway in brake-disc design

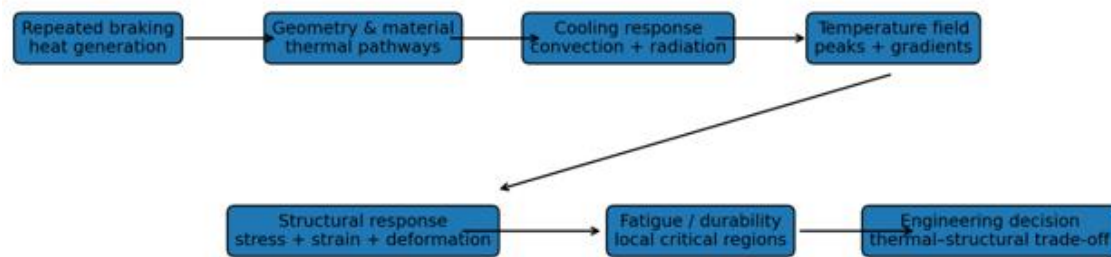


Figure 2: Principal themes and relationships considered in the literature review.

2.2 Heat Generation and Thermal Behaviour of Disc Brakes

During braking, friction between the pads and rotating disc converts part of the vehicle's kinetic energy into thermal energy. The resulting heat is initially concentrated within the swept friction region before being conducted through the rotor and rejected to the environment. The temperature reached by the disc consequently depends on the braking energy, material properties, rotor mass,

geometry and heat-transfer conditions. Vdovin and Le Gigan (2020) emphasise that brake-disc thermal behaviour is closely connected to airflow and heat rejection, making cooling conditions an important part of any realistic thermal model. Comparative modelling by Belhocine and Bouchetara (2012) also showed that ventilation changes the transient thermal response relative to a solid rotor.

An important feature of braking is that temperature is not uniformly distributed throughout the rotor. The friction track receives the greatest thermal input, while the centre and other unswept regions generally remain cooler. This produces thermal gradients that can cause different regions of the rotor to expand by different amounts. Goo (2018) demonstrated experimentally and numerically that temperature distribution in a disc-pad system can be non-uniform and is related to thermo-elastic behaviour and contact conditions.

This relationship between temperature and structural behaviour is central to brake-disc analysis. Belhocine and Bouchetara (2013) used a sequential thermo-mechanical approach in ANSYS in which a transient temperature field was calculated before being used to evaluate thermal stress and deformation. Their work illustrates that maximum disc temperature alone gives an incomplete description of brake performance because similar maximum temperatures can be accompanied by different thermal gradients and therefore different structural responses.

For this reason, the present investigation considers both maximum temperature and the accumulated thermal condition of the disc. Temperature results are subsequently transferred to structural analyses so that the mechanical consequences of the thermal field can be evaluated rather than considering thermal performance in isolation.

2.3 Repeated Braking and Heat Accumulation

A single braking event provides useful information about instantaneous brake-disc heating but does not fully represent motorsport operation. During repeated braking, the rotor may begin a new braking event before the heat generated during the previous event has been completely dissipated. Residual thermal energy therefore accumulates and progressively raises the bulk temperature of the component.

Grzes (2019) identified this inability to return to the initial temperature as a defining characteristic of repeated braking. The study evaluated mean, flash, maximum and bulk temperatures across successive braking applications and showed that these temperature measures do not necessarily follow identical trends.

This distinction is important for evaluating alternative rotor geometries. A design may reduce an instantaneous surface-temperature peak without necessarily providing the greatest reduction in accumulated disc temperature. Conversely, a geometry that initially reaches a relatively high temperature may dissipate heat more effectively during the cooling period. A comparison based only on the highest temperature could therefore overlook important differences in repeated-cycle performance.

Repeated heating and cooling also has structural implications. Expansion during braking is followed by contraction during cooling, producing cyclic thermo-mechanical loading. Bagnoli, Dolce and Bernabei (2009) linked repeated thermal stress cycles in grey-iron brake discs with residual tensile stress and thermal-fatigue cracking, demonstrating that repeated thermal loading contributes directly to durability concerns.

The use of five repeated laps in the present study is therefore more representative of track operation than analysis of an isolated braking event and allows both short-duration thermal peaks and accumulated temperature to be compared. Experimental and numerical work under repetitive braking also shows that the thermal state evolves with successive stops and cooling intervals (Jafari *et al.*, 2022).

2.4 Ventilation and Brake-Disc Cooling

Ventilated brake discs improve heat rejection by incorporating internal passages between the two friction plates. As the rotor turns, these passages can draw air into the disc and discharge it towards the outer circumference. Huang *et al.* (2022) describe a ventilated rotor as behaving in a manner like a centrifugal fan, illustrating why the detailed shape of the internal channels can have a strong influence on cooling performance.

The effectiveness of this process depends on vane geometry, ventilation-gap width, channel dimensions and rotational speed. Jafari and Akyüz (2022) numerically investigated nine geometric parameters of a radial ventilated brake disc and found ventilation-gap width to have the greatest influence on cooling. Increasing the gap from 8 mm to 14 mm reduced the calculated cooling time by approximately 21%, demonstrating that relatively modest geometric changes can significantly alter heat rejection.

Experimental and numerical evidence reinforces the importance of internal airflow. García-León, Afanador-García and Gómez-Camperos (2021) demonstrated that airflow speed and ventilation geometry affect heat transfer, while Sathiyar *et al.* (2024) measured the mass flow of air through

ventilated brake discs and found that it was strongly influenced by disc temperature and operating condition. Their tests also identified differences exceeding 10% between nominally similar discs, highlighting the sensitivity of brake cooling to geometry and physical conditions.

Numerical airflow studies also show that rotational speed and internal ventilation strongly influence convective heat transfer (García-León, Afanador-García and Gómez-Camperos, 2021; Jafari *et al.*, 2022). More detailed CFD methods can reproduce airflow through individual passages. Mingozi *et al.* (2023), for example, combined CFD with experimental testing when investigating heat dissipation from a pin-vented rotor. Such methods can improve understanding of local airflow but substantially increase computational demand.

This creates a modelling compromise. A full CFD solution for every braking event and every geometry would provide greater airflow detail but would be impractical for an eight-design, five-lap comparative study. The present research therefore distinguishes between external-surface and internal-vane convection while maintaining the same modelling approach across the geometries.

2.5 Drilled and Grooved Brake-Disc Geometries

Drilled holes and grooves are widely associated with performance brake rotors because they modify the friction surface and potentially improve heat rejection. However, these features do not provide an automatic performance advantage. Any material removed from the friction ring also changes its thermal mass, structural continuity and local stress distribution.

Park *et al.* (2022) directly examined this thermal–structural compromise using brake discs with different hole and slit configurations. Their analysis showed that holes and slits can enhance heat dissipation, but the discontinuities introduced by these features also become locations of concentrated stress and potential crack initiation. The study therefore treated temperature and structural stability as simultaneous optimisation objectives rather than assuming that greater cooling alone represents an improved design.

Yan *et al.* (2015) experimentally and numerically examined cross-drilled ventilated discs and showed that drilled passages can enhance cooling by altering internal flow and heat-transfer behaviour. This finding is particularly relevant to drilled rotors. Cross-drilling can increase convective heat transfer by modifying both internal and external flow, but the benefit depends on the detailed hole arrangement (Yan *et al.*, 2015). Increasing the number of holes increases the number of exposed internal surfaces and changes the conduction path through the friction ring. At the same time, each hole reduces material and introduces an edge around which stress can concentrate. Park *et al.* (2022) found that

the number, diameter, shape and arrangement of apertures all affected rotor performance. This suggests that there may be an optimum drilling configuration rather than a simple relationship in which additional holes always produce better results.

Grooved or slotted rotors alter the friction track differently because the features do not necessarily pass through the complete rotor thickness. Pranta *et al.* (2022) investigated modified ventilated rotors containing curved vents, holes and slots and found measurable differences in both temperature and structural response relative to a reference configuration. Their work again indicates that geometry should be assessed using several performance parameters rather than temperature alone.

Tauviquirrahman *et al.* (2023) examined changes in ventilation-hole and groove angles together with material variation. Their numerical findings showed that maximum temperature changed with hole and groove orientation and that geometric thinning could produce local areas of concentrated temperature. This is significant because it demonstrates that the thermal effect of a feature depends not only on whether a hole or groove exists, but also on its geometry and interaction with the surrounding rotor material.

The published evidence therefore does not support the assumption that the most heavily drilled or grooved disc must provide the best overall performance. A greater feature density may enhance heat rejection but could also reduce thermal capacity or increase local stress. This uncertainty provides the rationale for comparing Dense Drill with Less Drill and Dense Groove with Less Groove configurations in the present research.

2.6 Disc Thickness and Lightweight Design

Rotor thickness presents another important design compromise. A greater volume of material generally provides increased thermal capacity, allowing more energy to be absorbed for a given temperature rise. Increased thickness can also improve structural stiffness. However, these advantages are accompanied by increased rotating and unsprung mass.

Lightweighting is particularly important in performance and motorsport vehicles. Oshinibosi *et al.* (2021) investigated topology optimisation of an automotive brake rotor with the combined objectives of improving thermal performance and reducing weight. Their work highlights the importance of treating mass reduction as an optimisation problem rather than simply removing material.

Tauvqiirrahman *et al.* (2023) similarly observed that thinning brake-disc geometry could lead to local temperature concentration. This demonstrates that mass reduction can alter heat-transfer behaviour as well as overall thermal capacity.

The Thin Disc investigated in the present research therefore provides a means of determining whether a reduction in material produces a meaningful thermal or structural penalty. The Thin Drill-Groove design extends this comparison by combining lightweighting with surface features intended to influence cooling.

2.7 Brake-Disc Materials and EN-GJL-250

Grey cast iron remains widely used for automotive brake discs because it offers a practical combination of thermal conductivity, friction characteristics, damping, wear behaviour, manufacturability and cost (Li *et al.*, 2020). EN-GJL-250 was selected for the friction rings in the present study and is therefore particularly relevant when interpreting both thermal and fatigue results. The material designation is consistent with the grey cast-iron classification covered by BS EN 1561:2023 (British Standards Institution, 2024).

A major issue is that grey cast iron does not retain identical mechanical properties as temperature increases. Pevec *et al.* (2014) experimentally investigated EN-GJL-250 at room temperature and at 500°C, 600°C and 700°C. The researchers found a clear deterioration in fatigue performance as temperature increased. At 500°C, the tested samples achieved approximately half the fatigue life recorded at room temperature, while performance deteriorated much more significantly at 700°C.

The same study also demonstrated differences between tensile and compressive material behaviour. This matters because grey cast iron is relatively brittle and does not respond to multi-axial loading in the same way as a ductile steel. Equivalent von Mises stress is useful for maintaining a consistent numerical comparison between the investigated designs, but it should therefore not be interpreted as a complete brittle-fracture criterion for EN-GJL-250.

Temperature dependency also presents a limitation for fatigue prediction. If room-temperature S–N information is used while substantial regions of the rotor operate at several hundred degrees Celsius, the resulting life prediction cannot be expected to reproduce physical service life exactly. Consequently, the fatigue results in the present study are used primarily to support comparison between designs and are interpreted alongside stress distribution and fatigue contours.

Keeping the same cast-iron material across the principal single-piece geometries is nevertheless valuable because it allows the effect of geometry to be separated from the effect of material variation.

2.8 Two-Piece and Floating Brake Discs

A two-piece brake disc separates the friction ring from the central mounting bell. The friction ring can remain cast iron to provide the required braking surface, while the centre can be manufactured from a lighter aluminium alloy. This construction can reduce rotor mass and provides greater design flexibility than a conventional one-piece casting (Mîndru *et al.*, 2026).

A floating or semi-floating connection can also allow limited movement between the friction ring and bell as the ring expands with temperature. Mîndru *et al.* (2026) investigated the fixture arrangement of a floating brake disc and demonstrated that the connection layout itself is an important structural design parameter.

From a numerical perspective, however, the connection introduces substantial complexity. The predicted result can depend on bolt preload, contact friction, clearance, connection stiffness and the amount of permitted relative movement. An idealised bonded interface prevents movement that would occur in a real floating rotor and may therefore increase locally calculated thermal stresses.

This consideration is directly relevant to the Two-Piece Disc in the present study, which contains a cast-iron friction ring, aluminium bell and steel fasteners. Bonded contacts were used to obtain a stable comparative model. Consequently, very high stresses occurring within individual bolts or connection regions should not automatically be interpreted as representative stresses throughout the rotor.

The most meaningful comparison therefore concerns the thermal and structural response of the cast-iron ring and aluminium bell, while extreme local fastener values are discussed separately as a modelling and connection issue.

2.9 Thermo-Mechanical Stress, Deformation and Fatigue

Thermal stress develops when different parts of a component experience different temperatures and therefore attempt to expand by different amounts. Within a brake rotor, the swept friction region heats rapidly while the centre and surrounding areas can remain significantly cooler. The resulting differential expansion can produce stress even before mechanical clamping loads are considered.

Bagnoli, Dolce and Bernabei (2009) demonstrated that repeated braking thermal cycles can generate residual stresses associated with thermal-fatigue cracking in grey-iron brake discs. Their work highlights the importance of cyclic heating because the stress state during high-temperature braking can change as the disc cools.

Belhocine and Bouchetara (2013) similarly demonstrated the relationship between the transient temperature field and the resulting stress and deformation using sequential thermal–structural FEA. Their methodology supports transferring thermal results into a separate structural analysis rather than applying a single uniform disc temperature.

Geometry also has an important influence on where maximum stresses occur. Holes, groove ends, vane intersections, mounting features and sharp geometric transitions can all produce local concentrations. This creates an important issue when interpreting finite element output: the numerical maximum may occur over only a very small area or even a limited number of mesh nodes, while most of the component experiences substantially lower stress.

The same principle applies to fatigue calculations. A local peak can produce an extremely low reported minimum fatigue life even when most of the rotor remains within a much higher life range. For this reason, a single minimum value should not be used independently to rank brake-disc designs. Fatigue contours, stress distribution, damage and safety factor provide important context.

Furthermore, Pevec *et al.* (2014) demonstrated the strong temperature dependence of EN-GJL-250 fatigue behaviour. Accurate service-life prediction would therefore require appropriate high-temperature fatigue data and more detailed loading histories. The fatigue analysis used in this dissertation is consequently regarded as an additional comparative indicator rather than an exact prediction of physical rotor life.

2.10 Finite Element Modelling of Brake Discs

Finite element analysis is widely used in brake development because it provides access to temperature, stress, strain and deformation throughout complex geometries before physical prototypes are manufactured. Review literature confirms that thermal and thermo-mechanical finite element methods remain central to disc-brake design studies (Thakre, Shahare and Awari, 2021; Xu *et al.*, 2026). Reviews of disc-brake thermal modelling show that FEA is particularly useful for comparing ventilation and rotor concepts when the assumptions and boundary conditions are clearly controlled (Thakre, Shahare and Awari, 2021; Xu *et al.*, 2026). It is especially useful for comparative studies because alternative geometries can be exposed to identical numerical operating conditions.

Brake-disc models vary greatly in complexity. Fully coupled approaches may include rotating geometry, frictional pad-disc contact, transient heat generation and structural loading simultaneously. These models can provide detailed local behaviour but may involve difficult contact convergence and substantial computational cost. Sequential methods provide an alternative by first solving the transient thermal problem and then transferring selected temperature fields into structural analyses.

Belhocine and Bouchetara (2013) successfully used sequential thermo-mechanical coupling to calculate temperature, stress and deformation in ventilated brake discs. This approach is particularly appropriate when the research objective is comparative because all designs can be exposed to the same calculated thermal history.

Nevertheless, FEA predictions remain dependent on modelling assumptions. Vdovin and Le Gigan (2020) identify airflow representation, convective heat-transfer coefficients, surrounding components and thermal boundary conditions as important sources of uncertainty in brake thermal modelling.

Experimental validation is therefore desirable. Sathiyar *et al.* (2024) demonstrated that physical cooling behaviour can vary even between nominally similar rotors, while Mingozi *et al.* (2023) combined CFD with experimental measurements to assess brake-disc heat dissipation.

Full experimental validation was outside the practical scope of the present study. The numerical model is therefore intended primarily to identify relative differences between geometries under consistently applied assumptions, rather than claim exact prediction of every real-world brake-disc phenomenon.

2.11 Critical Synthesis and Research Gap

The literature demonstrates that brake-disc thermal and structural behaviour are closely interconnected. Improved heat rejection is desirable, but modifications intended to achieve it frequently introduce competing effects. Additional drilled holes may improve heat transfer while generating stress-concentration sites. Grooves modify the friction surface while removing material. Reduced thickness offers mass savings but lowers thermal mass and stiffness. Two-piece construction can reduce weight and permit differential material selection but introduces complex connections between the ring and bell.

This makes it difficult to identify one universally superior brake-disc geometry, a point also emphasised in the recent review by Xu *et al.* (2026). Published studies also use substantially different rotor dimensions, vehicle speeds, brake energies, materials, convection coefficients and modelling assumptions. A design found to perform well in one study cannot therefore be assumed to provide the same advantage under another braking condition.

A further limitation is the frequent separation of thermal optimisation from structural assessment. Research focusing on heat dissipation can favour designs with increased ventilation or greater surface modification, while structural studies may favour designs retaining more material. Park *et al.* (2022) directly demonstrated this conflict by treating thermal performance and structural stability as competing design requirements.

Similarly, detailed CFD and fully coupled contact models can provide increased physical detail but require substantial computational resources. Vdovin and Le Gigan (2020) show that brake thermal modelling inevitably involves assumptions concerning airflow and boundary conditions. A controlled comparative study can therefore provide useful engineering evidence even when every physical phenomenon is not explicitly represented, provided the assumptions are applied consistently and acknowledged.

The research gap addressed in this dissertation is consequently a controlled comparison of several distinct brake-disc design strategies under one repeated telemetry-derived braking condition. Eight geometries are evaluated using the same fundamental thermal and structural modelling procedure. The comparison considers peak temperature, accumulated temperature, stress, strain, deformation and fatigue-related behaviour rather than relying on one isolated performance measure.

The purpose is therefore not to establish a universally optimum brake disc, but to determine which of the investigated configurations provides the most favourable thermal–structural compromise within the operating conditions and assumptions adopted in this study.

Table 1 – Critical synthesis of literature directly relevant to the present investigation.

Study	Main area	Main finding	Relevance to this study

Bagnoli, Dolce and Bernabei (2009)	Thermal fatigue cracking	Repeated thermal stress can generate residual tensile stress and radial cracking in grey-iron brake discs	Supports the structural and fatigue discussion
Pevec <i>et al.</i> (2014)	EN-GJL-250 fatigue	Fatigue performance deteriorates substantially at elevated temperature	Defines an important limitation of simplified fatigue prediction
Goo (2018)	Thermo-elastic behaviour	Temperature and contact response are spatially non-uniform	Supports temperature-distribution and stress analysis
Grzes (2019)	Repeated braking	Maximum, mean and bulk temperatures evolve differently during repeated braking	Supports the five-lap repeated-event methodology
Vdovin and Le Gigan (2020)	Thermal/CFD modelling	Cooling predictions depend strongly on airflow and boundary-condition assumptions	Supports a controlled comparative modelling approach
Oshinibosi <i>et al.</i> (2021)	Lightweighting	Rotor mass reduction should be considered together with thermal performance	Supports the Thin and Thin Drill-Groove designs
Park <i>et al.</i> (2022)	Holes and slits	Cooling improvement can conflict with structural stability and stress concentration	Directly supports drilled/grooved comparison

Jafari and Akyüz (2022)	Ventilation	Ventilation-gap geometry strongly affects cooling time	Supports the treatment of internal ventilation
Tauviqueerahman <i>et al.</i> (2023)	Geometry/material	Hole, groove and thinning choices alter thermal response	Supports geometry comparison
Sathiyar <i>et al.</i> (2024)	Experimental cooling	Internal mass airflow affects disc cooling and varies with thermal condition	Supports convection discussion
Mindru <i>et al.</i> (2026)	Floating rotor	Connection layout influences two-piece brake-disc behaviour	Supports two-piece modelling limitations
Xu <i>et al.</i> (2026)	Review of thermal optimisation	Recent research confirms the multi-variable nature of brake thermal design and optimisation	Supports the research gap and multi-objective approach

2.12 Chapter Summary

The literature demonstrates that brake-disc performance depends on the interaction between heat generation, cooling, geometry, material behaviour and structural constraint. Repeated braking is particularly demanding because heat accumulates between braking events, while cyclic temperature changes generate changing thermo-mechanical stresses.

Ventilation, drilling, grooving, thickness reduction and two-piece construction each offer potential advantages but can also introduce structural or modelling disadvantages. The literature therefore supports evaluating alternative discs through a combined thermal–structural framework rather than ranking them solely by maximum temperature. This provides the basis for the methodology presented in Chapter 3, where eight brake-disc geometries are subjected to a common telemetry-derived repeated-braking condition.

Chapter 3 – Methodology

3.1 Introduction

This chapter describes the methodology used to investigate the thermal and structural behaviour of eight ventilated brake-disc configurations under repeated braking conditions. The study adopted a numerical comparative approach using CAD geometry, measured MoTeC vehicle telemetry and finite element analysis in ANSYS Mechanical. The methodology was designed to maintain consistent loading and boundary conditions across the investigated geometries so that differences in performance could primarily be related to the design modifications.

The final numerical procedure consisted of a transient thermal analysis followed by sequential static structural analyses. Temperature fields obtained from representative late-cycle thermal states were transferred to the structural models, after which stress, strain and deformation were evaluated. A Stress-Life fatigue assessment was subsequently performed using the two structural states. Sequential thermal–structural analysis is established in brake-disc modelling because it allows transient temperature distributions to be used as thermal loads in a subsequent structural analysis (Belhocine and Bouchetara, 2013).

Overall simulation and analysis workflow

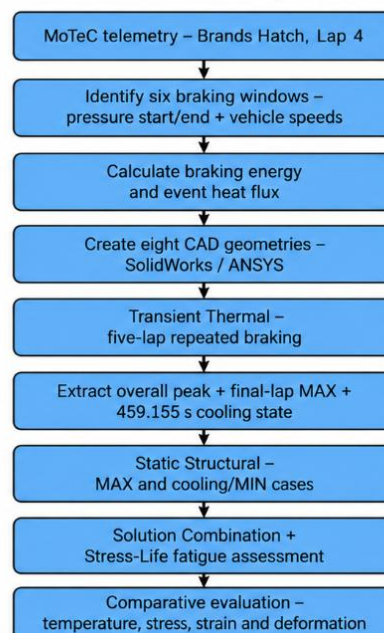


Figure 3: Numerical methodology adopted for the comparative brake-disc investigation.

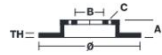
3.2 Baseline Brake-Disc Geometry and Design Variants

The baseline model was developed from the technical specification of the Brembo Max 09.5166.75 ventilated front brake disc (Brembo, no date). The component had an outside diameter of **256 mm**, nominal thickness of **20 mm**, minimum allowable thickness of **18 mm**, overall height of **39 mm**, a **65 mm centring bore**, four mounting holes and a ventilated internal construction. These dimensions provided the principal geometric envelope from which the baseline CAD model was produced.

Table 2 – Baseline brake-disc dimensions

Parameter	Value
Disc type	Ventilated front brake disc
Manufacturer / part	Brembo Max 09.5166.75
Outside diameter	256 mm
Nominal thickness	20 mm
Minimum thickness	18 mm
Overall height	39 mm
Centring bore	65 mm
Number of mounting holes	4
Axle position	Front

Technical specifications



Diameter Ø 256 mm	Thickness (TH) 20 mm	Height (A) 39 mm
Number of holes (C) 4	Brake disc type Ventilated	Centering (B) 65 mm
Min. thickness 18 mm	Tightening torque 110 Nm	Axle Front
Units per box 2	EAN code 8020584012154	

TECHNICAL DATA SHEET

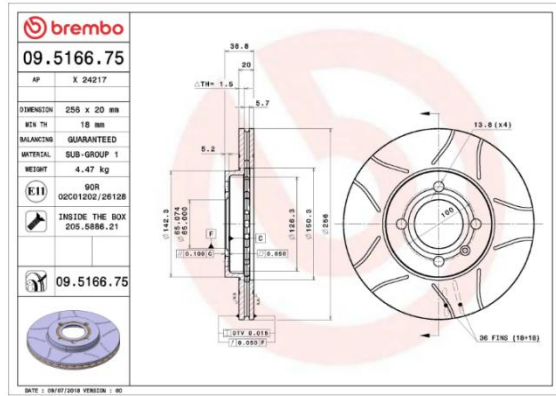


Figure 4: Technical specification used to define the baseline brake-disc geometry. Source: Brembo (no date).

Seven alternative designs were subsequently created while retaining the baseline as the reference configuration. The modifications were selected to examine different approaches to thermal management, surface modification and lightweighting. The complete design set consisted of the Baseline Disc, Dense Drill Disc, Dense Groove Disc, Less Drill Disc, Less Groove Disc, Thin Disc, Thin Drill-Groove Disc and Two-Piece Disc.

Table 3 – Investigated geometries

Design	Configuration	Main purpose
01	Baseline Disc	Reference ventilated geometry
02	Dense Drill Disc	Examine high-density drilled features
03	Dense Groove Disc	Examine high-density surface grooves
04	Less Drill Disc	Compare lower drilling density
05	Less Groove Disc	Compare lower groove density
06	Thin Disc	Investigate reduced material/thickness
07	Thin Drill-Groove Disc	Combine lightweighting with surface features
08	Two-Piece Disc	Investigate cast-iron ring with aluminium bell

The drilled and grooved designs were produced at two feature densities to determine whether increasing the number of surface modifications generated a proportional thermal advantage or

introduced additional structural penalties. The Thin and Thin Drill-Groove configurations investigated the effect of reducing material, while the Two-Piece Disc introduced both geometric and material variation.



Figure 5: Eight brake-disc geometries investigated in the study.

3.3 Material Definition

EN-GJL-250 grey cast iron was assigned to the friction rotor for the baseline and modified single-piece configurations. Maintaining the same principal friction material was important because it reduced the number of changing variables and allowed the investigation to focus primarily on geometry. Grey cast iron is widely used for brake-disc applications because of its thermal, friction and damping characteristics. However, its mechanical and fatigue behaviour changes significantly at elevated temperature. Pevec *et al.* (2014) demonstrated experimentally that the fatigue performance of EN-GJL-250 deteriorates as temperature increases, particularly between 500°C and 700°C.

For the Two-Piece Disc, EN-GJL-250 was retained for the outer friction ring, while wrought Aluminium 6061-T6 was assigned to the centre bell and structural steel to the bolts and nuts. This arrangement represented the intended lightweight two-material construction.

Table 4 – Representative material properties used in the finite element models.

Property	EN-GJL-250	Al 6061-T6	Structural steel	Unit
Density	7200	≈2700	≈7850	kg/m ³
Young's modulus	110	≈69	≈200	GPa
Poisson's ratio	0.28	≈0.33	≈0.30	–
Thermal conductivity	52	≈167	≈60	W/m·K
Specific heat	420	≈896	≈434	J/kg·K
Thermal expansion	1.1×10^{-5}	$\approx 2.36 \times 10^{-5}$	$\approx 1.2 \times 10^{-5}$	K ⁻¹

These properties were used consistently within the comparative models. A fully temperature-dependent material database was not developed for every constituent. This is recognised as a limitation because experimental research shows that EN-GJL-250 mechanical and fatigue behaviour changes at high brake operating temperatures (Pevac *et al.*, 2014).

3.4 MoTeC Telemetry and Braking-Event Selection

The thermal loading applied to the brake disc was derived from measured MoTeC telemetry rather than from an arbitrary constant heat input. The selected vehicle dataset originated from **Brands Hatch, Lap 4**, and provided vehicle-speed and brake-pressure information from which the braking periods could be identified. Earlier project development had used individual high-braking events to establish the calculation method before the final repeated-event loading table was constructed.

The final thermal input contained **six braking events within a 91.831 s lap**, which was repeated five times to produce a total simulation period of **459.155 s**. Consequently, the complete transient analysis represented 30 braking events.

Brake application and release were identified from the brake-pressure traces rather than simply using the complete period of vehicle deceleration. This was important because vehicle speed can continue to decrease after brake release because of aerodynamic drag, rolling resistance and other vehicle effects. Using the brake-pressure window ensured that the calculated frictional heat input corresponded more closely to the period during which the braking system was actively generating heat.

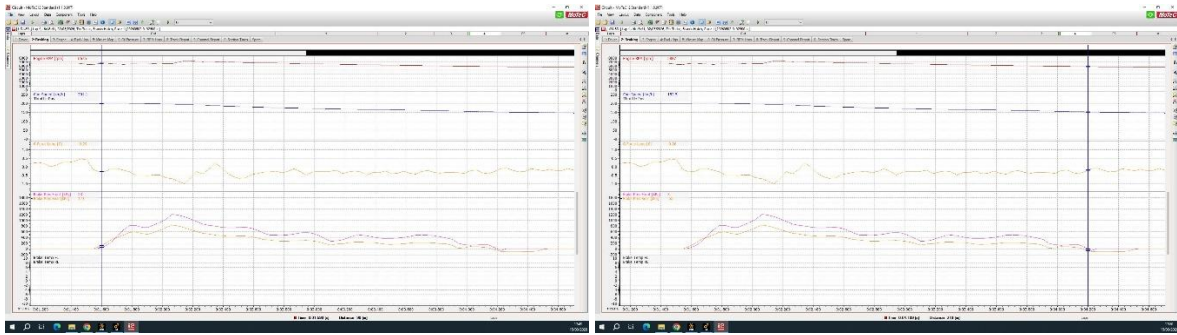


Figure 6: Representative MoTeC telemetry for Braking Event 1 showing (a) brake application at 1.599 s and (b) brake release at 4.199 s.

Table 5 – MoTeC braking events and final heat-flux values used for one lap.

Event	Start (s)	End (s)	Duration (s)	Start speed (km/h)	End speed (km/h)	Peak front (kPa)	Peak rear (kPa)	Heat flux used (MW/m ²)
1	1.599	4.199	2.600	201.1	152.5	1210	835	0.9705
2	10.664	13.422	2.758	171.6	90.9	1593	674	1.1278
3	29.135	30.870	1.735	168.5	127.6	1436	837	1.0249
4	46.652	47.876	1.224	205.9	173.8	1590	1068	1.4622
5	54.484	56.087	1.603	175.5	167.3	1710	1128	0.2575
6	63.623	65.039	1.416	182.6	157.2	1485	1058	0.8950

The final calculation sheet also showed an average pressure-derived front ratio of approximately 0.618, but the final ANSYS methodology retained a **front brake-bias factor of 0.55**, following the supervisor-guided calculation procedure. The measured pressure ratio was therefore used as a check rather than substituted directly into the final load calculation.

3.5 Braking Energy and Heat-Flux Calculation

The thermal energy produced during each braking event was estimated from the reduction in vehicle kinetic energy between brake application and brake release. The change in kinetic energy was calculated using:

$$\Delta E_k = \frac{1}{2}m(V_i^2 - V_f^2)$$

(Equation 3.1)

where:

m = vehicle running mass (kg),

V_i = vehicle speed at brake application (m/s), and

V_f = vehicle speed at brake release (m/s).

The vehicle running mass was taken as approximately 630 kg. Not all the calculated vehicle kinetic-energy reduction was assumed to enter a single front disc. Several allocation factors were therefore applied.

The energy entering one front rotor was calculated as:

$$E_d = \gamma B_f \eta_b \left(\frac{\Delta E_k}{2} \right)$$

(Equation 3.2)

where:

$\gamma=0.88$ - assumed fraction of relevant braking heat entering the disc,

$B_f=0.55$ - adopted front brake-bias factor,

$\eta_b=0.90$ - fraction of vehicle energy reduction attributed to the braking system, and

the division by two represents the two brake discs on the front axle.

These values were engineering assumptions developed through the supervisor-guided modelling procedure and were held constant for all eight designs. The intent was therefore to provide a consistent comparative heat input rather than claim that the factors represented exact experimentally measured energy partitions.

The heat was distributed over the swept annular region on both sides of the disc. The swept area was calculated as:

$$A_s = 2\pi(r_o^2 - r_i^2)$$

(Equation 3.3)

where $r_o=0.130m$ and $r_i=0.107m$. This produced:

$$A_s \approx 0.03425 \text{ m}^2$$

The average heat flux for each event was then calculated from:

$$q''_{\text{calc}} = \frac{E_d}{A_s t_b}$$

(Equation 3.4)

where t_b is the braking-event duration.

Finally, the project input sheet applied a **0.95 scaling factor** to the calculated heat flux:

$$q''_{\text{ANSYS}} = 0.95 q''_{\text{calc}}$$

(Equation 3.5)

The 0.95 factor was retained from the final adopted ANSYS input methodology. Since no experimental calibration was available to independently validate this reduction, it was treated as a modelling assumption rather than a universal physical constant. The final calculation sheet confirms the use of $\gamma=0.88$, $Bf=0.55$, $\eta b=0.90$, $m=630 \text{ kg}$, $A_s=0.0342496 \text{ m}^2$ and the 0.95 scaling factor.

Heat flux was set to zero outside the braking periods. Therefore, the thermal loading followed a repeated pulse pattern rather than a continuous heat input.

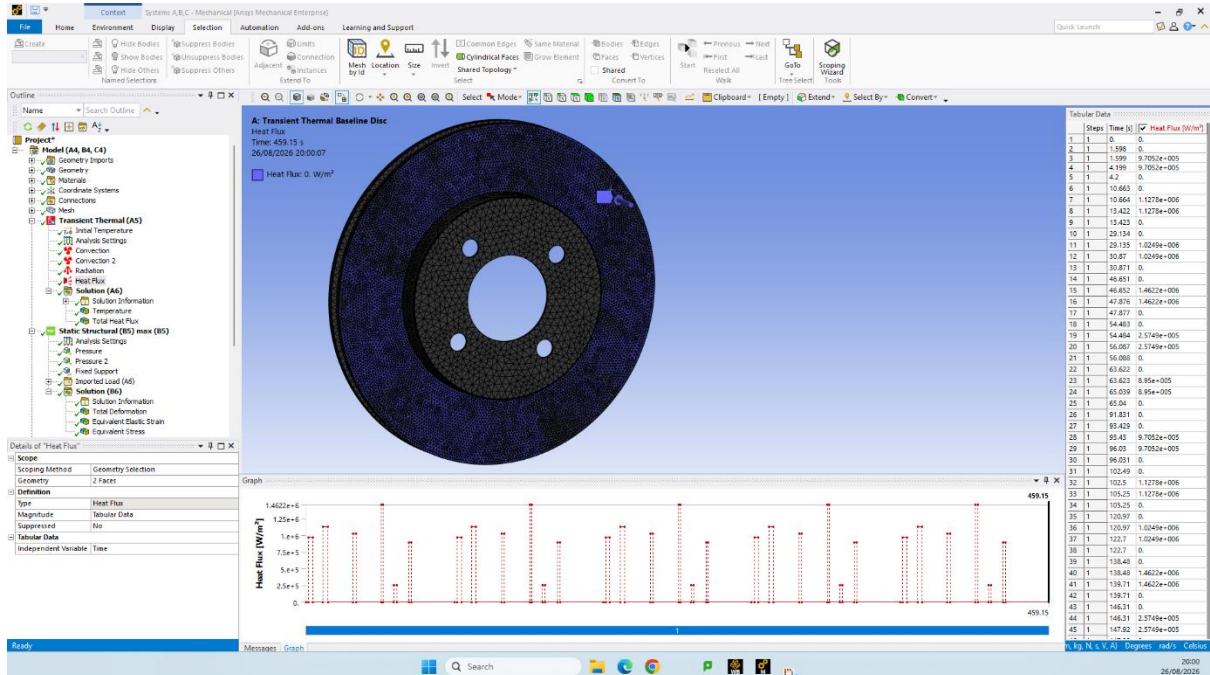


Figure 7: Heat-flux boundary condition applied to the two swept friction surfaces.

3.6 Thermal Boundary Conditions

Heat rejection from a rotating ventilated brake disc occurs through a combination of conduction within the material, convection to the surrounding air and radiation. Detailed CFD can be used to resolve airflow through individual ventilation channels, but this substantially increases computational cost and introduces additional modelling requirements (Vdovin and Le Gigan, 2020). Experimental work by Sathiyar *et al.* (2024) also confirms that airflow through ventilated-disc passages has an important influence on cooling behaviour.

For the present comparative study, airflow was therefore represented through prescribed speed-dependent convection coefficients. The external disc convection coefficient was calculated using the project correlation:

$$h_{\text{ext}}(t) = 58 \left(\frac{V(t)}{50} \right)^{0.5} \quad (\text{Equation 3.6})$$

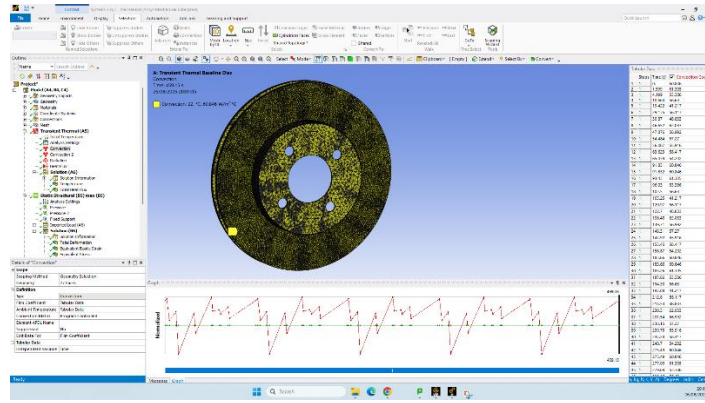
where $h_{\text{ext}}(t)$ is measured in $\text{W/m}^2\text{K}$ and $V(t)$ is vehicle speed in m/s . The reference value was therefore $58 \text{ W/m}^2\text{K}$ at 50 m/s .

The internal vane surfaces were assigned a higher coefficient:

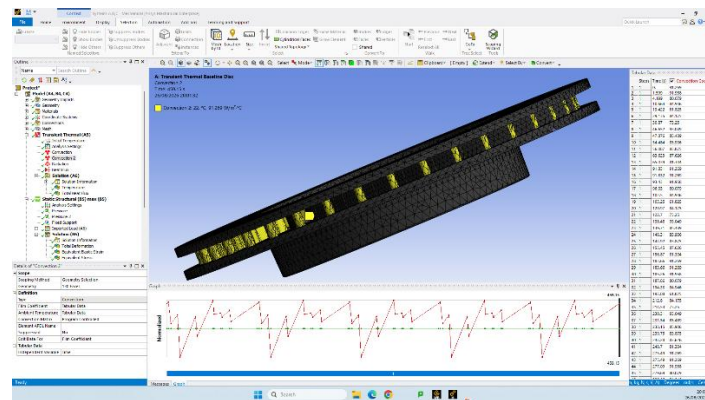
$$h_{\text{vane}} = 1.5 h_{\text{ext}} \quad (\text{Equation 3.7})$$

to represent the stronger airflow associated with the ventilated internal channels. The final project spreadsheet confirms the $58 \text{ W/m}^2\text{K}$ reference value and 1.5 vane multiplier.

The initial and ambient temperatures were both set to **22°C**. Thermal radiation was included with a constant surface emissivity of **0.8**. These assumptions were applied consistently across the geometries.



(a) exposed external disc surfaces



(b) internal ventilation-channel surfaces

Figure 8: Convection boundary-condition scoping: (a) exposed external disc surfaces and (b) internal ventilation-channel surfaces.

3.7 Mesh and Transient Thermal Analysis Settings

A finite element mesh was generated separately for each geometry because the drilled, grooved and two-piece configurations contained different geometric features. The general meshing approach was kept as consistent as practical, while sufficient resolution was maintained around surface modifications, ventilation passages and component interfaces. Mesh quality and geometry representation were visually reviewed before each final solution.

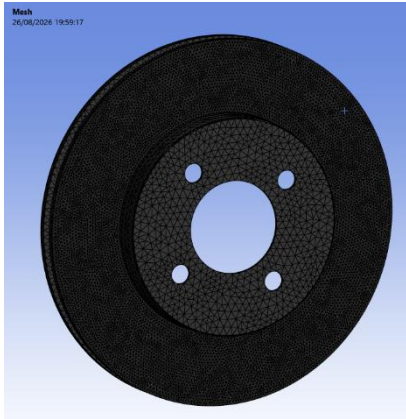


Figure 9: Finite element mesh applied to the baseline ventilated brake disc.

The final transient analysis represented five repetitions of the **91.831 s** lap, producing an end time of **459.155 s**. Automatic Time Stepping was enabled.

Table 6 – Transient thermal settings

Setting	Final value
Number of laps	5
Lap duration	91.831 s
Analysis end time	459.155 s
Initial temperature	22°C
Initial time step	0.1 s
Minimum time step	0.01 s
Maximum time step	0.5 s
Automatic Time Stepping	On
Radiation emissivity	0.8

These are the final values recorded in the ANSYS input workbook.

The main thermal outputs extracted for comparison were maximum temperature, temperature distribution and final average temperature. Temperature-time histories were also reviewed to ensure that the model displayed the expected repeated heating and cooling response.

3.8 Selection of Thermal States for Structural Analysis

The overall peak temperature from the full five-lap simulation was retained as an important thermal performance metric. However, using the absolute overall peak for structural transfer could result in

different designs being compared at substantially different points within the repeated-braking history.

For this reason, the structural **MAX** condition was selected from the final lap. For each geometry, the maximum temperature trace was reviewed between approximately **367 s and 459.155 s**, and the highest local peak within this final-lap region was selected as the Imported Body Temperature source time. This provided a consistent late-cycle, heat-soaked comparison.

The second structural condition was the final thermal state at **459.155 s**. This was referred to throughout the ANSYS models as the **MIN** or cooling case. The term MIN is a project label and does not imply that the temperature at 459.155 s was the absolute lowest temperature anywhere in the five-lap transient; rather, it represents the selected cooler end-of-cycle comparison state.

ANSYS supports sequential thermal-stress analysis by transferring a temperature solution into a structural system through an Imported Body Temperature load, with the required thermal source time selected before mapping (Ansys, 2026a).

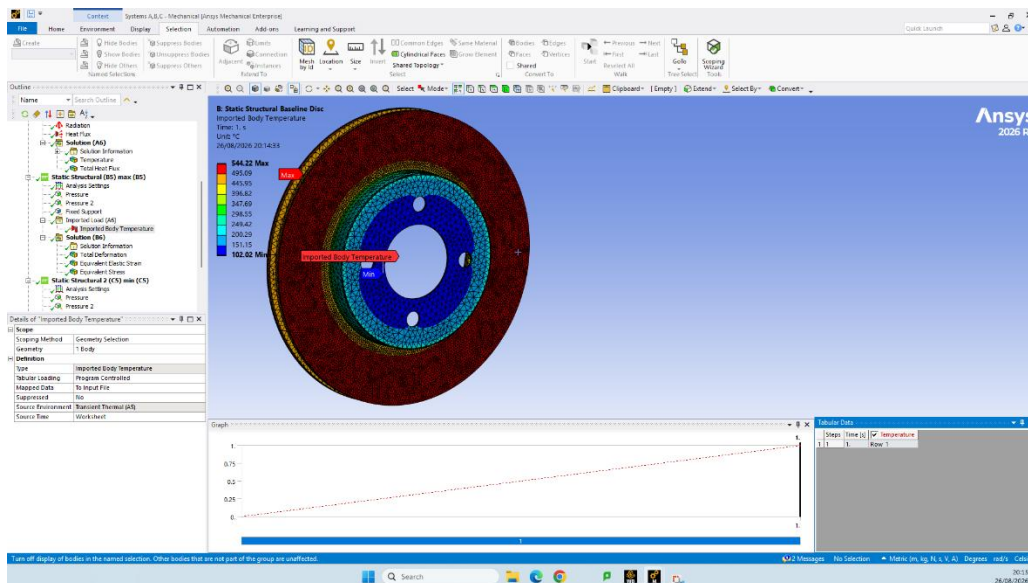


Figure 10: Imported Body Temperature used for the representative late-cycle MAX structural case.

3.9 Static Structural Analysis

Two linked Static Structural analyses were created for each disc geometry: one for the final-lap MAX condition and one for the cooling condition. The thermal fields were imported from the preceding transient analysis and acted as temperature-dependent structural loads.

The rear mounting face of the central hub was constrained using a fixed support to represent attachment to the vehicle drive flange. This is an idealisation because the real hub connection has finite stiffness, but it provided a repeatable constraint across all designs.

Mechanical pad loading was represented by a uniform pressure of:

$$p = 251\,000 \text{ Pa} = 251 \text{ kPa}$$

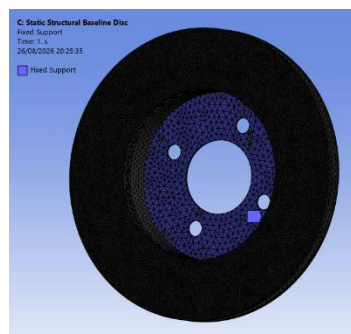
(Equation 3.8)

applied to each of the two swept friction faces. This approach avoided explicit brake-pad bodies and nonlinear rotating contact while retaining a representative mechanical loading on the rotor. The final project model therefore treated the frictional heat input and mechanical pad load as distributed loads over the swept annulus.

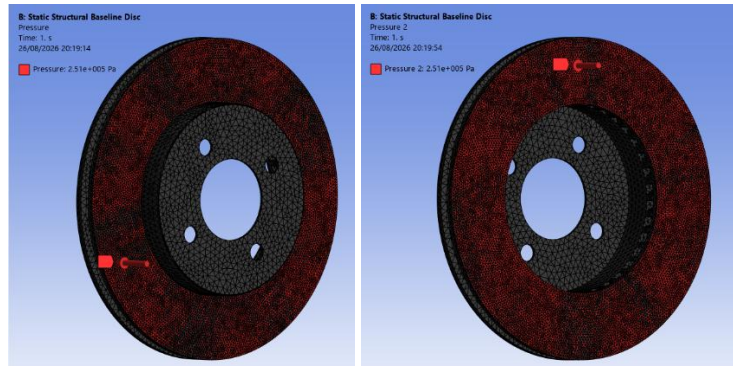
Three principal outputs were evaluated from both structural states:

- total deformation;
- equivalent von Mises stress; and
- equivalent elastic strain.

Equivalent von Mises stress was used as a common comparative metric across the eight models. However, EN-GJL-250 is a brittle grey cast iron, and its tensile and compressive behaviour is not identical. The von Mises value was therefore interpreted primarily as a consistent comparative stress measure rather than a complete brittle-fracture criterion. The high-temperature behaviour reported by Pevec *et al.* (2014) further supports caution when interpreting absolute structural limits.



(a) fixed mounting region



(b) equivalent pressure applied to the swept friction surfaces.

Figure 11: Structural boundary conditions: (a) fixed mounting region and (b) equivalent pressure applied to the swept friction surfaces.

3.10 Two-Piece Disc Modelling

The Two-Piece Disc required additional modelling because it consisted of an EN-GJL-250 friction ring, an aluminium 6061-T6 centre bell and steel bolts and nuts. Two-piece or floating brake discs can permit relative movement between the hot friction ring and the carrier, reducing the extent to which thermal expansion is restrained. Connection design is therefore an important feature of such rotors (Mîndru *et al.*, 2026).

In the present model, automatically detected interfaces between the components were retained as bonded contacts. This was selected to achieve a stable comparative solution but does not reproduce the relative radial movement, bolt preload, interface friction or clearances of a real floating disc.

The complete assembly initially produced very high local stresses in the steel fixings. Following review with the project supervisor, the principal comparative results were therefore scoped to the cast-iron friction ring and aluminium bell, while abnormal bolt and nut stresses were retained as a separate modelling observation rather than being used to represent rotor stress.

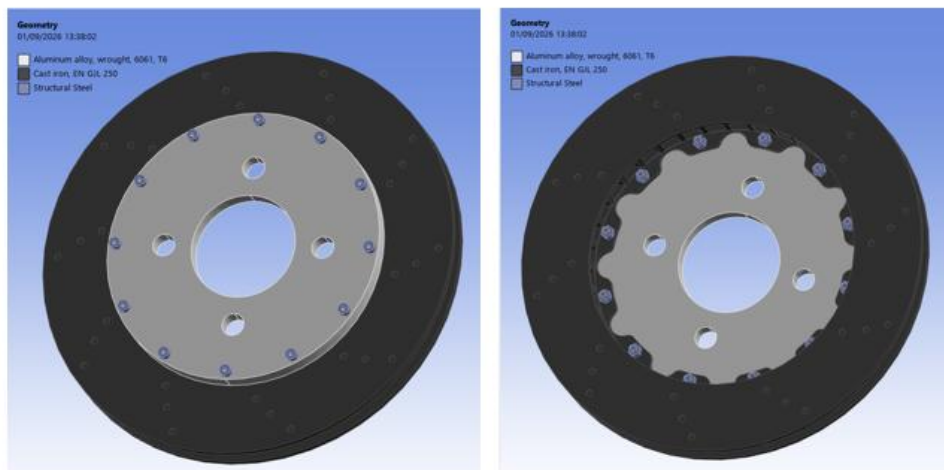


Figure 12: Two-piece brake-disc configuration showing the cast-iron friction ring and aluminium centre bell.

3.11 Fatigue Assessment

Following completion of the two structural analyses, their stress states were combined using the ANSYS Solution Combination system. A Stress-Life Fatigue Tool was subsequently applied.

Table 7 – Structural and fatigue settings

Setting	Definition
Structural states	Final-lap MAX and 459.155 s cooling condition
Pressure	251 kPa on each friction surface
Fatigue method	Stress-Life
Loading type	Non-Proportional
Mean stress correction	Goodman
Stress component	Equivalent von Mises
Fatigue strength factor	1.0
Scale factor	1.0
Outputs	Life, Damage, Safety Factor

The non-proportional loading option was used because fatigue was evaluated between two independently calculated structural stress states rather than treating the second condition as a simple scalar multiple of the first. ANSYS specifically provides Non-Proportional Stress-Life loading for a Solution Combination containing two stress states (Ansys, 2026b).

Life, damage and safety factor were extracted for comparative assessment. ANSYS defines the Fatigue Tool as capable of calculating these quantities using Stress-Life or Strain-Life methods.

The fatigue outputs were interpreted cautiously. In several geometries, the reported minimum fatigue life occurred within a very small high-stress region, while most of the disc contour showed substantially greater life. The final supervisory review indicated that isolated minima may be associated with only one or a small number of highly stressed mesh nodes and should not automatically be interpreted as the fatigue life of the complete brake disc. The spatial fatigue contour was therefore considered together with minimum life, damage and safety factor.

This interpretation is further justified by the temperature sensitivity of EN-GJL-250 fatigue behaviour demonstrated experimentally by Pevec *et al.* (2014). The Stress-Life values were consequently treated as comparative indicators, not precise predictions of physical service life.

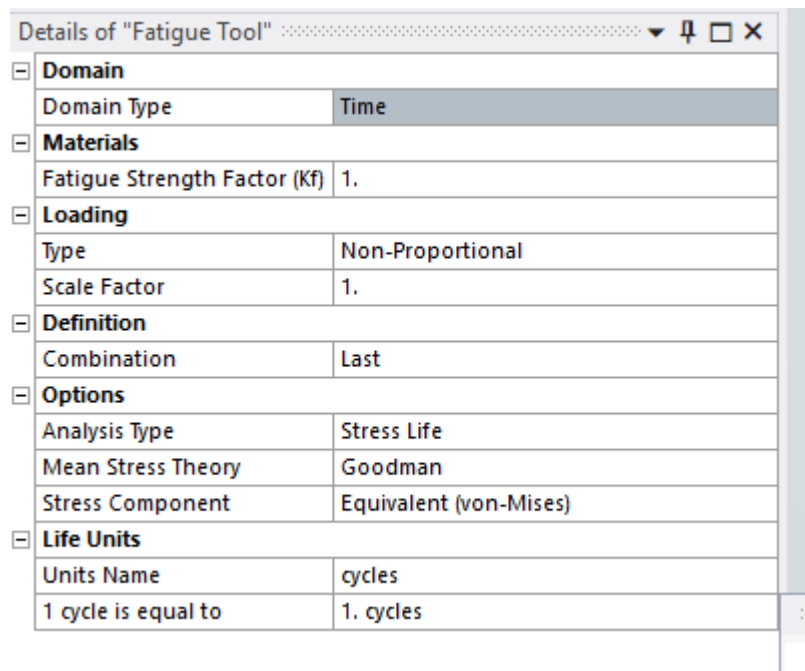


Figure 13: Stress-Life fatigue-analysis setup used in the Solution Combination system.

3.12 Model Reliability, Assumptions and Limitations

The final methodology was developed primarily to provide a controlled comparison between geometry variants. The same basic thermal loading, cooling assumptions, structural pressure and result definitions were maintained across the designs. All eight final simulation configurations solved successfully, and contour distributions were reviewed in addition to global numerical maxima and minima.

Full experimental validation was outside the available project scope. Model credibility was therefore assessed through physical consistency and comparison with published trends. Temperature histories were checked to confirm that braking produced rapid temperature increases followed by cooling periods, with residual heat carrying into subsequent events. Experimental research confirms that ventilated-disc cooling is strongly dependent on airflow and thermal condition (Sathiyan *et al.*, 2024), while brake-modelling literature emphasises the sensitivity of predicted temperatures to heat-transfer boundary conditions (Vdovin and Le Gigan, 2020).

Several simplifications remain. Frictional heating was applied as a distributed heat flux rather than generated through explicit rotating pad-disc contact. Cooling was represented through prescribed convection rather than full CFD. Pad and disc wear, temperature-dependent friction, detailed crack propagation and fully temperature-dependent material behaviour were not included. The structural model employed an idealised fixed mounting surface and uniform clamping pressure. The Two-Piece model used bonded contacts rather than a physically floating interface, and a comprehensive mesh-independence investigation was not completed for every long transient model.

These assumptions limit the reliability of the absolute predicted temperature, stress and fatigue-life values. They do not, however, prevent meaningful comparative analysis when the same assumptions are applied consistently. The conclusions of this research are therefore based primarily on relative differences and trends between the eight investigated geometries.

3.13 Chapter Summary

This chapter established the numerical methodology used to compare eight brake-disc configurations. MoTeC telemetry was converted into six event-specific heat-flux inputs and repeated over five laps for a total transient duration of 459.155 s. Speed-dependent convection, radiation and a common initial temperature were applied before representative final-lap thermal states were transferred into structural analyses. Stress, strain and deformation were evaluated under common mechanical loading, followed by a comparative Stress-Life fatigue assessment. The resulting thermal and structural data form the basis of the comparative analysis presented in Chapter 4.

Chapter 4 – Results and Discussion

4.1 Introduction

This chapter presents and critically discusses the thermal, structural and fatigue-related results obtained for the eight brake-disc configurations. The purpose of the analysis was not simply to identify the geometry producing the lowest temperature, but to determine whether improved thermal behaviour was accompanied by changes in stress, strain, deformation or fatigue-related performance. The Baseline Disc was therefore used as the reference against which the seven modified configurations were assessed.

The principal thermal measures were overall peak temperature and final average temperature following the five-lap repeated-braking simulation. Structural behaviour was assessed using equivalent von Mises stress, total deformation and equivalent elastic strain for both the representative final-lap MAX condition and the end-of-cycle cooling condition. Fatigue life, damage and safety factor were also evaluated; however, highly localised fatigue minima were interpreted using the complete contour rather than treating one scalar minimum as representative of the whole rotor. This approach followed the final supervisory review, which identified that isolated FEA maxima or minima may arise within only a small number of elements or nodes.

Table 8 summarises the complete numerical dataset used throughout this chapter.

4.2 Overall Results

Table 8 – Summary of thermal, structural and fatigue-related results

Design	Peak Temp. (°C)	Final Avg. Temp. (°C)	MAX Stress (MPa)	Cooling Stress (MPa)	MAX Def. (mm)	Cooling Def. (mm)	MAX Strain	Cooling Strain	Fatigue Life	Damage	Safety Factor
01 Baseline	623.39	380.05	255.54	244.20	0.75038	0.58188	0.0021513	0.0020864	2.7584 × 10 ⁶	362.53	0.88236
02 Dense Drill	568.37	329.38	271.23	248.31	0.76298	0.51438	0.0022091	0.0021089	Local min ≈ 0*	Local max*	0.56540
03 Dense Groove	574.74	329.72	252.66	234.19	0.74969	0.49433	0.0021066	0.0020256	1,204.1	8.3053 × 10 ⁵	0.65163
04 Less Drill	580.95	318.31	274.09	248.83	0.78908	0.49027	0.0022400	0.0021605	Local min ≈ 0*	Local max*	0.59395
05 Less Groove	573.66	333.84	259.55	235.44	0.75687	0.52526	0.0021635	0.0020735	Local min ≈ 0*	Local max*	0.64164
06 Thin	620.58	321.35	274.21	252.09	0.81574	0.50322	0.0022299	0.0021842	Local min ≈ 0*	Local max*	0.78661
07 Thin Drill-Groove	578.04	284.97	268.86	236.09	0.79033	0.43444	0.0021865	0.0020735	Local min ≈ 0*	Local max*	0.56581
08 Two-Piece Design	582.89	327.84	327.64	373.04	1.3448	0.79182	0.0063029	0.0065321	Local min ≈ 0*	Local max*	0.65709

The minimum fatigue-life and corresponding maximum-damage values occurred within highly localised regions and were not interpreted as representative whole-disc fatigue lives.

The percentage difference between each modified design and the Baseline Disc was calculated using:

$$\% \Delta X = \frac{X_{\text{modified}} - X_{\text{baseline}}}{X_{\text{baseline}}} \times 100$$

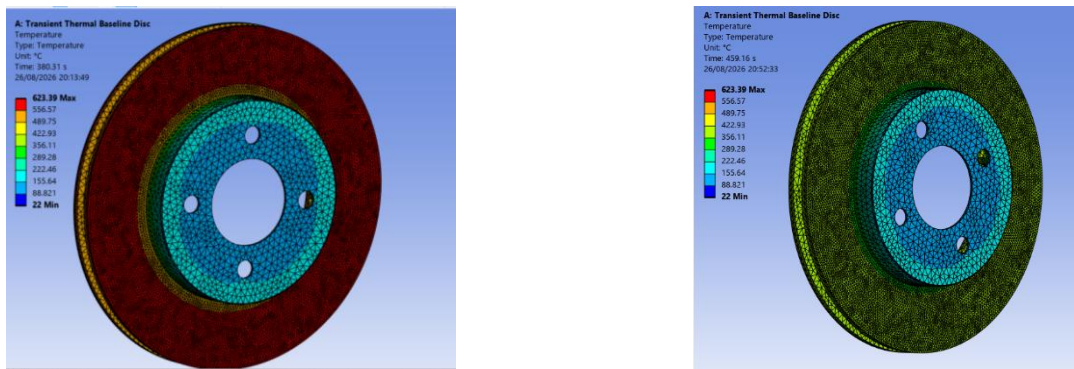
(Equation 4.1)

where X represents the performance parameter being compared. A negative percentage indicates a reduction relative to the baseline, while a positive percentage represents an increase.

4.3 Baseline Disc Response

The Baseline Disc provided the reference condition for the complete investigation. Its overall maximum temperature reached **623.39°C**, while its final average temperature after five repeated laps remained **380.05°C**. The transient response showed the expected pattern of rapid temperature increase during individual braking events followed by partial cooling. Because the disc did not return to the initial 22°C condition between successive braking events, residual heat accumulated and produced a progressively elevated bulk temperature.

This pattern agrees with the repeated-braking behaviour described by Grzes (2019), who demonstrated that the thermal state remaining after one braking event influences the initial condition of the next. Maximum and average temperature therefore describe different parts of the thermal problem. The 623.39°C peak represents the most severe local thermal condition, while the final average temperature provides a better indication of the quantity of heat remaining within the complete rotor after the repeated cycle.



(a) overall maximum-temperature condition. (b) final thermal condition following five repeated laps.

Figure 14: Baseline Disc temperature distribution

The structural MAX condition produced an equivalent stress of **255.54 MPa**, total deformation of **0.75038 mm** and equivalent elastic strain of **0.0021513**. At the cooling condition, stress reduced to **244.20 MPa**, while deformation reduced to **0.58188 mm**. These values established the structural reference against which the modified discs were compared.

The fatigue tool returned a minimum life of approximately **2.7584×10^6 cycles**, damage of **362.53** and a safety factor of **0.88236**. The safety factor remaining below unity indicates a demanding calculated cyclic condition within the adopted fatigue model. However, this should not be interpreted directly as evidence that the physical brake disc would fail under the five-lap simulation. Pevec *et al.* (2014) demonstrated that EN-GJL-250 fatigue properties change considerably with

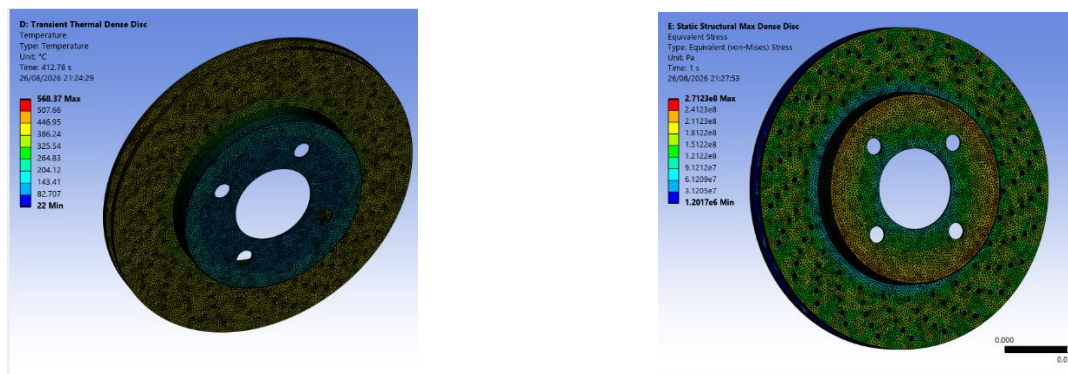
temperature, meaning that simplified Stress-Life predictions are sensitive to the material data and assumptions used.

The baseline therefore represents a relatively strong structural reference but also the poorest thermal configuration of the eight investigated designs.

4.4 Dense Drill Disc

The Dense Drill Disc produced the lowest overall peak temperature of the complete study, reaching **568.37°C**. This represents a reduction of approximately **8.83%** relative to the Baseline Disc. Its final average temperature was **329.38°C**, approximately **13.33% lower** than the baseline.

The result indicates that dense drilling was effective at reducing severe local temperature. Additional exposed surfaces around the drilled features can provide alternative heat-transfer paths and increase the surface area available for cooling. Park *et al.* (2022) similarly found that drilled brake-disc features can improve heat dissipation, although their performance depends on hole geometry, size and distribution.



(a) Dense Drill maximum-temperature contour. (b) Dense Drill MAX equivalent-stress contour.

Figure 15: Dense Drill Disc: (a) maximum-temperature distribution and (b) equivalent von Mises stress at the representative MAX structural state.

The improved thermal response was accompanied by a structural penalty. MAX stress increased from **255.54 MPa to 271.23 MPa**, an increase of approximately **6.14%**. Maximum deformation increased by approximately **1.68%** to **0.76298 mm**, while equivalent strain increased by approximately **2.69%** to **0.0022091**.

This behaviour is physically reasonable because every drilled feature interrupts the continuity of the friction ring. Although drilling increases exposed surface area, it also removes load-bearing material

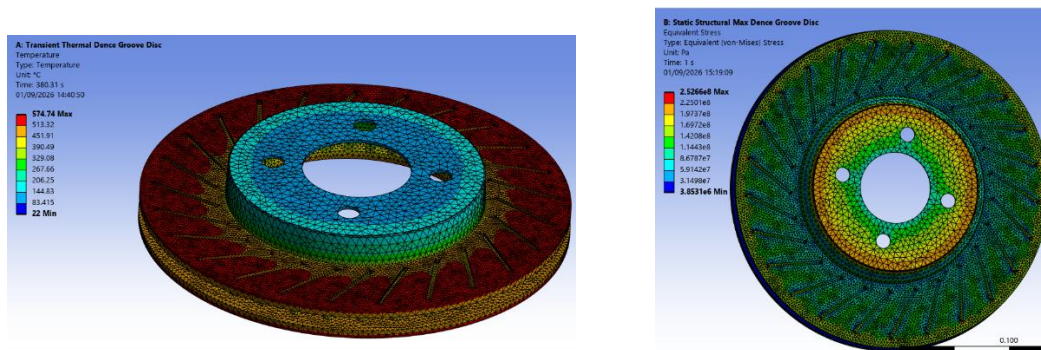
and introduces geometric edges around which local stress may concentrate. Park *et al.* (2022) identified this exact thermal–structural conflict, showing that brake-disc holes can enhance cooling while simultaneously introducing potential stress-concentration locations.

The fatigue results also required caution. ANSYS reported a highly localised minimum life approaching zero in a small region, rather than over most of the rotor. Following supervisory review, this scalar value was therefore not interpreted as the service life of the complete disc. Nevertheless, the minimum safety factor of **0.56540** was substantially lower than the baseline value of 0.88236, suggesting a more severe local cyclic condition.

Dense Drill can therefore be regarded as the most effective geometry for suppressing peak temperature, but it did not provide the strongest overall thermal–structural compromise.

4.5 Dense Groove Disc

The Dense Groove Disc produced an overall peak temperature of **574.74°C**, representing a reduction of approximately **7.80%** compared with the Baseline Disc. Its final average temperature was **329.72°C**, approximately **13.24% below baseline**. Although the peak was around 6.4°C higher than Dense Drill, the structural behaviour was noticeably more favourable.



(a) Dense Groove maximum-temperature contour.

(b) Dense Groove MAX stress contour.

Figure 16: Dense Groove Disc: (a) maximum-temperature distribution and (b) equivalent stress at the representative MAX structural condition.

MAX equivalent stress decreased slightly from **255.54 MPa to 252.66 MPa**, a reduction of approximately **1.13%**. Maximum deformation reduced slightly to **0.74969 mm**, while maximum equivalent strain reduced to **0.0021066**, approximately **2.08% below baseline**. The cooling-state stress was also lower than baseline at **234.19 MPa**.

This combination is significant. Unlike the drilled configurations, the Dense Groove Disc achieved a substantial temperature reduction without increasing the principal static structural measures. A plausible explanation is that shallow grooves altered the surface heat-transfer characteristics while retaining greater material continuity through the friction ring than through-drilled holes. This interpretation agrees with Park *et al.* (2022), who demonstrated that slit-type features can provide a useful balance between thermal performance and structural stability.

The fatigue calculation returned a minimum life of **1,204.1 cycles**, damage of **8.3053×10^5** and safety factor of **0.65163**. These values were less favourable than the baseline fatigue result. The fatigue outcome therefore prevents the Dense Groove configuration from being described as superior in every category.

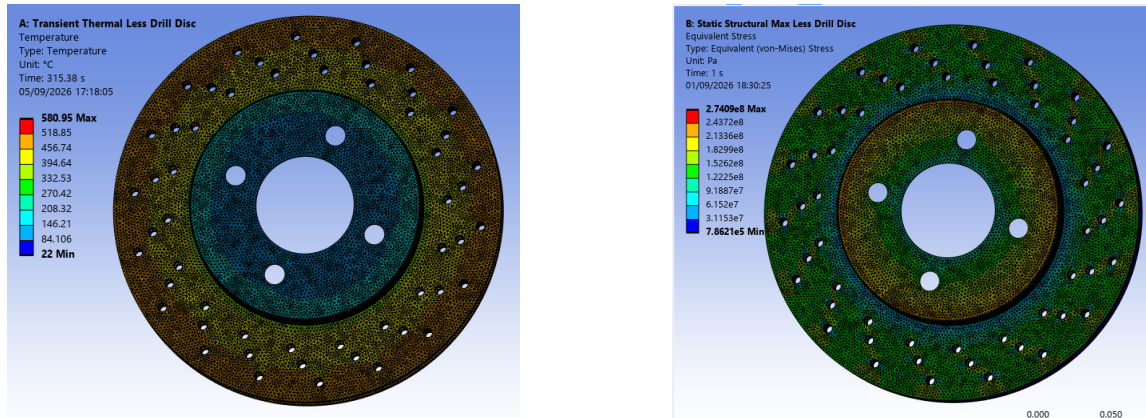
However, because the fatigue model is sensitive to local stress fields and simplified high-temperature material data, the principal conclusion from this geometry is that it provided a great thermal improvement while maintaining static stress, strain and deformation at approximately baseline levels or slightly below.

This makes Dense Groove particularly important when the overall engineering compromise is considered later in the chapter.

4.6 Less Drill Disc

The Less Drill Disc reached a peak temperature of **580.95°C**, approximately **6.81% below baseline**. Its final average temperature was **318.31°C**, a larger reduction of approximately **16.25%**.

The result is interesting because the Less Drill configuration produced a higher peak temperature than Dense Drill but a lower final average temperature. This confirms that peak temperature and accumulated heat should not be treated as interchangeable measures. Grzes (2019) similarly distinguished between maximum and bulk thermal behaviour during repeated braking.



(a) maximum-temperature contour.

(b) MAX stress contour.

Figure 17: Less Drill Disc thermal and structural response under repeated braking.

The structural response was less favourable. MAX stress increased to **274.09 MPa**, approximately **7.26% above baseline**. Maximum deformation increased to **0.78908 mm**, while strain increased to **0.0022400**.

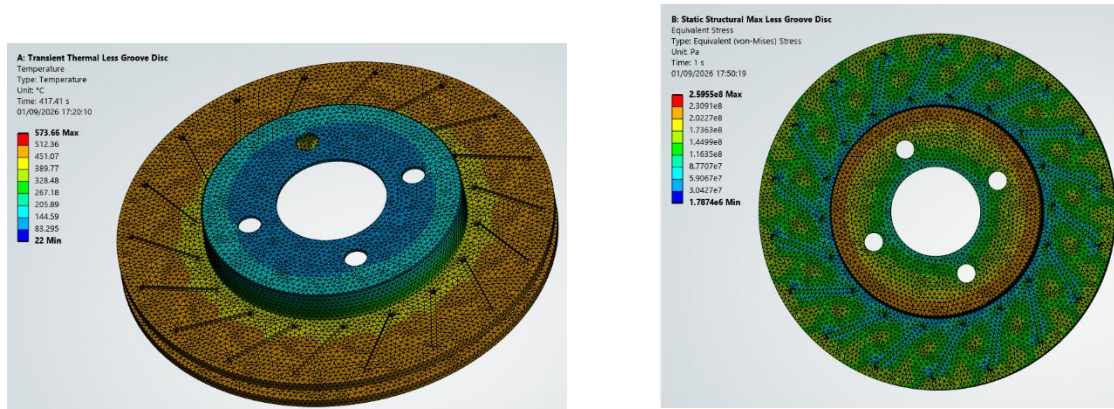
The fact that Less Drill produced a slightly greater stress increase than Dense Drill demonstrates that feature count alone cannot explain the structural result. Hole position, spacing, proximity to the vane structure and interaction between local stress fields are also important. This supports the design-optimisation findings of Park *et al.* (2022), where hole arrangement was shown to be as important as the simple presence or number of drilled features.

The fatigue safety factor was **0.59395**, again below baseline. Consequently, Less Drill achieved effective reduction in accumulated temperature but did not provide an improvement in structural performance.

4.7 Less Groove Disc

The Less Groove Disc reached a maximum temperature of **573.66°C**, approximately **7.98% below baseline**, and a final average temperature of **333.84°C**, approximately **12.16% lower**.

Interestingly, the peak temperature was slightly lower than that of Dense Groove despite the lower number of groove features. The difference between **573.66°C** and **574.74°C** is small and should not be overinterpreted because modelling uncertainties such as mesh discretisation and prescribed convection are larger than this sub-two-degree difference.



(a) Maximum-temperature contour.

(b) Max stress contour.

Figure 18: Less Groove Disc temperature and equivalent-stress distributions.

MAX stress was 259.55 MPa, only 1.57% greater than baseline. Maximum deformation was 0.75687 mm, and strain was 0.0021635. Cooling-state stress reduced to 235.44 MPa, which was lower than the Baseline Disc.

The structural results therefore remained much closer to the baseline than those of the drilled configurations. Both grooved designs appear to have preserved more of the stiffness and continuity of the friction ring while still providing useful thermal improvement.

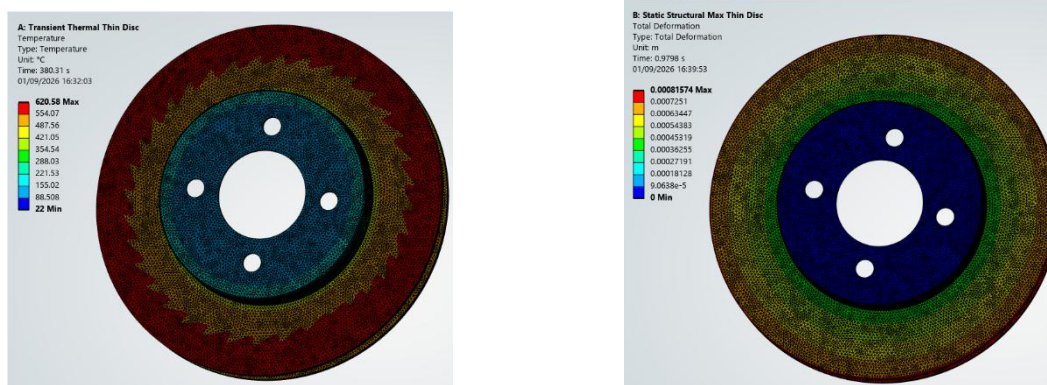
Dense Groove and Less Groove can therefore both be considered balanced designs. Dense Groove produced a slightly lower final average temperature and lower MAX stress, whereas Less Groove produced the marginally lower peak. Given the small magnitude of the difference, the wider structural trend is more important than the exact one-degree thermal ranking.

4.8 Thin Disc

The Thin Disc produced a peak temperature of **620.58°C**, only approximately **0.45% below the baseline**. This was the smallest reduction in peak temperature among all modified designs. However, its final average temperature fell substantially to **321.35°C**, approximately **15.45% below baseline**.

This result illustrates the competing effects of reducing disc material. Lower thermal mass can cause local temperature to rise rapidly because there is less material available to absorb energy. At the same time, the increased surface-area-to-volume relationship can allow the stored heat to be rejected more rapidly between braking events. Tauviqirrahman *et al.* (2023) similarly reported that

geometric thinning can alter temperature concentration, while Huang *et al.* (2022) showed that friction-plate thickness has a strong influence on thermo-mechanical deformation.



(a) Thin Disc maximum-temperature contour.

(b) Total Deformation MAX contour.

Figure 19: Thin Disc: (a) maximum temperature and (b) total deformation at the representative MAX structural condition.

The structural penalty was more evident. MAX stress increased to **274.21 MPa**, approximately **7.31% above baseline**, while deformation increased by approximately **8.71% to 0.81574 mm**. MAX strain increased to **0.0022299**.

The increased deformation is consistent with the reduced bending stiffness expected when material thickness is decreased. The Thin Disc therefore provided useful end-of-cycle cooling but almost no improvement in the most severe peak temperature, while simultaneously producing the largest deformation among the single-piece configurations.

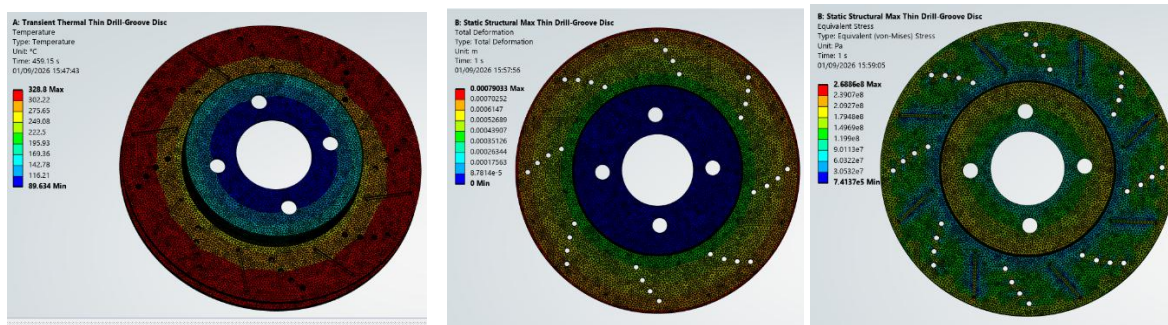
Its fatigue safety factor of **0.78661** was the highest of the modified designs, although still below the baseline. This demonstrates that individual performance parameters do not always produce the same ranking. The main justification for the thin design would therefore be potential mass reduction rather than maximum thermal or structural performance.

4.9 Thin Drill-Groove Disc

The Thin Drill-Groove Disc produced the lowest final average temperature of all eight configurations, reaching **284.97°C** after five laps. Relative to the baseline value of 380.05°C, this represents a reduction of approximately **25.02%**, or approximately **95°C**. Its overall peak temperature was also reduced to **578.04°C**, approximately **7.27% below baseline**.

This result indicates that combining reduced thermal mass with additional surface features produced particularly effective heat rejection over the repeated cycle. Although the maximum local

temperature remained above that of Dense Drill, the rotor retained considerably less thermal energy by the end of the simulation.



(a) Maximum-temperature contour. (b) MAX total-deformation contour. (c) MAX equivalent-stress contour.

Figure 20: Thin Drill-Groove Disc thermal and structural response: (a) maximum temperature, (b) MAX total deformation and (c) MAX equivalent von Mises stress.

The structural results were less favourable. MAX equivalent stress increased to **268.86 MPa**, approximately **5.21% above baseline**, and maximum deformation increased to **0.79033 mm**. Equivalent strain increased to **0.0021865**. Interestingly, cooling-state deformation reduced to **0.43444 mm**, substantially below the baseline cooling-state value, reflecting the much lower end-of-cycle thermal condition.

Its minimum fatigue safety factor was **0.56581**, almost identical to Dense Drill and among the lowest values obtained. The result therefore provides one of the clearest examples of the multi-objective nature of brake-disc development. If the designs were ranked purely by final average temperature, Thin Drill-Groove would appear to be the best configuration. Once stress, deformation and fatigue-related indicators are included, that conclusion is no longer justified.

The design can therefore be described as providing the strongest accumulated cooling performance, but not the most favourable overall thermal–structural balance.

4.10 Two-Piece Disc

The Two-Piece Disc reached an overall peak temperature of **582.89°C**, approximately **6.50% below baseline**, and a final average temperature of **327.84°C**, approximately **13.74% lower**. Its thermal response was therefore clearly improved compared with the Baseline Disc.

The structural behaviour differed considerably from the other seven designs because the assembly contained an EN-GJL-250 friction ring, aluminium 6061-T6 bell and steel fixings. When the results were scoped to the cast-iron ring and aluminium bell, MAX equivalent stress was **327.64 MPa**, while

maximum deformation reached **1.3448 mm**. These represent increases of approximately **28.21%** and **79.22%**, respectively, compared with the baseline. MAX equivalent strain was **0.0063029**, almost three times the baseline value.

At the cooling condition, equivalent stress increased further to **373.04 MPa**, despite the lower disc temperature. This initially appears counter-intuitive but is mechanically possible. Structural stress depends on the spatial temperature gradient and the restraint imposed on thermal expansion and contraction rather than simply the highest temperature present. During cooling, the cast-iron ring and aluminium bell attempt to contract by different amounts because of their different coefficients of thermal expansion.

A real floating two-piece disc is designed to permit some relative movement between these components. Mîndru *et al.* (2026) showed that connection arrangement and fastener layout have a measurable influence on floating-disc deformation and structural stability. In the present model, however, the bonded contacts constrained relative radial movement. It is therefore reasonable to interpret the high strain and cooling stress as a combination of genuine dissimilar-material behaviour and the idealised bonded connection.

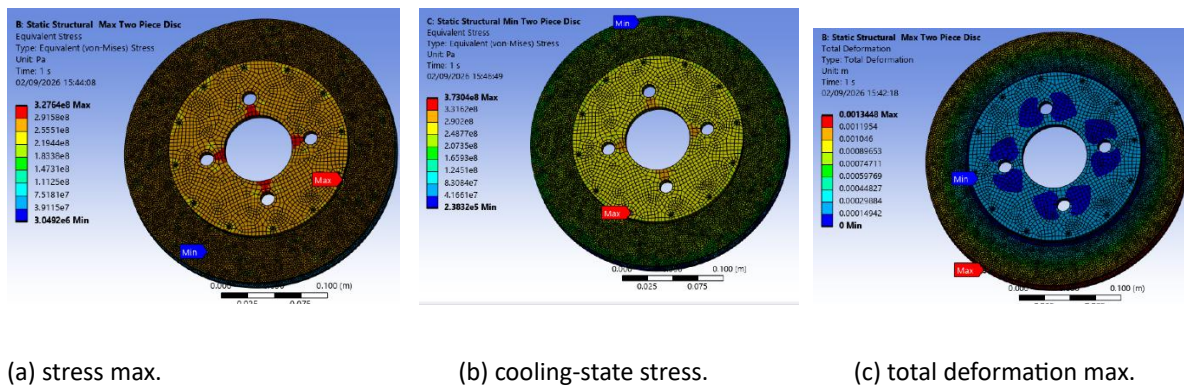


Figure 21: Two-Piece Disc principal-component results: (a) MAX equivalent stress, (b) cooling-state equivalent stress and (c) MAX total deformation. Results scoped to the cast-iron friction ring and aluminium bell.

The complete assembly initially produced a local equivalent-stress peak of approximately **2.29 GPa** within the bolts and nuts. Supervisory review confirmed that such fixing stresses would be expected to differ substantially from the stress level of the main rotor components and that abnormal fixing values could reasonably be discussed separately. Consequently, the 2.29 GPa fastener value was not entered as the representative Design 08 stress in Table 8. The main comparison instead used the cast-iron disc and aluminium bell.

This is an important distinction. The high fixing stress remains useful evidence that the connection deserves more detailed modelling but allowing it to dominate the comparison would provide a misleading assessment of the actual friction ring.

The Two-Piece Disc should therefore not be rejected solely based on the present structural results. Instead, the analysis demonstrates that its behaviour is highly sensitive to contact and fastening assumptions and that a more realistic floating-interface model is required before a definitive structural judgement can be made.

4.11 Comparative Thermal Performance

All seven modified configurations reduced peak temperature relative to the Baseline Disc, but the degree of improvement varied substantially. Dense Drill produced the lowest peak at **568.37°C**, followed closely by Less Groove at **573.66°C** and Dense Groove at **574.74°C**. By contrast, Thin Disc remained close to baseline at **620.58°C**.

The ranking changes considerably when final average temperature is considered. Thin Drill-Groove was the strongest configuration at **284.97°C**, followed by Less Drill at **318.31°C** and Thin Disc at **321.35°C**. Dense Drill and Dense Groove were both around 329°C.

As illustrated in Figure 22, the difference between these rankings demonstrates why repeated-braking performance should not be represented using one temperature value alone. Grzes (2019) distinguished between instantaneous maximum temperature and mean or bulk temperature during repeated braking. A geometry capable of suppressing the highest local peak may not necessarily remove accumulated energy at the greatest rate.

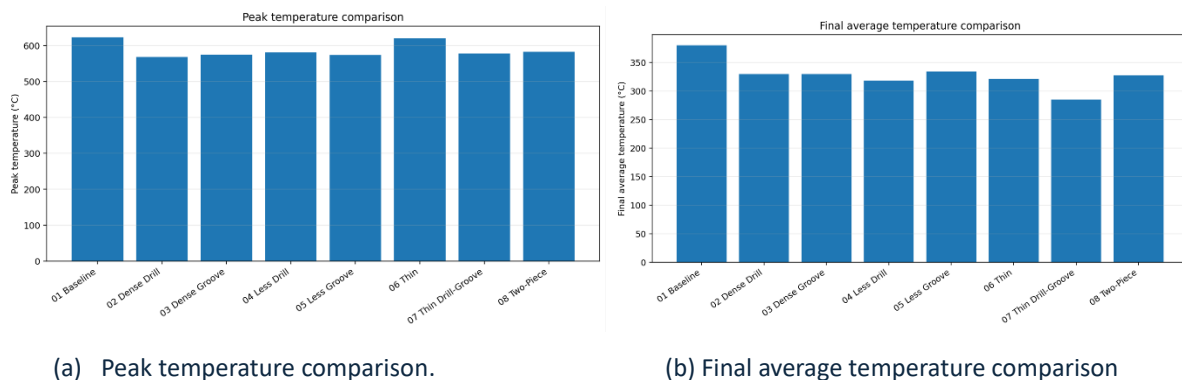


Figure 22: Comparison of thermal performance across the eight brake-disc configurations: (a) peak temperature and (b) final average temperature after five repeated laps.

Table 9 – Percentage change in principal performance measures relative to the Baseline Disc.

Design	Peak Temp. Change	Final Avg. Temp. Change	MAX Stress Change	MAX Def. Change	MAX Strain Change
Baseline	0.00%	0.00%	0.00%	0.00%	0.00%
Dense Drill	−8.83%	−13.33%	+6.14%	+1.68%	+2.69%
Dense Groove	−7.80%	−13.24%	−1.13%	−0.09%	−2.08%
Less Drill	−6.81%	−16.25%	+7.26%	+5.16%	+4.12%
Less Groove	−7.98%	−12.16%	+1.57%	+0.86%	+0.57%
Thin	−0.45%	−15.45%	+7.31%	+8.71%	+3.65%
Thin Drill-Groove	−7.27%	−25.02%	+5.21%	+5.32%	+1.64%
Two-Piece	−6.50%	−13.74%	+28.21%	+79.22%	+192.98%

The percentages show clearly that the greatest thermal improvement does not coincide with the strongest structural result.

4.12 Comparative Structural Performance

The static structural comparison produced a substantially different ranking from the thermal comparison. Dense Groove had the lowest MAX stress at **252.66 MPa**, slightly below the Baseline Disc. Less Groove was also close to baseline at **259.55 MPa**.

Dense Drill, Less Drill, Thin and Thin Drill-Groove all produced higher MAX stress values. This indicates that modifications involving material removal can improve cooling while simultaneously creating local structural penalties. The MAX and cooling-state equivalent-stress results for all eight configurations are compared in Figure 23.

The comparison between Dense Drill and Dense Groove is particularly useful. Dense Drill reduced peak temperature by an additional **6.37°C** compared with Dense Groove, but MAX stress increased by approximately **18.57 MPa**. In other words, a relatively small further reduction in peak temperature was obtained at the expense of a clear increase in structural stress.

This behaviour is consistent with Park *et al.* (2022), who reported that friction-surface features need to be optimised simultaneously for heat dissipation and structural stability. It also supports the results of Pranta *et al.* (2022), who found that modifications to vented rotor geometry can alter both thermal and structural behaviour.

Deformation followed a similar pattern. The Baseline Disc produced 0.75038 mm maximum deformation, while Dense Groove was marginally lower at 0.74969 mm. Thin Disc produced the greatest single-piece deformation at **0.81574 mm**, reflecting reduced stiffness. Two-Piece was an obvious outlier at **1.3448 mm**, largely because its bonded dissimilar-material assembly created a substantially different constraint condition.

The strain results reinforced this interpretation. Designs 01–07 remained close to approximately 0.0021–0.00224, while Design 08 reached approximately 0.0063. This does not prove that physical floating two-piece discs inherently produce such high strain; rather, it highlights the importance of correctly modelling the connection between the ring and bell.

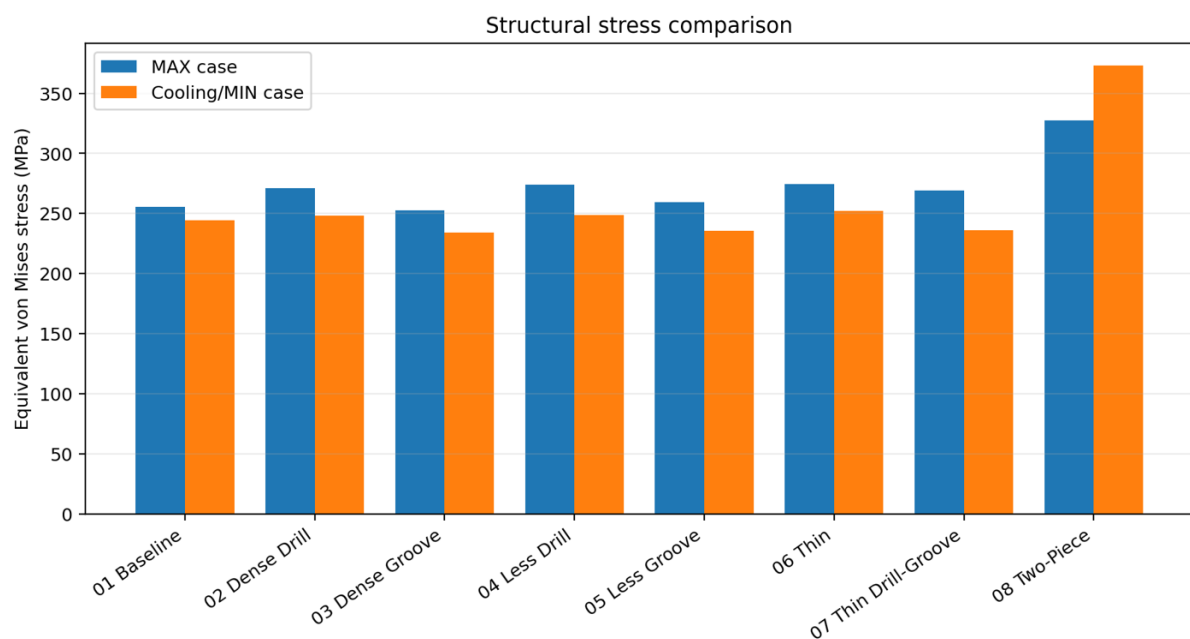


Figure 23: Comparison of equivalent von Mises stress for the representative MAX and cooling-state structural conditions.

4.13 Fatigue-Related Results and Localised FEA Extremes

The fatigue results require greater caution than the thermal and static structural outputs. The Baseline Disc produced the highest minimum safety factor at **0.88236**. Among the modified designs,

Thin Disc had the highest value at **0.78661**, followed by Two-Piece at **0.65709**, Dense Groove at **0.65163** and Less Groove at **0.64164**. Figure 24 shows the relative minimum fatigue safety factors obtained for the eight brake-disc configurations.

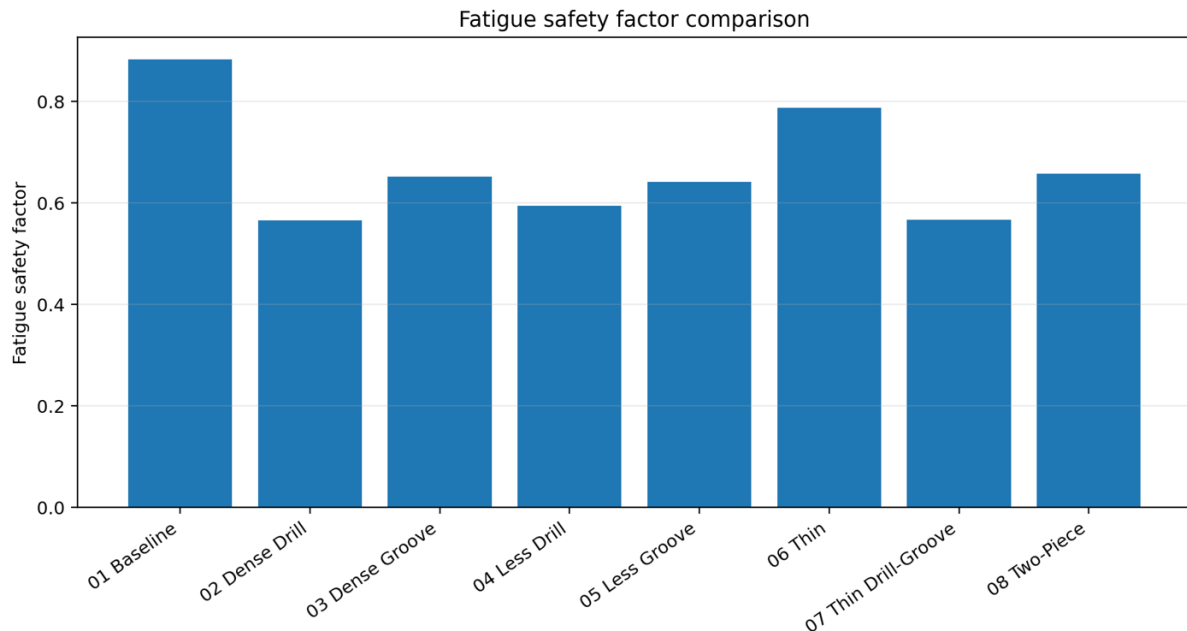
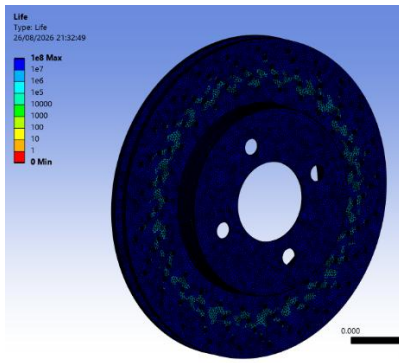


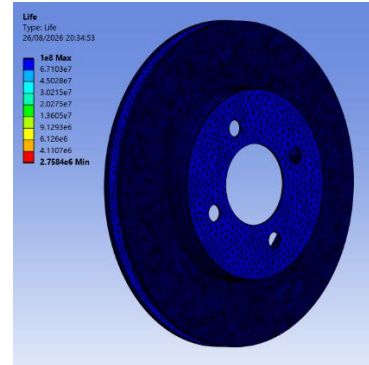
Figure 24: Comparison of minimum fatigue safety factor across the eight brake-disc configurations.

Several geometries produced minimum fatigue-life values approaching zero and extremely large damage values. If these numbers were interpreted without spatial context, they would suggest immediate failure. However, inspection of the fatigue contours showed that the minima occurred in very small local regions, while much of the rotor remained within substantially higher fatigue-life ranges.

This observation was specifically discussed with the project supervisor. The supervisor noted that such extreme values may arise from a very small local mesh region, potentially even one node, particularly around stress concentrations. Consequently, the overall contour is more informative than the scalar minimum when judging whether an isolated value is representative of the component.



(a) Dense Drill fatigue-life contour.



(b) Baseline Disc fatigue-life contour.

Figure 25: Example fatigue-life contour illustrating the distinction between a highly localised minimum and the wider rotor response.

Pevac *et al.* (2014) provide an additional reason for cautious interpretation. Their experimental work on EN-GJL-250 demonstrated substantial deterioration in fatigue performance at elevated temperatures. A simplified Stress-Life calculation based on limited material data cannot therefore reproduce all aspects of high-temperature thermo-mechanical fatigue.

Accordingly, the fatigue results were used as secondary comparative evidence in this study. Local low-life regions remain important because they identify possible areas of design vulnerability, but they were not treated as exact predictions of whole-component service life. This interpretation is more defensible than reporting an isolated numerical minimum as though every part of the rotor experienced the same fatigue condition.

4.14 Overall Thermal–Structural Trade-Off

No investigated geometry was superior in every category. Dense Drill achieved the lowest peak temperature but increased stress and produced one of the lowest fatigue safety factors. Thin Drill-Groove achieved the lowest final average temperature but also increased stress and deformation. Thin Disc improved end-of-cycle cooling but produced almost no reduction in peak temperature and the greatest deformation among the single-piece rotors. Two-Piece reduced temperature but produced substantially greater deformation and strain within the simplified bonded model. The relationship between final average temperature and MAX equivalent stress for the investigated designs is illustrated in Figure 26.

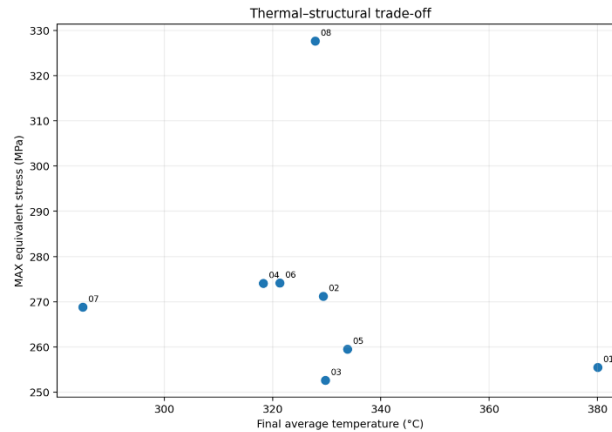


Figure 26: Thermal–structural trade-off between final average temperature and MAX equivalent stress for the eight brake-disc configurations.

Note: 01 Baseline; 02 Dense Drill; 03 Dense Groove; 04 Less Drill; 05 Less Groove; 06 Thin; 07 Thin Drill-Groove; 08 Two-Piece.

The grooved designs produced the strongest overall static thermal–structural balance. Dense Groove reduced peak temperature from **623.39°C to 574.74°C** and final average temperature from **380.05°C to 329.72°C** while slightly reducing MAX stress, deformation and strain. Less Groove also performed well, although its MAX stress was slightly above baseline and its final average temperature was higher than Dense Groove.

Table 10 – Engineering interpretation of the principal performance trade-offs.

Design	Main advantage	Main disadvantage	Overall interpretation
Dense Drill	Lowest peak temperature	Increased stress and low fatigue safety factor	Best peak-temperature control
Dense Groove	Strong thermal reduction with near-baseline or slightly improved structural response	Fatigue result below baseline	Best overall thermal–structural compromise
Less Drill	Good reduction in accumulated temperature	Increased stress and deformation	Thermal improvement with structural penalty
Less Groove	Strong peak reduction and near-baseline structural response	Slightly higher final average than Dense Groove	Strong balanced alternative

Thin	Good cooling at end of cycle; potential mass reduction	Almost no peak reduction; high deformation	Mainly attractive for lightweighting
Thin Drill- Groove	Lowest final average temperature	Higher stress/deformation and low safety factor	Best accumulated cooling
Two- Piece	Thermal improvement and lightweight bell concept	High strain/deformation; interface sensitive	Requires improved floating-joint model

Within the assumptions and boundary conditions of the present finite element model, the Dense Groove Disc demonstrated the most favourable overall thermal–structural compromise among the investigated modified single-piece designs.

This conclusion needs to remain carefully qualified. It does not mean that Dense Groove has been experimentally proven to provide the longest physical service life. The Baseline Disc retained the highest fatigue safety factor, and the Dense Groove fatigue output was not superior to baseline. Experimental testing and temperature-dependent fatigue modelling would therefore be required before making a production or safety-critical design recommendation.

The present conclusion is instead a design-screening judgement: Dense Groove provided the strongest combination of meaningful thermal reduction and acceptable static structural response within the common numerical framework.

4.15 Comparison with Published Research

The direction of the principal results is broadly consistent with published research. The temperature reductions achieved by the drilled and grooved designs support the findings of Park *et al.* (2022), Pranta *et al.* (2022) and Tauviquirrahman *et al.* (2023), who demonstrated that surface and ventilation modifications can change brake-disc thermal performance.

The increased stresses in the drilled configurations also agree with the structural concern identified by Park *et al.* (2022), where holes were shown to create potential stress-concentration regions. The present Dense Drill result is particularly consistent with this trend: it achieved the greatest peak-temperature reduction but simultaneously produced a 6.14% increase in MAX stress.

The Thin Disc result agrees qualitatively with studies examining thickness and lightweighting. Huang *et al.* (2022) showed that rotor plate thickness and its distribution can influence thermally induced distortion. In the present study, thinning reduced accumulated temperature but increased deformation and had almost no effect on the overall peak. This reinforces the view that lightweighting must be evaluated together with structural stiffness and thermal capacity.

Repeated thermal behaviour also corresponded with Grzes (2019), who showed that residual heat affects later braking events and that bulk and maximum temperature should be considered separately. The different ranking produced by peak and final average temperature in the present research directly supports this distinction.

The Two-Piece results agree qualitatively with Mîndru *et al.* (2026), who demonstrated the importance of connection arrangement in floating-disc behaviour. The unusually high strain, deformation and local fixing stress obtained in the bonded model therefore reinforce the need for improved interface modelling rather than demonstrating that two-piece brake discs are inherently structurally inferior.

Direct numerical agreement with published studies was neither expected nor required because vehicle mass, braking histories, rotor dimensions, materials and cooling coefficients differ between investigations. Agreement in the direction of the observed trends provides a more meaningful form of qualitative validation.

4.16 Reliability and Limitations of the Results

The main strength of the results is the consistency of the comparative procedure. The same braking-energy method, heat-flux history, cooling approach, structural pressure and principal output definitions were applied across all eight geometries. This provides a reasonable basis for comparing the relative influence of design changes.

Nevertheless, the absolute results remain sensitive to several assumptions. Frictional heating was represented using a distributed surface heat flux rather than explicit moving pad contact. Convection was prescribed as a function of vehicle speed rather than determined individually for each geometry using CFD. The structural mounting face was idealised as fixed, while pad loading was represented using uniform pressure.

Material representation also introduces uncertainty. EN-GJL-250 behaviour varies substantially with temperature, and the simplified material model does not reproduce every high-temperature elastic and fatigue property. Equivalent von Mises stress was used for consistency, but grey cast iron is brittle, meaning that maximum principal stress or another brittle-material criterion would be more appropriate for a detailed fracture assessment.

Mesh sensitivity is particularly important for drilled holes, groove edges and fixing regions. A sharp idealised geometric feature may create a very high local stress over a very small mesh region. The final supervisor discussion confirmed that such values should be evaluated using the complete contour rather than accepted automatically as representative whole-component stresses or fatigue lives. This principle was applied to both the highly local fatigue minima and the approximately 2.29 GPa fixing stress in the Two-Piece model.

The results should therefore be considered strongest as **relative design evidence**, while exact temperatures, stresses and fatigue lives would require additional experimental and numerical validation.

4.17 Chapter Summary

The results demonstrate that brake-disc geometry significantly influenced both thermal and structural behaviour under the repeated five-lap braking condition. All seven modified geometries reduced final average temperature relative to the baseline, although the extent of improvement and associated structural consequences varied considerably.

Dense Drill achieved the lowest peak temperature of **568.37°C**, while Thin Drill-Groove achieved the lowest final average temperature of **284.97°C**. Neither configuration provided the strongest overall structural response. Dense Groove produced a slightly higher peak temperature of **574.74°C**, but it also reduced MAX equivalent stress, deformation and strain relative to baseline. Consequently, within the assumptions of the present model, Dense Groove provided the most favourable thermal–structural compromise among the modified single-piece configurations.

The Two-Piece Disc showed useful thermal improvement but demonstrated high sensitivity to the bonded connection between the cast-iron ring and aluminium bell. Its results indicate that more realistic floating-interface and fastener modelling is required before its structural performance can be assessed confidently.

Finally, the fatigue analysis demonstrated the importance of examining spatial FEA contours rather than relying on one reported minimum. Highly local values were retained and discussed but were not interpreted as complete rotor service-life predictions. These findings provide the basis for the conclusions and recommendations presented in Chapter 5.

Chapter 5 – Conclusions and Recommendations

5.1 Introduction

This dissertation investigated how changes to ventilated brake-disc geometry influenced thermal and structural behaviour under repeated motorsport braking conditions. Eight configurations were analysed using a common finite element procedure based on MoTeC telemetry, transient thermal analysis and sequential structural assessment. The investigation demonstrated that geometry had a measurable influence on peak temperature, accumulated temperature, stress, strain and deformation. However, the design producing the greatest thermal improvement did not necessarily provide the most favourable structural response. This chapter summarises the principal findings, evaluates achievement of the research objectives, answers the research questions and identifies limitations and recommendations for further work.

5.2 Achievement of the Research Objectives

The first objective was to critically review research on brake-disc heat generation, repeated braking, ventilation, surface modification, lightweighting, thermo-mechanical behaviour and fatigue. The literature showed that thermal improvement should not be considered independently of structural performance, as drilled and grooved features may improve cooling while also creating stress concentrations (Park *et al.*, 2022).

The second objective was to develop a baseline ventilated brake disc and seven modified configurations. All eight geometries were successfully created and analysed: Baseline, Dense Drill, Dense Groove, Less Drill, Less Groove, Thin, Thin Drill-Groove and Two-Piece.

The third objective was to derive a repeated-braking thermal load from MoTeC telemetry. Six braking events were identified and repeated over five laps, producing a total transient simulation time of **459.155 s**. Changes in vehicle kinetic energy were converted into event-specific heat-flux inputs.

The fourth objective was achieved by conducting transient thermal analyses for all eight designs under consistent boundary conditions. The fifth objective involved transferring representative MAX and cooling temperature fields into static structural analyses to evaluate equivalent stress, elastic strain and total deformation.

The sixth objective was to assess fatigue-related behaviour using life, damage and safety factor. These results were interpreted cautiously because some minimum-life values were highly localised.

The seventh objective was to identify the most favourable overall design. Dense Drill achieved the lowest peak temperature, Thin Drill-Groove produced the lowest final average temperature, while Dense Groove provided the most favourable overall thermal–structural compromise among the modified single-piece designs.

5.3 Answers to the Research Questions

5.3.1 How did geometry affect temperature under repeated braking?

Geometry influenced both the severe local temperature peaks and the amount of heat retained after five laps. The Baseline Disc reached a maximum temperature of **623.39°C** and a final average temperature of **380.05°C**. Every modified geometry reduced final average temperature, confirming that changes in exposed area, thermal mass and heat-transfer paths altered the repeated-braking thermal response.

The different rankings of peak and final average temperature also confirmed that these parameters describe different physical effects. This supports the distinction made by Grzes (2019) between maximum and accumulated thermal behaviour during repeated braking.

5.3.2 Which geometry achieved the greatest thermal improvement?

Dense Drill produced the lowest overall peak temperature at **568.37°C**, representing an **8.83% reduction** relative to baseline.

Thin Drill-Groove produced the lowest final average temperature at **284.97°C**, a reduction of approximately **25.02%**.

There was therefore no single thermal winner. Dense Drill controlled the most severe local temperature most effectively, whereas Thin Drill-Groove provided the greatest reduction in accumulated end-of-cycle temperature.

5.3.3 How did the modifications affect stress, strain and deformation?

The structural results showed clear differences between the design families. Dense Drill, Less Drill, Thin and Thin Drill-Groove all increased MAX equivalent stress compared with baseline. By contrast,

Dense Groove reduced MAX stress from **255.54 MPa to 252.66 MPa**, while also producing slightly lower deformation and strain.

The Two-Piece Disc showed the largest structural response, with MAX deformation of **1.3448 mm** and considerably greater strain than the single-piece discs. These values were strongly influenced by the bonded representation of the cast-iron ring and aluminium bell. A real floating rotor permits relative movement between the components, making connection modelling an important consideration (Mîndru *et al.*, 2026).

5.3.4 What thermal–structural trade-offs were identified?

The principal trade-off was between cooling improvement and structural continuity. Drilled holes reduced temperature but also removed load-bearing material and introduced local stress concentrations. Thin configurations improved end-of-cycle cooling but generally increased deformation because of reduced stiffness.

The grooved configurations produced a more balanced response. They improved thermal behaviour while maintaining a larger proportion of the continuous friction-ring structure. This agrees with the general conclusion of Park *et al.* (2022) that brake-disc surface modifications need to be assessed through combined thermal and structural objectives rather than cooling alone.

5.3.5 Which design provided the most favourable overall performance?

Within the assumptions of the present FEA model, **Dense Groove was the preferred modified single-piece configuration**. It reduced peak temperature by approximately **7.80%** and final average temperature by approximately **13.24%**, while reducing MAX equivalent stress by approximately **1.13%**, MAX deformation slightly and MAX strain by approximately **2.08%**.

This conclusion should not be interpreted as proof that Dense Groove would provide the longest physical service life. The Baseline Disc retained the highest fatigue safety factor, and experimental validation would be required before making a production recommendation.

Table 11 – Summary of Research Questions and Findings

Research question	Principal finding
Effect of geometry on temperature	All modifications altered repeated-braking temperature and reduced final average temperature
Best peak-temperature design	Dense Drill: 568.37°C

Best accumulated-cooling design	Thin Drill-Groove: 284.97°C final average
Best static structural response among modified designs	Dense Groove
Main design trade-off	Greater cooling can increase stress, strain or deformation
Preferred overall modified design	Dense Groove, within the assumptions of the model

5.4 Principal Engineering Contributions

The first contribution of the study is the use of measured motorsport telemetry to define a repeated thermal loading history. Rather than applying a single arbitrary braking event, six braking events were represented over five repeated laps. This provided a closer connection between the numerical model and vehicle operating behaviour.

The second contribution is the controlled comparison of several different design strategies within one modelling framework. Because the same fundamental loading and boundary conditions were used, differences between the models can be interpreted primarily as consequences of geometric modification.

A further contribution concerns interpretation of numerical results. The project demonstrated that a single reported maximum or minimum from FEA can be misleading when it occurs within a very small local mesh region. This was particularly evident in the fatigue analyses and in the Two-Piece Disc fasteners. The supervisor's final review confirmed that isolated numerical peaks should be considered alongside the wider contour distribution rather than automatically treated as representative of the whole component.

5.5 Limitations of the Study

Several limitations should be recognised when interpreting the results. Heat generation was represented using a distributed heat flux rather than explicit rotating pad-disc contact. Cooling was approximated through speed-dependent convection rather than geometry-specific CFD, and material properties were simplified compared with their full temperature dependence.

The structural analysis used a fixed mounting region and uniform pad-pressure representation. Equivalent von Mises stress provided a consistent comparison across the designs, although grey cast

iron is brittle and maximum principal stress or another brittle-material criterion would provide a more appropriate basis for detailed fracture assessment.

Fatigue predictions were also limited by localised stress concentrations and the available material data. Pevec *et al.* (2014) demonstrated that EN-GJL-250 fatigue capability changes substantially at elevated temperature. Consequently, the calculated fatigue-life values should be regarded as comparative indicators rather than exact service-life predictions.

The Two-Piece Disc contained an additional limitation because bonded contacts prevented the relative movement expected from a true floating rotor. The structural results for this design are therefore particularly sensitive to the connection assumptions.

5.6 Recommendations for Future Work

Future work should first concentrate on experimental validation. Temperature histories for the Baseline and Dense Groove configurations could be measured through brake-dynamometer testing or vehicle testing using thermocouples or infrared thermography.

A second priority should be the use of CFD to calculate geometry-specific airflow and heat-transfer coefficients for the ventilation passages, drilled holes and grooves. This would improve the representation of cooling compared with the prescribed convection model used in the present study.

Temperature-dependent thermal, elastic and fatigue properties should also be introduced for EN-GJL-250, particularly above 500°C. Structural assessment could be extended to include maximum principal stress and an appropriate brittle-material failure criterion.

The Two-Piece Disc should be remodelled using a realistic floating interface incorporating contact friction, clearances and bolt pretension. Finally, a formal mesh-convergence study should be undertaken for the Baseline Disc, Dense Groove design and critical local stress regions.

Further optimisation of the Dense Groove configuration could then investigate groove depth, width, number, spacing and orientation rather than considering only the two feature densities used in the current study.

5.7 Final Conclusion

The investigation confirms that ventilated brake-disc geometry has a significant effect on the thermal and structural response of a rotor subjected to repeated braking. Importantly, the configuration with the lowest peak temperature was not the configuration with the lowest accumulated temperature, and neither provided the strongest overall structural balance.

Dense Drill achieved the greatest reduction in maximum temperature, while Thin Drill-Groove achieved the greatest reduction in final average temperature. However, both designs introduced structural penalties. Dense Groove provided a more balanced response by substantially reducing temperature while maintaining stress, strain and deformation at approximately the same level as, or slightly below, the Baseline Disc.

The Two-Piece concept also achieved useful thermal improvement but showed that numerical representation of the connection between dissimilar materials is critical. Its structural performance should therefore be reassessed using a realistic floating-interface model.

Overall, the study demonstrates that brake-disc optimisation should not be based on a single ANSYS result. Temperature history, stress distribution, deformation, strain, fatigue behaviour and the limitations of the modelling assumptions need to be considered together. Within this comparative framework, the project achieved its aim and identified **Dense Groove as the most favourable modified single-piece design for further development and experimental validation.**

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Appendix

Appendix A – MoTeC Telemetry Evidence

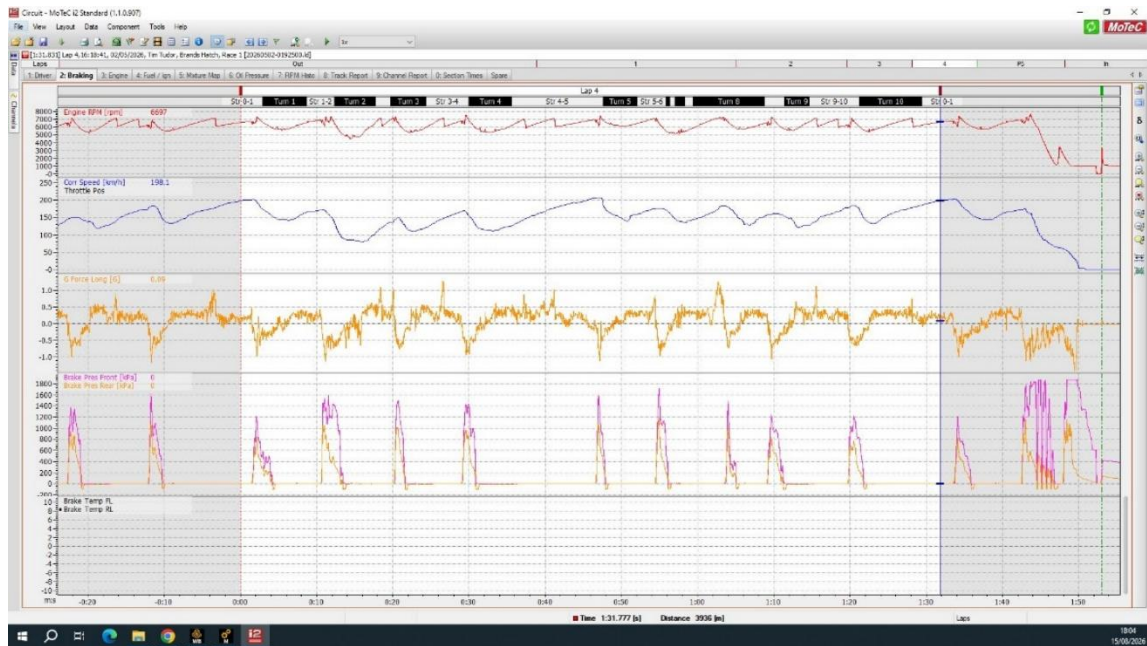


Figure 27: Full-lap MoTeC telemetry for Brands Hatch Lap 4 showing vehicle speed and repeated front and rear brake-pressure events.

Circuit - MoTeC i2 Standard (1.1.0.907)

File View Layout Data Component Tools Help

1:11.831 Lap 4: 18-18-41, 02/05/2026, Tim Tudor, Brands Hatch, Race 1 [20260902-918260348].out

Lap 4

1: Driver 2: Braking 3: Engine 4: Fuel / Ign 5: Mature Map 6: Oil Pressure 7: RPM Map

Brands Hatch, Brands Hatch 26
Tim Tudor, MCL_yellow
02/05/2026, 6
18-18-41, Race 1

	Lap 1	Lap 2	Lap 3	Lap 4	Electric	Rolling Minimum
Sr 0-1 (Snd)	2:04.753	1:05.265	0:02.929	0:02.945	0:02.929	0:02.929
Turn 1	0:09.112	0:06.488	0:05.863	0:05.790	0:05.790	0:05.882
Sr 1-2	0:06.027	0:03.975	0:03.584	0:03.629	0:03.584	0:03.584
Turn 2	0:06.475	0:05.707	0:05.447	0:05.448	0:05.447	0:05.447
Sr 2-3	0:03.340	0:01.809	0:01.899	0:01.888	0:01.899	0:01.899
Turn 3	0:06.637	0:04.887	0:04.698	0:04.763	0:04.693	0:04.693
Sr 3-4	0:08.947	0:05.495	0:05.532	0:05.517	0:05.495	0:05.532
Turn 4	0:07.632	0:05.861	0:05.884	0:05.719	0:05.719	0:05.884
Sr 4-5	0:19.796	0:12.010	0:12.009	0:11.862	0:11.862	0:12.009
Turn 5	0:06.290	0:04.794	0:04.943	0:04.580	0:04.413	0:04.413
Sr 5-6	0:05.521	0:03.780	0:03.678	0:03.694	0:03.678	0:03.678
Turn 6	0:00.966	0:00.606	0:00.538	0:00.564	0:00.539	0:00.539
Sr 6-7	0:00.907	0:00.693	0:00.580	0:00.610	0:00.580	0:00.580
Turn 7	0:01.618	0:01.423	0:01.436	0:01.348	0:01.348	0:01.423
Sr 7-8	0:01.297	0:00.578	0:01.014	0:00.596	0:00.578	0:00.578
Turn 8	0:14.356	0:08.856	0:09.576	0:09.457	0:08.856	0:08.856
Sr 8-9	0:04.935	0:02.494	0:02.525	0:02.505	0:02.495	0:02.496
Turn 9	0:04.818	0:01.626	0:01.434	0:01.322	0:01.322	0:01.626
Sr 9-10	0:13.010	0:06.365	0:06.475	0:06.461	0:06.365	0:06.365
Turn 10	0:22.857	0:08.309	0:08.336	0:08.248	0:08.248	0:08.309
Sr 0-1 (Start)	0:11.752	0:02.296	0:02.378	0:02.375	0:02.375	0:02.396
Totals	4:40.055	2:35.945	1:30.576	1:31.831	1:30.662	1:31.755

Figure 28: MoTeC Section Times confirming a Brands Hatch Lap 4 duration of 91.831 s.

Appendix B – Additional Thermal Setup Evidence

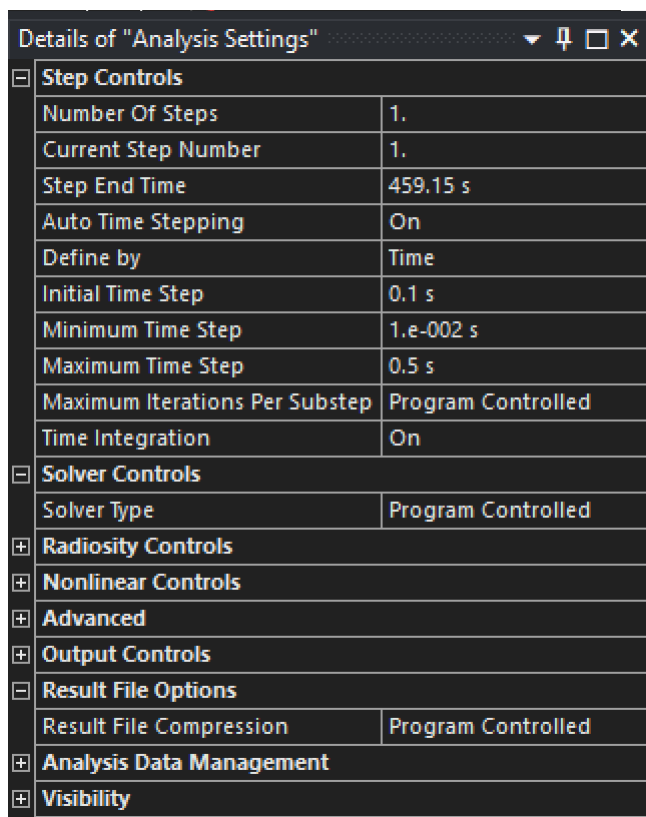


Figure 29: ANSYS transient thermal Analysis Settings used for the five-lap simulation.

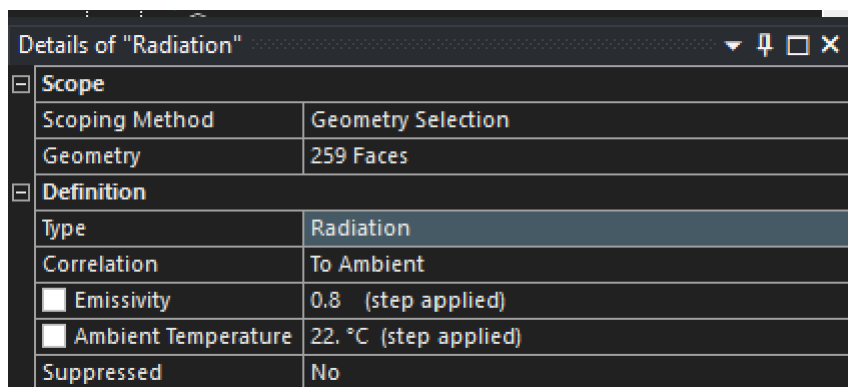


Figure 30: ANSYS thermal-radiation boundary condition showing the adopted emissivity and ambient-temperature settings.

Appendix C – Critical Numerical Evidence

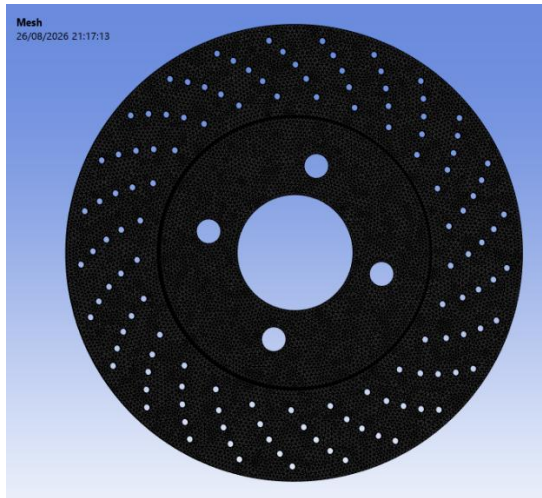


Figure 31: Finite element mesh refinement around the drilled features of the Dense Drill Disc.

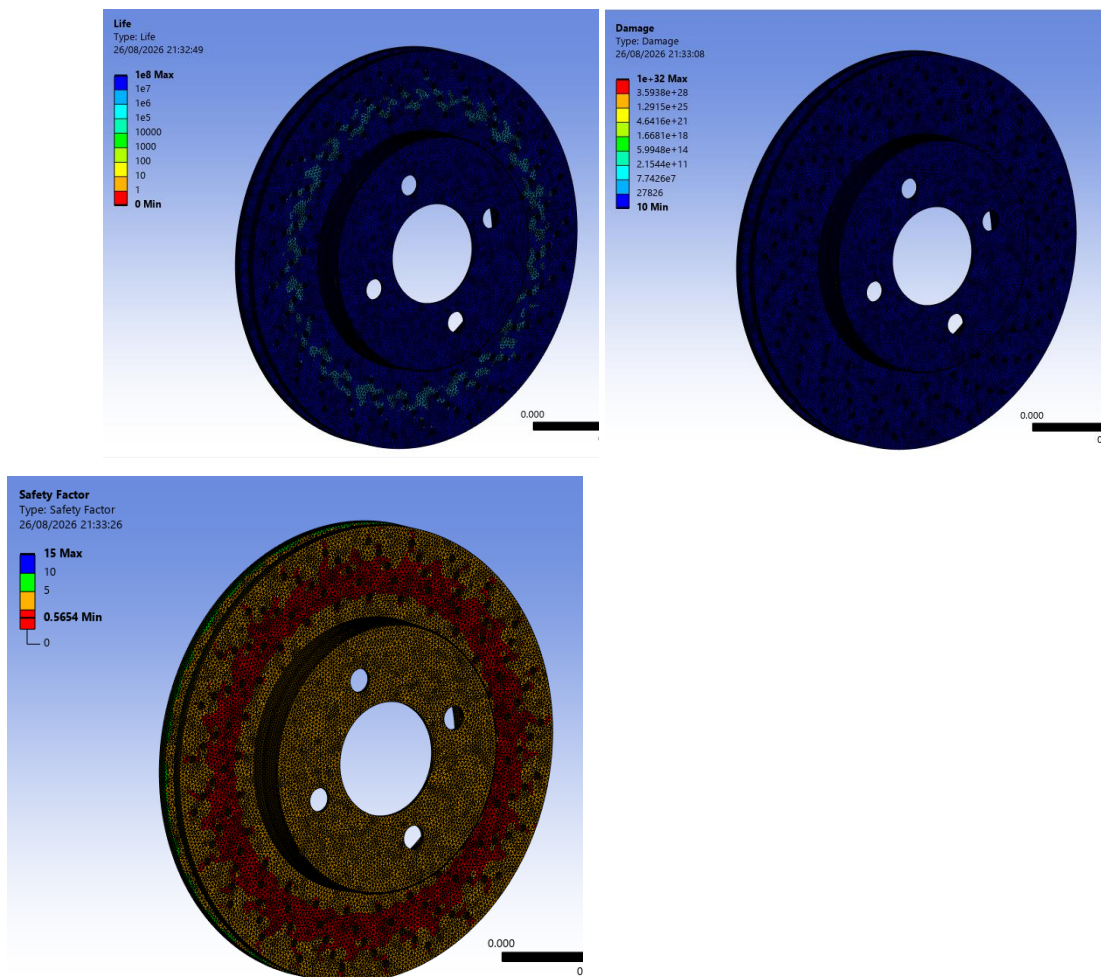


Figure 32: Dense Groove Disc fatigue assessment showing (a) Life, (b) Damage and (c) Safety Factor.

Appendix D – Two-Piece Disc Additional Evidence

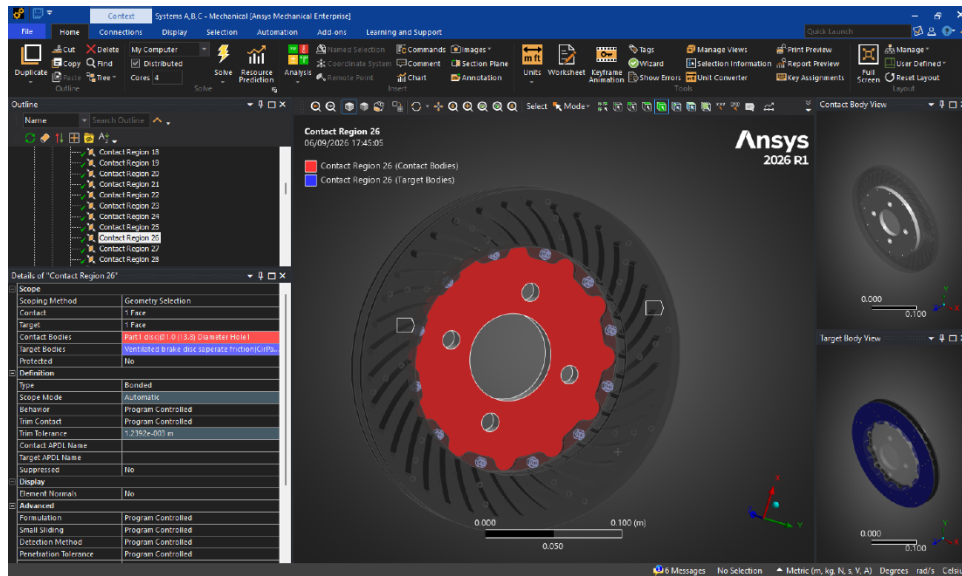


Figure 33: Bonded contact definition used between components of the Two-Piece Disc model.

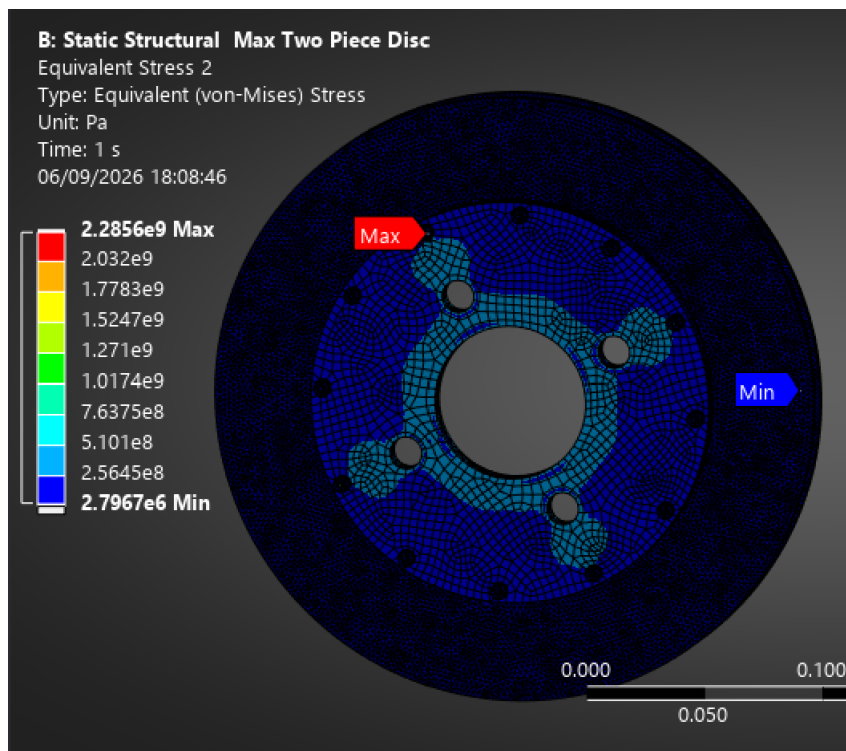


Figure 34: Localised equivalent stress within the Two-Piece Disc fixing system. The fixing stress was not used as the representative rotor stress in the principal design comparison.