



Thermal and Structural Analysis of Automotive Disc Brakes Using Finite Element Method

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Abstract

Automotive disc brakes convert a major portion of vehicle kinetic energy into frictional heat within a short braking interval. The resulting non-uniform temperature field produces thermal expansion, rotor coning, local stress concentration and, under severe or repeated braking, conditions favorable to fade, hot spotting and crack initiation. This methodology paper presents a compact finite element procedure for evaluating the coupled thermal and structural response of a ventilated grey-cast-iron brake rotor during a single severe braking event. The procedure begins with calculation of braking energy and interface heat flux, followed by three-dimensional meshing, transient thermal analysis with convection on exposed surfaces, transfer of the nodal temperature field to a structural model, and evaluation of equivalent stress and total deformation. Mesh and time-step convergence are included as mandatory checks. The proposed framework is supported by pre-2017 literature on thermoelastic disc-brake modelling and uses published results as validation benchmarks rather than as replacement experimental data. The expected response is a peak temperature near the swept friction track late in the stop, followed by cooling, while thermal gradients generate the highest structural demand near geometric transitions, ventilation features and constrained hub regions. The method is suitable for comparing rotor geometry, material properties and cooling assumptions before prototype testing.

Introduction

Disc brakes are safety-critical vehicle components because their ability to generate stable braking torque must be maintained while substantial mechanical energy is converted into heat. During braking, the pads press against the rotating disc and frictional work raises the temperature of the contact track. The rotor then stores part of this energy and rejects heat by conduction into the hub and by convection and radiation from exposed surfaces. Since the heat source is localized and moves relative to the disc, the temperature field is highly transient and may also become circumferentially non-uniform. The associated thermal expansion is restrained by

cooler material and by the connection of the disc to the wheel hub, generating thermal stress and out-of-plane distortion. These mechanisms are directly related to practical problems such as coning, judder, disc thickness variation, thermal fatigue and cracking.

Finite element analysis is widely used to study these effects because it can represent realistic rotor geometry, non-uniform heat input, temperature-dependent boundary conditions and structural constraints more effectively than a purely closed-form model. Valvano and Lee (2000) combined thermal calculations with finite-element thermal stress analysis to predict

rotor distortion, while Cho and Ahn (2002) developed a transient thermoelastic finite-element formulation for a rotating disc subjected to non-axisymmetric heating. Choi and Lee (2004) further demonstrated that frictional heating, contact pressure and thermoelastic instability must be considered together when repeated braking produces non-uniform temperature and pressure fields. Such studies establish that thermal and structural analyses should not be treated as unrelated calculations.

For engineering design, a sequentially coupled approach is often attractive: the transient thermal problem is solved first, and the resulting temperature field is imported as a body load for structural analysis. Apte and Ravi (2006) showed the usefulness of finite-element prediction of disc temperatures and stresses for design validation, and Gao et al. (2007) used thermomechanical coupling to examine thermal-fatigue-related stress development. The present paper therefore formulates a step-by-step methodology for a passenger-car ventilated rotor under a severe single stop. The objective is not to claim a new experimentally validated brake design, but to provide a reproducible analysis workflow, define the principal governing equations and boundary conditions, and establish literature-based checks for temperature, deformation and stress trends.

Literature Review

Valvano and Lee (2000) proposed an analytical and finite-element procedure for estimating the thermal distortion of brake rotors under transient and steady operating conditions. Their work emphasized that heat input and cooling characteristics should be calculated consistently before thermal stress is evaluated. Cho and Ahn (2002) addressed the three-dimensional transient thermoelastic problem by combining finite-element analysis with a fast Fourier transform technique. They showed the importance of representing the rotating disc and the spatially localized pad heating rather than assuming a perfectly uniform temperature field.

Choi and Lee (2004) investigated transient thermoelastic behavior in disc brakes with frictional heat generation. Their coupled solution linked temperature and pressure distributions and examined thermoelastic instability, demonstrating that local pressure and temperature can reinforce one another. Apte and Ravi (2006) focused on an engineering workflow for predicting thermal performance and stresses of solid and ventilated brake discs, including heat-transfer coefficient estimation and correlation against existing test data. Gao et al.

(2007) used a three-dimensional thermostructural model to investigate transient stresses caused by a moving heat source and reported that severe braking can produce cyclic surface stress fluctuations and residual tensile stress during cooling.

Yildiz and Duzgun (2010) compared solid and ventilated brake-disc configurations using finite-element stress analysis. Their experiments and simulations showed that ventilation features can create local stress concentrations, while a more realistic variation of pad loading can reduce predicted stresses. Adamowicz and Grzes (2011) compared two- and three-dimensional finite-element models for a single braking event and showed that a three-dimensional non-axisymmetric heat load is useful when circumferential temperature variation is important. Yevtushenko and Grzes (2011) examined heat partition in a pad/disc system and demonstrated that the selected heat-partition model strongly affects pad temperature and therefore must be defined explicitly in thermal simulations.

Jung et al. (2011) developed a fully coupled three-dimensional model of a ventilated disc, pads and pistons to study hot judder and compared numerical outputs with dynamometer observations. Belhocine and Bouchetara (2012a) analyzed a solid brake disc using ANSYS and emphasized the influence of braking mode, rotor design and material on temperature. In a related thermomechanical study, Belhocine and Bouchetara (2012b) used sequential thermal-structural coupling to obtain stress and deformation fields, while Kang and Cho (2012) reported lower maximum temperature for a ventilated disc than a solid disc and quantified thermal deformation in the ventilated geometry. Belhocine and Bouchetara (2014) integrated thermal and structural analysis of an automotive rotor and reported an umbrella-like distortion pattern caused by non-uniform heating. Belhocine, Abu Bakar and Bouchetara (2015) later examined a single-stop event at assembly level, reinforcing the applicability of sequential thermal and structural analysis for practical brake design.

Methodology

1. Analysis scope and assumptions

A ventilated passenger-car front brake rotor is considered. The rotor is treated as homogeneous grey cast iron and the pad is represented through an applied heat-flux region rather than as a fully deformable friction body in the baseline model. The analysis assumes dry friction, constant

nominal friction coefficient, uniform initial temperature, no phase change, and small structural deformation. Radiation may be included, but for a compact single-stop methodology the dominant cooling mechanism is represented by convection on exposed surfaces. The sequential coupling assumption means that the temperature field affects structural strain and stress, whereas structural deformation is not fed back into the thermal contact calculation during the same time step. This is appropriate for an initial design study and follows widely used brake-rotor procedures reported by Apte and Ravi (2006) and Belhocine and Bouchetara (2012b, 2014).

2. Geometry, material and braking inputs

A representative rotor geometry is defined by an outer diameter of 280 mm, nominal thickness of

26 mm, swept inner diameter of approximately 150 mm and a ventilated annular friction ring joined to a central hat. Illustrative constant grey-cast-iron properties are density 7100 kg/m³, thermal conductivity 52 W/m·K, specific heat 460 J/kg·K, Young's modulus 110 GPa, Poisson's ratio 0.28 and coefficient of thermal expansion $10.5 \times 10^{-6} \text{ K}^{-1}$. These values are placeholders for a methodology study; a production analysis should replace them with temperature-dependent supplier data. A vehicle mass of 1300 kg, initial speed of 100 km/h, constant deceleration of 8 m/s², front-axle brake share of 70%, ambient temperature of 25 °C and nominal pad friction coefficient of 0.35 are used to define the loading case.

Table 1. Baseline inputs for the proposed FEM case.

Parameter	Baseline value	Modelling note
Vehicle mass, m	1300 kg	Representative passenger car
Initial speed, v_0	100 km/h	Single severe stop
Deceleration, a	8 m/s ²	Constant for baseline case
Braking time, t_b	3.47 s	$t_b = v_0/a$
Front brake share	0.70	Split equally between front rotors
Initial / ambient T	25 °C	Uniform initial condition
Rotor material	Grey cast iron	Temperature-dependent data preferred
Local mesh size	2–3 mm	Friction track and vent roots

3. Braking energy and thermal loading

The kinetic energy removed from the vehicle is first calculated from vehicle dynamics. If aerodynamic and rolling losses are neglected during the short stop, the total energy to be dissipated by the brakes is approximately one-half of $m v_0^2$. The energy assigned to one front rotor is obtained by multiplying by the front brake share and dividing by two. For constant deceleration, vehicle speed varies as $v(t) = v_0 - at$, so the instantaneous friction power decreases with time. A heat-partition factor, η , assigns the selected fraction of interface heat to the rotor; this factor should be obtained from thermal effusivity or a validated partition model because Yevtushenko and Grzes (2011) showed its strong influence on predicted interface temperature.

$$E_{\text{rotor}} = 0.5 m v_0^2 \times \beta_f / 2$$

$$q(t) = \eta F_b v(t) / A_c, \quad v(t) = v_0 - at$$

The resulting heat flux is applied to the swept friction surfaces over the active pad region. The flux may be defined as a moving source for a detailed non-axisymmetric model or circumferentially averaged for a faster design model. Adamowicz and Grzes (2011) showed

that the three-dimensional moving/non-axisymmetric representation is preferable when local circumferential peaks are required. Exposed rotor surfaces receive convection, $q_{\text{conv}} = h(T - T_\infty)$, with separate coefficients permitted for friction faces, outer rim, hat and internal ventilation passages. A CFD-derived or test-correlated heat-transfer coefficient is preferred for final design validation.

4. Finite element model and mesh

The three-dimensional CAD model is imported into the finite-element environment and discretized with second-order tetrahedral or hexahedral thermal-solid elements where practical. Mesh refinement is concentrated at the swept track, vent roots, changes in section thickness and the hat-to-ring transition because these locations exhibit the steepest thermal gradients and structural stress concentration. The inner mounting region is constrained in a manner that reproduces wheel-hub attachment while avoiding unnecessary over-constraint. At least three successively refined meshes are recommended. Convergence is accepted when further refinement produces only a small change

in peak rotor temperature and equivalent stress while preserving the same hot-spot location.

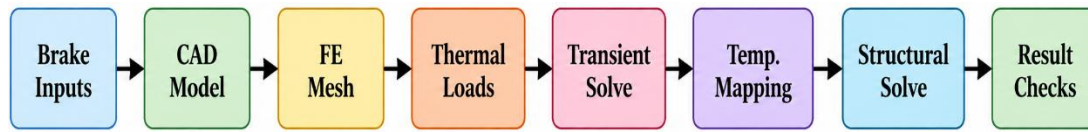


Figure 1. Sequential FEM workflow used for thermal and structural brake-rotor analysis.

5. Transient thermal analysis

The transient temperature field is governed by conservation of energy. In matrix finite-element form it may be written as $[C]\{\dot{T}\} + [K]\{T\} = \{Q\}$, where $[C]$ is the heat-capacity matrix, $[K]$ is the conductivity matrix and $\{Q\}$ contains frictional heat and boundary heat-flow terms. The initial temperature is set to 25 °C. The braking phase is solved with a sufficiently small time step to capture the rapidly changing power input, followed by a cooling phase with zero friction heat and continued convection. Peak surface temperature, radial and through-thickness gradients, and cooling rate are extracted. Belhocine and Bouchetara (2012a) and Kang and Cho (2012) demonstrate why rotor ventilation and realistic heat-transfer conditions are essential for meaningful comparison.

6. Structural analysis and thermal coupling

The nodal temperature field from selected thermal time steps is mapped to the structural mesh. Thermal strain is calculated as $\epsilon_{th} = \alpha(T - T_{ref})$, and the structural solution combines this strain with mechanical restraint from the hub and, if modelled, pad pressure. The primary outputs are total deformation, axial coning displacement and von Mises equivalent stress. The stress field is inspected at the hat transition, ventilation features and friction ring, where large thermal gradients can create tensile and compressive zones. Gao et al. (2007) showed that thermally driven stress cycles are relevant to fatigue, while Jung et al. (2011) linked non-uniform heating and distortion to hot-judder behavior.

$$\epsilon_{th} = \alpha (T - T_{ref})$$

7. Verification and validation strategy

Verification is performed by mesh refinement, time-step refinement and an energy balance between imposed friction energy and the sum of stored and dissipated thermal energy. Validation should compare surface temperature histories and rotor distortion against thermocouple, infrared or dynamometer data when available. In

the absence of project-specific test data, published trends are used only as benchmarks. The simulation should reproduce the established qualitative behavior: non-uniform heating near the friction track, lower temperature for effective ventilation, thermal coning caused by through-thickness gradients, and stress concentration near geometric discontinuities. A model that produces smooth but physically implausible temperature or stress patterns should not be accepted merely because the numerical solver converges.

Results and Discussion

The proposed methodology is expected to produce a rotor temperature that rises rapidly during the braking interval and reaches its maximum near the end of the stop or shortly thereafter, depending on thermal inertia and heat-transfer assumptions. The friction track should be hotter than the hub and internal material, establishing radial and through-thickness temperature gradients. A ventilated rotor should generally show lower sustained temperatures than a comparable solid rotor when the convection model adequately represents internal airflow. This trend agrees with the finite-element observations of Kang and Cho (2012) and with the thermal analysis of Belhocine and Bouchetara (2012a).

When the transient temperature field is transferred to the structural model, the heated friction ring expands relative to the cooler hat and opposite surfaces. The resulting incompatibility produces an umbrella- or cone-like displacement pattern and a non-uniform equivalent-stress field. The maximum stress does not necessarily coincide with the maximum temperature because geometric restraint and section changes strongly affect the stress distribution. Vent holes, vane roots and the hat-to-ring junction therefore require local mesh refinement. Yildiz and Duzgun (2010) showed that ventilation features can raise local stresses and that representing the pad-force distribution more realistically can reduce the predicted stress level.

Table 2. Literature benchmarks for interpreting and checking the proposed FEM model.

Reference	Benchmark topic	Published finding used for model checking
Yildiz & Duzgun (2010)	Ventilated-disc stress analysis	Variable pad loading reduced stress by 8.8–19.1%.
Adamowicz & Grzes (2011)	2D vs 3D transient thermal FE	3D loading resolved circumferential temperature variation.
Kang & Cho (2012)	Solid vs ventilated rotor	Ventilated disc: lower peak temperature; deformation ≈ 0.1162 mm.
Belhocine & Bouchetara (2014)	Sequential thermal–structural FE	Max displacement ≈ 0.28455 mm near braking time; umbrella-like distortion.

These benchmarks should not be used as universal acceptance limits because rotor geometry, vehicle mass, material properties, braking schedule and cooling conditions differ across studies. Instead, they provide order-of-magnitude and trend checks. For example, if a severe braking model predicts an almost uniform rotor temperature, negligible coning and no stress concentration near constrained or ventilated regions, the heat-flux application, contact area, convection assumptions or structural restraints should be re-examined. Conversely, an extremely high local stress that changes strongly with one mesh refinement may indicate a geometric singularity rather than a physically meaningful failure location. The most reliable interpretation therefore combines converged field results, energy balance, material limits and experimental correlation.

The main design value of the method is comparative. The same brake event can be applied to alternative rotor materials, vane geometries, disc thicknesses or cooling coefficients while keeping vehicle inputs unchanged. A favorable design should reduce peak and sustained temperature, limit through-thickness gradients, control coning and reduce stress concentration without adding excessive mass. This comparative use of FEA is consistent with the design-oriented studies of Apte and Ravi (2006), Belhocine and Bouchetara (2014), and Belhocine et al. (2015).

A sensitivity study is also recommended before any design conclusion is accepted. Heat-partition factor and convective heat-transfer coefficient primarily control the thermal response: increasing the fraction of friction heat entering the rotor raises the predicted surface temperature, whereas stronger convection lowers the post-braking temperature and accelerates recovery. The structural response is especially sensitive to the imposed hub constraints, the temperature gradient through the rotor thickness and the coefficient of thermal expansion. If the mounting region is fixed too

rigidly, the model may overpredict thermal stress; if it is under-constrained, rigid-body motion can contaminate the deformation result. Therefore, boundary-condition sensitivity should be reported together with the nominal solution.

For numerical reliability, peak temperature and equivalent stress should be plotted against mesh density and thermal time step. A converged model should show progressively smaller changes in these quantities while retaining the same physical hot-spot region. The temperature history should also be checked against the available braking energy: a solution that stores or dissipates substantially more energy than was introduced through friction indicates an error in heat flux, active contact area, units or boundary conditions. These checks are particularly important because a visually smooth contour plot can still be numerically or physically incorrect. The final outcome of the methodology is therefore not a single contour image but a verified set of temperature, deformation and stress histories that can be compared consistently across rotor designs.

Conclusion and Discussion

A structured methodology has been developed for thermal and structural analysis of an automotive disc brake using the finite element method. The approach converts vehicle braking conditions into a time-dependent frictional heat input, solves the three-dimensional transient thermal field, transfers the resulting nodal temperatures to a structural model and evaluates thermal deformation and equivalent stress. The method explicitly includes heat partition, convection, local mesh refinement, time-step control and convergence checks because each can materially affect predicted brake temperature and stress. Literature published up to 2016 consistently shows that non-uniform frictional heating creates large thermal gradients, that ventilation modifies both heat dissipation and local structural response, and

that thermal expansion can produce coning, hot spots and stress concentrations relevant to judder and fatigue. For a final production assessment, temperature-dependent material properties, experimentally correlated heat-transfer coefficients, realistic pad contact pressure and dynamometer validation should be added. Within these limits, the sequential FEM procedure provides an efficient basis for comparing rotor geometry and material choices before physical prototyping.

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