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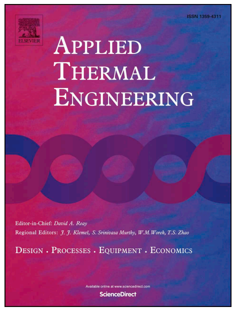

Reduced scale thermal characterization of automotive disc brake

Abdulwahab A. Alnaqi, David C. Barton, Peter C. Brooks

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Reduced scale thermal characterization of automotive disc brake

[1]Alnaqi, Abdulwahab A.*;[1]Barton, David C.; [1]Brooks, Peter C.

[1]University of Leeds, United Kingdom

* Corresponding author. Tel: +44-7423278626.
E-mail address: mnaaal@leeds.ac.uk (A.A. Alnaqi).

Abstract

The thermal behaviour of a disc brake is a critical factor that needs to be considered at the design phase. Most researchers utilise a full size brake dynamometer or a simple pin-on-disc rig to experimentally evaluate the performance of a friction pair (disc and pad). In the current paper, a scaling methodology is proposed to evaluate the thermal performance of a disc brake at a reduced scale. The resulting small scale disc brake has the advantage of low cost and reduced development time. The proposed scaling methodology was validated by comparing the results for the full and small scale discs using a conventional brake dynamometer. In addition, a two dimensional axisymmetric transient thermal finite element model was developed using Abaqus software to assist in the validation of the scaling methodology. The numerical simulations confirmed the equivalence between the full and small scale disc thermal performance using the proposed scaling methodology and also gave good agreement with the experimental results. It is concluded that the scaling methodology is an important tool with which to evaluate the thermal performance of disc brakes in the early design phase.

Keywords: Disc brake, thermal performance, dynamometer, small scale.

Notation

α	Thermal diffusivity	$[m^2/s]$
γ	The ratio of heat flux into the pad to the total heat flux	[---]
μ_a	The viscosity of the air	$[kg/ms]$
ρ	Material density	$[kg/m^3]$
ρ_a	The density of the air	$[kg/m^3]$
ω	Rig rotational speed	$[rad/s]$
τ	Torque	$[Nm]$

A	Pad area	$[m^2]$
A_F	Full scale pad area	$[m^2]$
A_S	Small scale pad area	$[m^2]$
c_p	Specific heat capacity	$[J/kg.K]$
d_o	Brake disc outer diameter	$[m]$
d_t	Wheel rolling diameter	$[m]$
F_n	Normal force	$[N]$
h	Convective heat transfer coefficient	$[W/m^2.K]$
k	Thermal conductivity coefficient	$[W/m.K]$
k_a	Thermal conductivity coefficient of the air	$[W/m.K]$
m_d	Disc mass	$[kg]$
γ	Heat partition ratio	[---]
Q	Heat energy quantity	$[J]$
q''_x	Heat flux per unit area	$[W/m^2]$
q''_{rad}	Radiation heat flux per unit area	$[W/m^2]$
r_m	Mean rubbing radius of disc brake	$[m]$
r_{mF}	Mean rubbing radius of full scale disc brake	$[m]$
r_{mS}	Mean rubbing radius of small scale disc brake	$[m]$
Re	Reynolds number	[---]
S	Scaling factor	[---]
t_d	Brake disc thickness	$[m]$

T	Temperature	[^{o}C]
V_i	Initial forward vehicle velocity	[m/s]
v	Sliding velocity	[m/s]

1. Introduction

The foundation brake is one of the most important systems in a road vehicle as it plays a major part in slowing the vehicle by converting the kinetic energy of the vehicle to heat energy that is dissipated through the disc brake and pads. To find an optimum design, the development process for disc brakes involves a number of steps and many aspects of the braking system need to be considered to ensure that it meets both legal and customer criteria. The conventional design process using full scale dynamometer testing is expensive and time consuming because to achieve the desired goal there are complicated experimental procedures which need to be carried out. One key aspect of these procedures is to assess the maximum temperature reached by the brake discs and pads during critical braking events since these temperatures not only affect the friction performance of the system but also ultimately the structural integrity of the brake. Thermal modelling using theoretical considerations and finite element software is another approach which can be used in the design process to save time and cost to investigate the thermal performance of disc brakes under different loading conditions [1-4].

The brake dynamometer is an excellent research platform as the test conditions and braking parameters can be carefully controlled. There are two major types of dynamometer: the inertial dynamometer and the CHASE dynamometer. The inertial dynamometer is used to evaluate full sized brakes but this is a very time consuming and expensive process. In contrast, the CHASE dynamometer uses a small amount of friction material rubbing against a drum and it requires a shorter testing time than the inertial dynamometer [5].

A small scale test rig presents an alternative way to potentially reduce the cost and time of disc brake design [6], since it involves lower material overheads than full scale testing and so increases the potential for rapid back-to-back testing [7]. A reduced scale testing system has been used in the past for different applications, such as screening for friction stability using the FAST machine and monitoring drum lining material using the CHASE machine [8, 9]. Furthermore, reduced scale testing can improve the accuracy and reproducibility of results by reducing spurious effects such as caliper and bracket deflection and pressure fluctuations [6]. Moreover, one of the areas that needs to be considered carefully is convective cooling as the cooling rates of the reduced scale and full size configuration are not equivalent because of the different physical geometries [6]. Therefore, scaling is a complex process and careful tuning of the scaled parameters is needed in order to obtain comparable results [10].

A pin-on-disc type rig has been utilised as an experimental setup in the literature [10-12] to investigate friction materials. This uses a single pad pushed against one side of a rotating disc. Other studies use two brake pads attached 180 degrees from each other, again pressed against one side of a rotating disc [6, 7]. However, none of the previous small scale studies has tried to implement a realistic brake caliper, which allows the pads to be applied to both sides of the disc simultaneously to represent the real world configuration of an automotive brake.

In the present research, the main goal is to develop a scaling methodology that can be used to inform the design of a reduced scale brake dynamometer especially with regard to the thermal performance. This paper firstly outlines the assumptions underlying the scaling process before deriving the equations required to give equivalent thermal performance between the small and full scale brake. An existing conventional brake dynamometer is then described, followed by the derivation of the design parameters of the equivalent small scale system. The paper proceeds to compare the measured disc surface temperatures between the full size and small scale discs for two different drag brake events. Finally the results of finite element simulations of the two differently scaled brake rotors are compared with the experimental data to demonstrate the validity of the scaling approach adopted.

2. Thermal analysis of solid brake rotor

Understanding the thermal performance of an automotive disc brake is the key factor in developing a scaling methodology that replicates real world conditions. In this section the thermal analysis of a disc brake is presented in brief. In order to predict the temperature distribution of the disc brake, the heat flux generated by friction between the pad and disc is required. The following assumptions apply:

- The kinetic energy of the vehicle is converted to thermal energy due to friction at the sliding interface without any other energy loss during the braking event.

- The heat flux generated by friction at the interface between the pad and the disc is transferred to the brake pads and disc according to their respective thermal properties.
- Heat loss by radiation from the disc is included in this study along with heat transfer by convection and conduction.
- All brake parts are in a steady state condition before braking commences.

A one dimensional schematic model of a disc brake is illustrated in Figure 1. This model was used to derive the finite difference equation required to evaluate the thermal performance of the disc. The numerical equations for the one dimensional disc brake model were derived from the energy balance equation and the heat diffusion equation with assumed constant thermal conductivity [13-15].

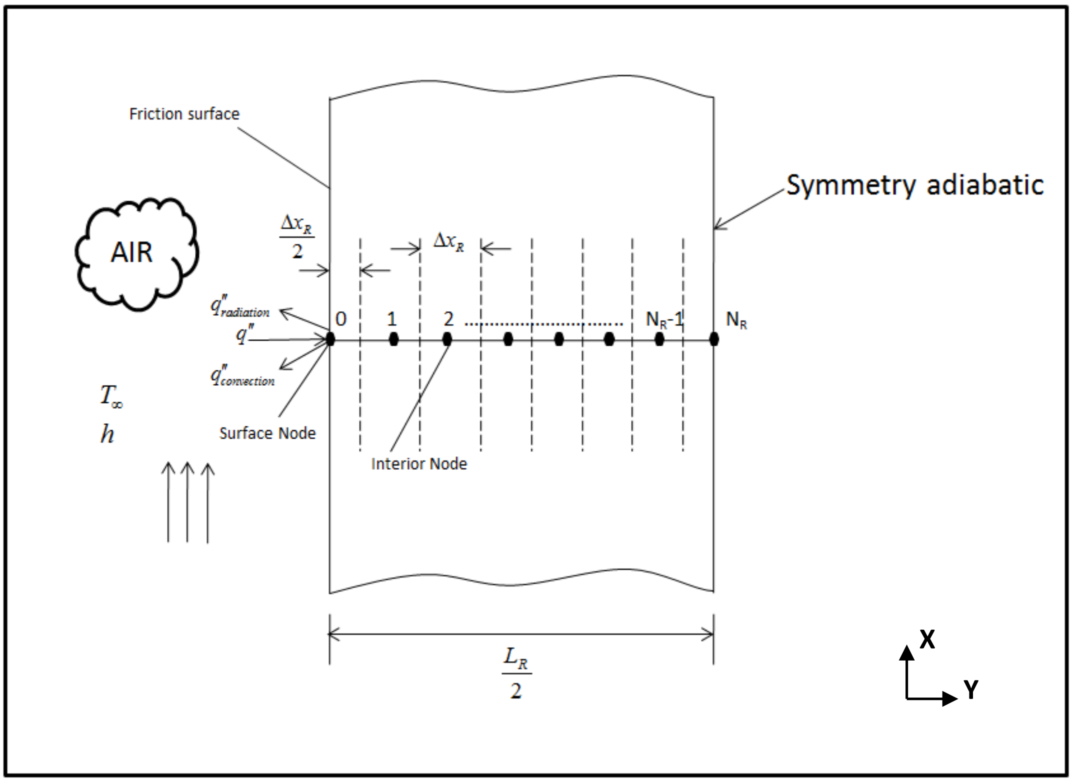


Figure 1: One dimensional thermal model for a brake disc

The heat diffusion equation or heat equation with constant thermal conductivity is as follows:

$$\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} + \frac{\partial^2 T}{\partial z^2} + \frac{\dot{q}}{k} = \frac{1}{\alpha}\frac{\partial T}{\partial t} \quad (1)$$

where $\alpha = \dfrac{k}{\rho c_p}$ is the thermal diffusivity of the disc material. This equation provides the temperature distribution $T(x, y, z)$ as a function of time, which aims in the derivative of the transient one dimensional numerical simulation of the brake. Considering the one dimensional system in Figure 1, under transient conditions with no internal heat generation and constant properties, equation (1) becomes:

$$\frac{1}{\alpha}\frac{\partial T}{\partial t} = \frac{\partial^2 T}{\partial x^2} \tag{2}$$

The central difference approximation to the second order spatial derivative is as follows:

$$\frac{\partial^2 T}{\partial x^2}\Big|_m \approx \frac{T_{m+1}^p + T_{m-1}^p - 2T_m^p}{\Delta x^2} \tag{3}$$

Where the subscript m is used to designate the location of the nodal point in x and the superscript p is used to define the time dependence of T where:

$$t = p\Delta t \tag{4}$$

Then the finite difference approximation to the time derivative in equation (3) can be expressed as for the one dimensional analysis:

$$\frac{\partial T}{\partial t}\Big|_m \approx \frac{T_m^{p+1} - T_m^p}{\Delta t} \tag{5}$$

Substitution of equations (5) and (3) in equation (2) yields:

$$T_m^{p+1} = \frac{1}{M}\left(T_{m+1}^p + T_{m-1}^p\right) + \left(1 - \frac{2}{M}\right)T_m^p \tag{6}$$

where

$$M = \frac{\Delta x^2}{\alpha \Delta t} \quad and \quad N = \frac{h\Delta x}{k}$$

Equation (6) is valid for the interior nodes of the disc. The following equation may be used for the node on the symmetry adiabatic boundary, with $T_{m+1}^p = T_{m-1}^p$:

$$T_N^{p+1} = \frac{1}{M}\left(2T_{N-1}^{p}\right) + \left(1 - \frac{2}{M}\right)T_N^{p} \tag{7}$$

The temperature of the surface node T_0^{p+1} with convection and radiation heat loss, can be derived using the energy equation as follows:

$$\dot{E}_{in} + \dot{E}_g - \dot{E}_{out} = \dot{E}_{st}$$

$$hA(T_\infty - T_0^{p}) + \frac{kA}{\Delta x}(T_1^{P} - T_0^{p}) + q''A - q''_{rad}A = \rho c A \frac{T_0^{p+1} - T_0^{p}}{\Delta t} \tag{8}$$

where $\dot{E}_{st}$ is the rate of stored energy (mechanical and thermal), $\dot{E}_g$ is the rate of the thermal energy generation and $\dot{E}_{in}$ and $\dot{E}_{out}$ are the rates of the energy entering and leaving the control surface (inflow and outflow energy). Rearranging equation (8) for T_0^{p+1}:

$$T_0^{p+1} = \left(1 - \frac{2N+2}{M}\right)T_0^{p} + \frac{2NT_\infty}{M} + \frac{2T_1^{p}}{M} + \frac{2\Delta x q''}{kM} - \frac{2\Delta x q''_{rad}}{kM} \tag{9}$$

The condition for mathematical stability must also be satisfied in order to realise a stable system and this requires choosing M to satisfy the following condition [13, 15]:

$$M \geq 2N + 2 \tag{10}$$

The above equations were embedded within a bespoke Matlab m-file and used in the development of the scaling methodology described below [3, 16].

3. Scaling methodology

The scaling factor is the fundamental relationship used in the scaling methodology. The physical specification of the small scale test brake was developed by applying the scaling factor to the full scale disc as explained below. The guiding principle of the scaling exercise is that both the tribological and thermal conditions at the friction interface should be the same for the small and full scale brakes. Since the friction coefficient is dependent on contact pressure and sliding speed as well as on temperature, the assumption is that both pressure and sliding velocity should be the same at both scales. Thus, provided the scaling technique ensures the same interface temperatures, the heat generation and tribological conditions should also be comparable between the two scales.

3.1 Disc mass

The main parameter for the proposed scaling process is the scaling factor (S), which is defined as the ratio between the full scale and small scale brake pad areas. If full scale parameters are denoted by a subscript F and small scale parameters are denoted by a subscript S, then:

$$S = \frac{A_F}{A_S} \tag{11}$$

where A_F is the full scale pad area and A_S is the small scale pad area.

The disc mass was scaled using the energy balance equation:

$$Q = m_d c_p \Delta T \tag{12}$$

where Q is the heat flow from or to the disc during a braking event, m_d is the disc mass, c_p is the specific heat and ΔT is the difference between the final and initial temperatures. As it is assumed that the energy density (heat flow per unit pad area) in the full and small scale cases should be equal, this leads to:

$$\frac{Q_F}{Q_S} = \frac{A_F}{A_S} = S \tag{13}$$

Substituting equation (12) into equation (13) leads to:

$$\frac{(m_d c_p \Delta T)_F}{(m_d c_p \Delta T)_S} = S \tag{14}$$

One of the main aims of the scaling exercise is to replicate the thermal condition acting on the full scale brake and this means that the temperature rise ΔT should be the same. Thus equation (14) reduces to:

$$\frac{(m_d c_p)_F}{(m_d c_p)_S} = S \tag{15}$$

If the specific heat of the material of the full and small scale brake discs is the same, this equation implies that the disc masses should scale linearly with the pad area ratio S .

3.2 Brake torque

The brake torque was calculated using the following equation:

$$\tau = 2\mu F_n r_m \tag{16}$$

where F_n is the normal force pushing each pad against the disc, μ is the average coefficient of friction and r_m is the mean rubbing radius. The contact pressure for small and full scale brakes was assumed constant in the current scaling methodology in order to give the same tribological conditions, which leads to:

$$\frac{F_{nF}}{A_F} = \frac{F_{nS}}{A_S} \tag{17}$$

where F_{nF} is the normal force for the full scale pad and F_{nS} is the normal force for the small scale pad. Substituting equation (16) into equation (17) leads to:

$$\frac{\tau_F}{A_F r_{mF}} = \frac{\tau_S}{A_S r_{mS}} \tag{18}$$

Assuming the same friction coefficient for both scales (since the sliding velocity, contact pressure and temperature are assumed to be the same) and scaling the rubbing radii with the square root of the pad area ratio S leads to:

$$\frac{\tau_F}{\tau_S} = S^{3/2} \tag{19}$$

Equation (19) was used to calculate the brake torque to be generated by the small scale disc brake assembly from the equivalent full scale value.

3.3 Rotational speed

In order to obtain the same tribological conditions at the friction interface, the small scale rig sliding velocity is set equal to the full scale sliding velocity v which is derived from the initial forward speed of the vehicle V_i as follows:

$$v = r_{mF} \frac{2V_i}{d_t} \tag{20}$$

where r_{mF} is the full scale mean rubbing radius and d_t is the road wheel rolling diameter. The rotational speed of the small scale rig is given by:

$$\omega_S = \frac{v}{r_{mS}} \tag{21}$$

where ω_S is the initial rig rotational speed and r_{mS} is the mean rubbing radius for the small scale case. Equating the sliding velocity in equations (20) and (21), leads to:

$$\omega_S = \frac{2V_i}{d_t}\left(\frac{r_{mF}}{r_{mS}}\right) \tag{22}$$

Equation (22) defines the small scale rotational speed as a function of the vehicle wheel rolling diameter, initial vehicle speed and ratio of the mean rubbing radius for small and full scale rigs. The mean disc rubbing radius is a linear quantity and thus scales with $S^{0.5}$ according to the scaling methodology; this relation is used to evaluate the relation between the full and small scale rotational speeds by utilising equation (22) as follows:

$$\frac{\omega_F}{\omega_S} = S^{-0.5} \tag{23}$$

In other words, the disc rotational speed should scale with the inverse of the square root of the pad area ratio S .

4. Full scale brake dynamometer and full/reduced scale brakes

A full scale brake dynamometer, shown in Figure 2, was used to test both the small scale brake designed using the above scaling methodology and the equivalent full scale brake in order to validate the scaling methodology. A Lorey Somer LSK1604M04 45 kW DC electric motor rotates the main dynamometer shaft via an enclosed belt drive. The main shaft is supported by two roller element bearings between which is mounted a torque meter (Torquemaster TM 213) and speed encoder. An ACME screw linear actuator (LMR 01) [17] was used to pressurise the hydraulic system via a standard brake master cylinder. The brake dynamometer was controlled and monitored using an in-house LabVIEW based data acquisition system.

The rotor and brake pad geometries for the full scale brake are shown in Figure 3 and for the small scale brake in Figure 4. Note that the full size rotor has the conventional ‘‘top-hat’’ structure for connecting the rotor to the hub whereas the small scale rotor is a plain disc without such a structure. Both discs were manufactured from standard grey cast iron whilst the pads were made from proprietary friction material supplied by the manufacturer.

In the case of the full scale disc a BENDIX brake caliper (No. 520 1889 794997) was used with a Girling Aluminium Master Cylinder unit. A Wilwood PS1 brake caliper (part no. 120-8374), was used for the small scale disc with Wilwood Go Kart master cylinder (part no. 260-5520) [18]. For both small and full size brakes, K type sliding thermocouples were used to monitor the rubbing surface temperature of the rotor as shown in Figure 3 and 4. The sliding thermocouple was placed at the mean rubbing radius in both cases. All the sensors and actuators were calibrated before commencing any test in order to ensure that the output data were accurate and reliable.

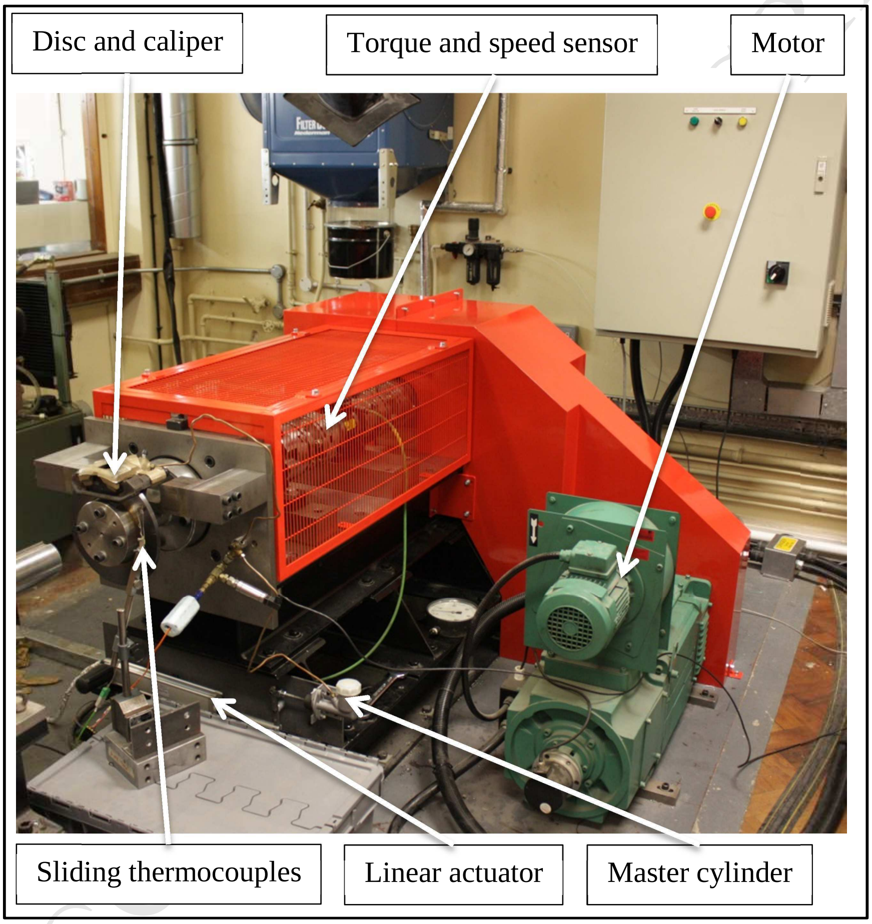


Figure 2: Brake dynamometer with full scale brake mounted for testing.

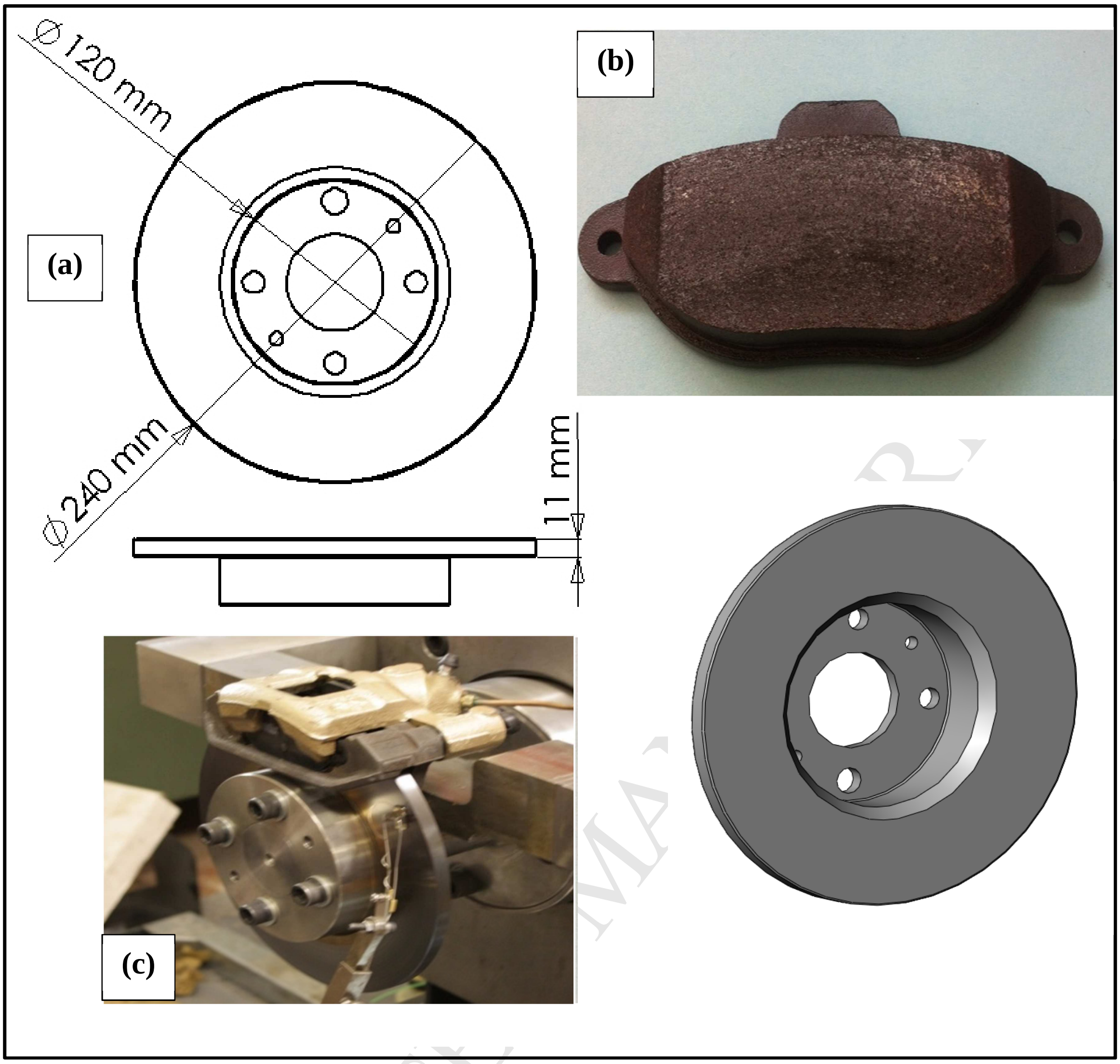


Figure 3: Full scale disc and pad with thermocouple position: (a) disc geometry, (b) brake pad, and (c) brake assembly mounted on dynamometer.

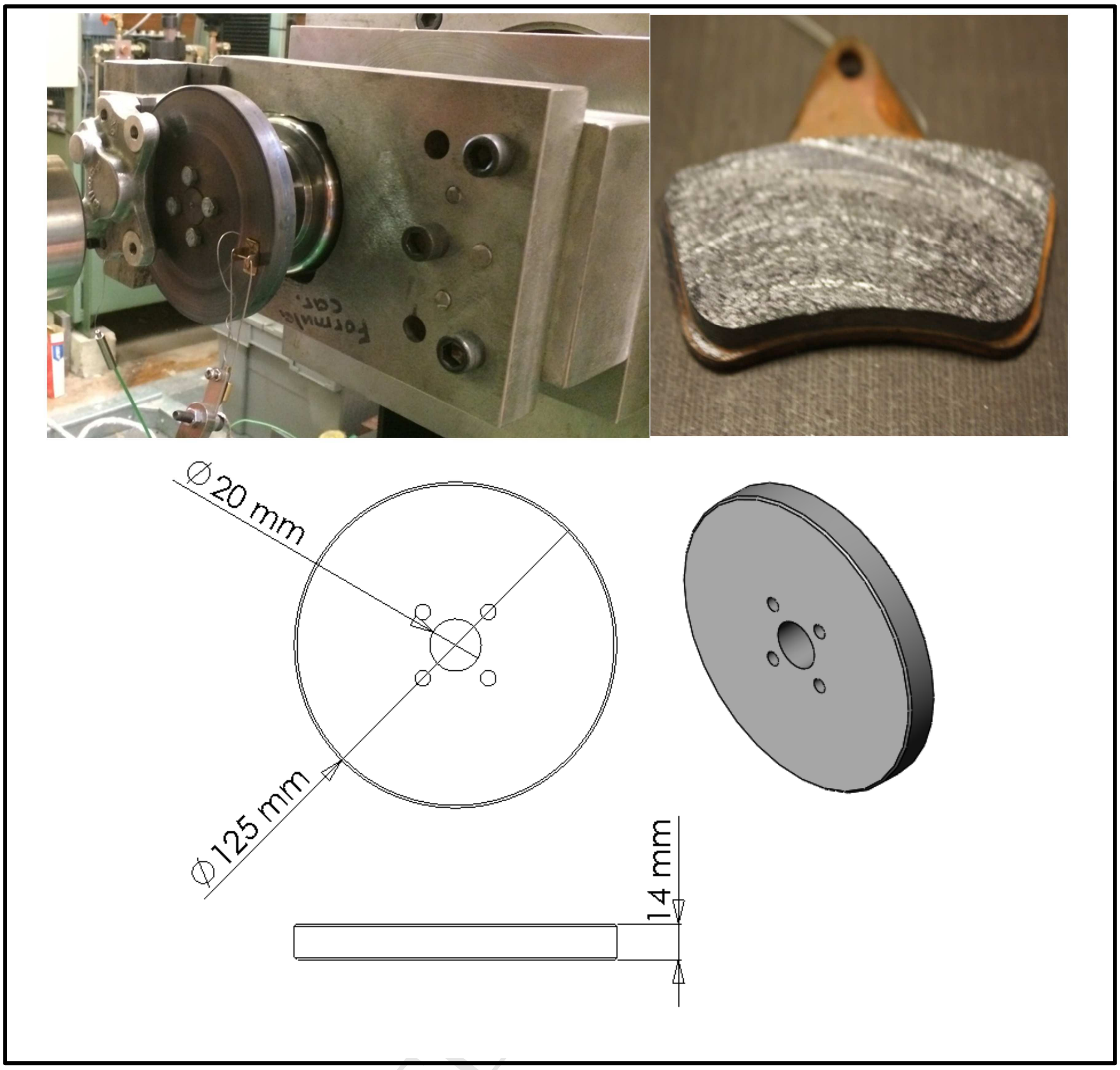


Figure 4: Small scale disc and pad with thermocouple position

The control and data acquisition system of the brake dynamometer is shown in Figure 5. The user controls the test rig using LabVIEW software that has been developed in-house in order to give full control over the braking conditions. The user controls the braking pressure through the linear actuator and the speed of the motor by changing the input voltage whilst the system measures, braking pressure, braking torque, motor speed and the temperatures of the disc and pad.

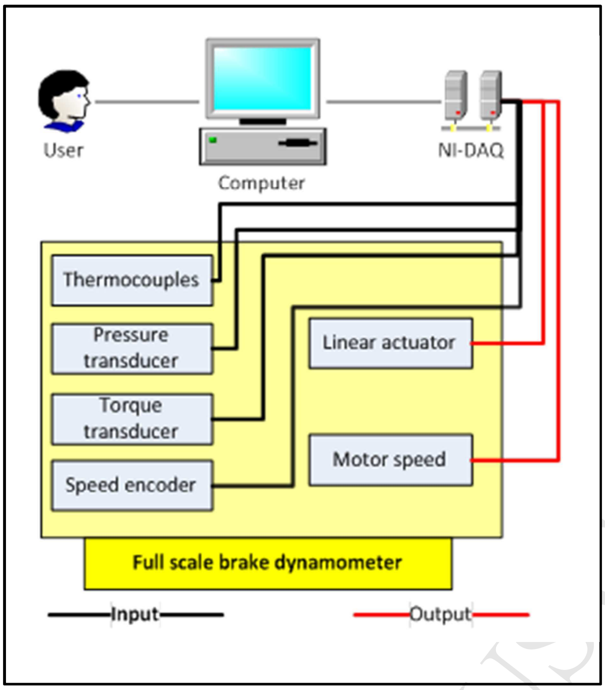


Figure 5: General hardware configuration of the full scale brake dynamometer

5. Experimental validation of scaling methodology

In this section a comparison between the experimental results from the small scale and full scale brakes is presented. The scaling methodology described in Section 3 was used to calculate the reduced scale parameters as shown in Table 1 for a scaling factor of 3. In addition, the thermal properties of both full and small scale grey cast iron discs were assumed to be as follows: density 7100 kg/m^3, specific heat 500 J/kg.K and temperature invariant thermal conductivity 51.5 W/m.K.

Table 1: Full and small scale parameters for scaling factor of 3

Vehicle Parameter	Full scale	Small scale	Relation
Pad area (mm^2)	2736	900	$S = \frac{A_F}{A_S}$
Mean rubbing radius (mm)	95	54	
Disc mass (kg)	3.57	1.2	$r_F = r_S S^{1/2}$
Disc thickness (mm)	11	14	$\frac{m_F}{m_S} = S$
Disc outside diameter (mm)	240	125	

A drag braking scenario was used to validate the scaling methodology. Drag braking is a type of brake application used to maintain a constant vehicle velocity on a downhill descent rather than to completely stop the vehicle. The drag braking conditions used for the full scale disc brake assumed a vehicle speed of 40 km/h on a slope of 4% for a vehicle of mass 1000 kg which is equivalent to a rig rotational speed of 350 rpm and a target brake torque of 35 Nm. In the case of the small scale disc brake, the target torque of 7 Nm was calculated based on equation (19). The equivalent small scale rig rotational speed was calculated from equation (23) to be 620 rpm. The actual torque achieved over the 270 s duration of the test is shown for both full and small scale discs in Figure 6. Since the hydraulic pressure acting on the brake pads was kept constant, the torque variations according to equation (16) are due to variations in coefficient of friction caused mainly by temperature changes at the sliding interface.

The surface temperature for both full scale and small scale discs was monitored at the mean rubbing radius throughout the duration of the tests and the results are shown in Figure 7. The surface temperature of the small scale disc showed good agreement with that of the full scale disc with a maximum difference of the order of 10 ^{o}C. There are factors not accounted for in the scaling methodology that might have an effect on the temperature of both small and full sized discs. For example, the convective cooling rate may be different in each case because of the different disc diameters and rotational speeds; there may also be geometric effects due to the different pad aspect ratios (circumferential arc length/ radial width) along with the presence of the ‘‘top-hat’’ structure for the full scale rotor which is not present on the small scale disc.

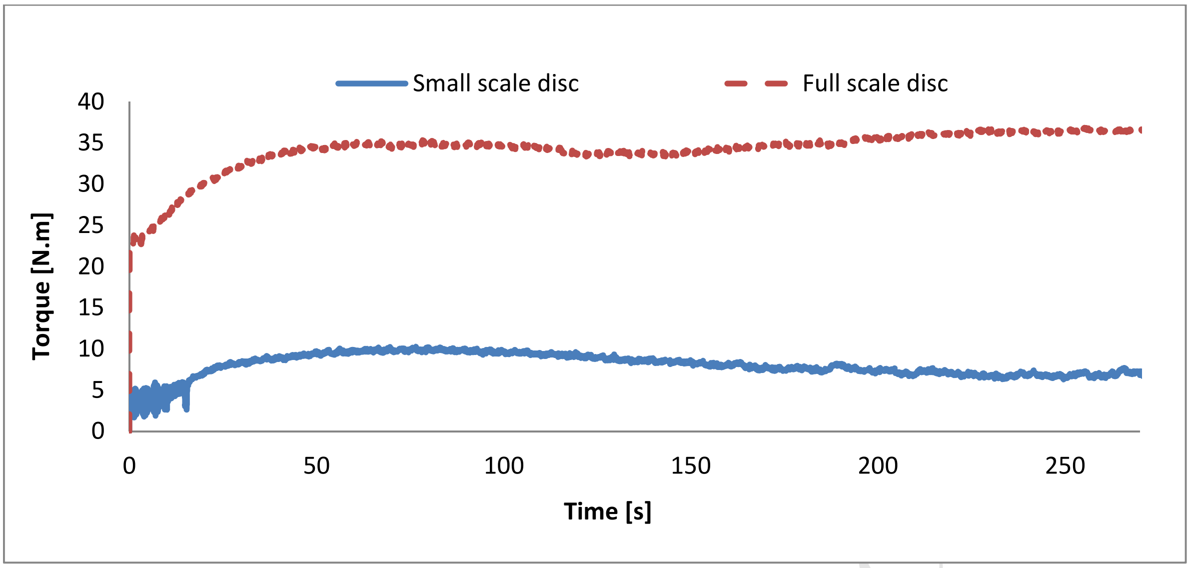


Figure 6: Measured torque response for full scale and small scale brakes

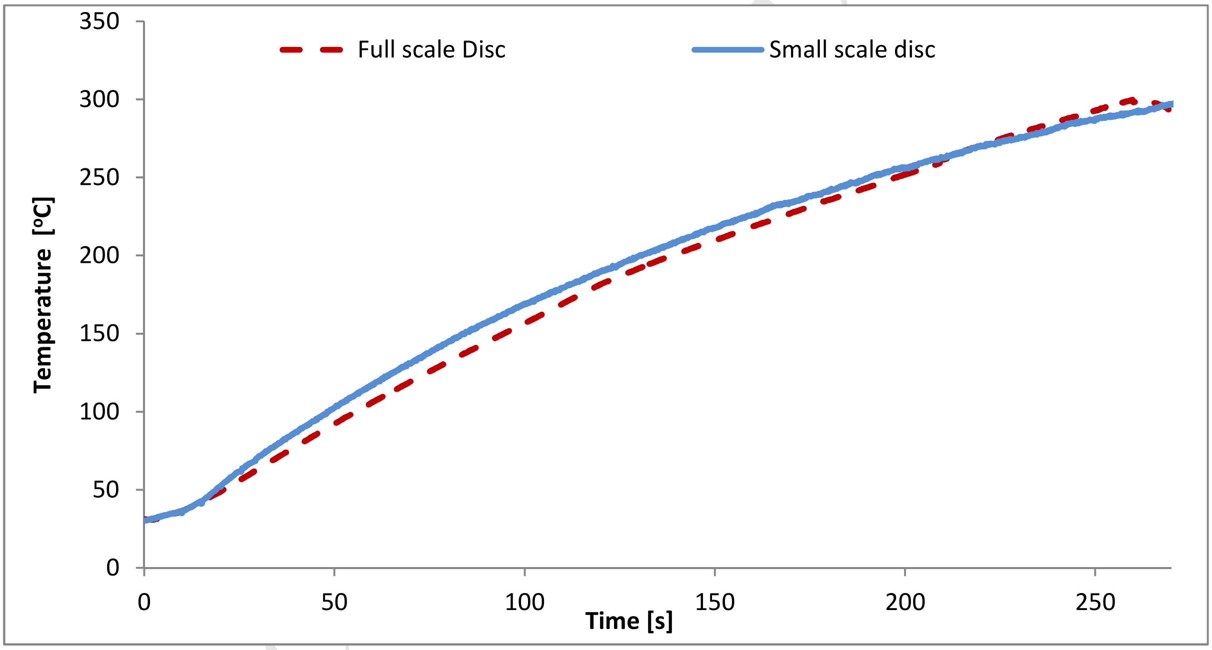


Figure 7: Disc surface temperature for small scale and full scale brakes.

6. Numerical validation of scaling methodology

Abaqus FEA software [19] was used to further investigate the scaling methodology and the thermal performance of the small and full scale solid brakes. Two dimensional axisymmetric transient heat transfer models of the discs were developed in Abaqus\standard. The model setup and boundary conditions for the small and full scale discs are shown in Figure 8 and 9 respectively. The models were meshed using 4-node linear axisymmetric heat transfer quadrilateral elements (DC2D4).The total number of elements for the full scale disc brake was 335 whilst 220 elements were used for the small scale disc brake rotor.

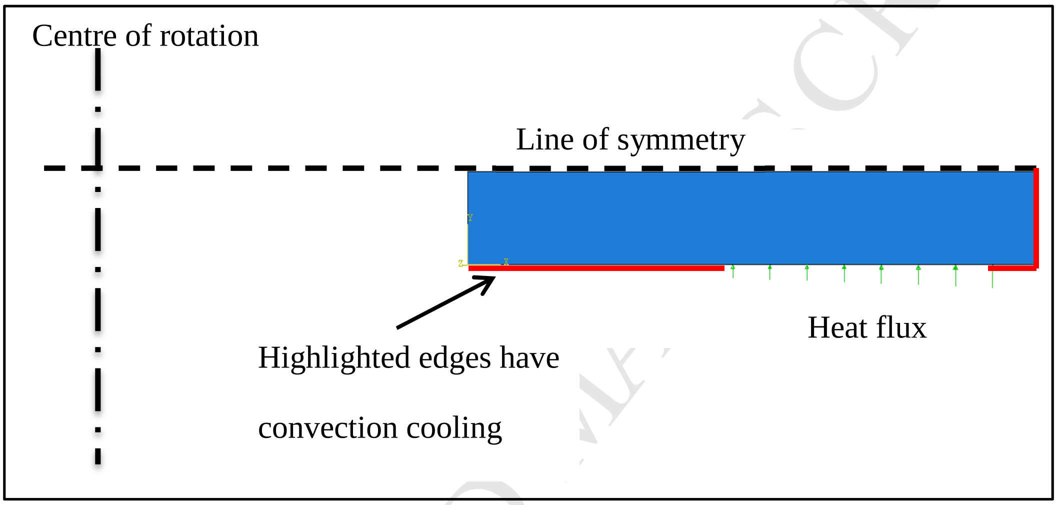


Figure 8: Two dimensional axisymmetric finite element model of small scale disc

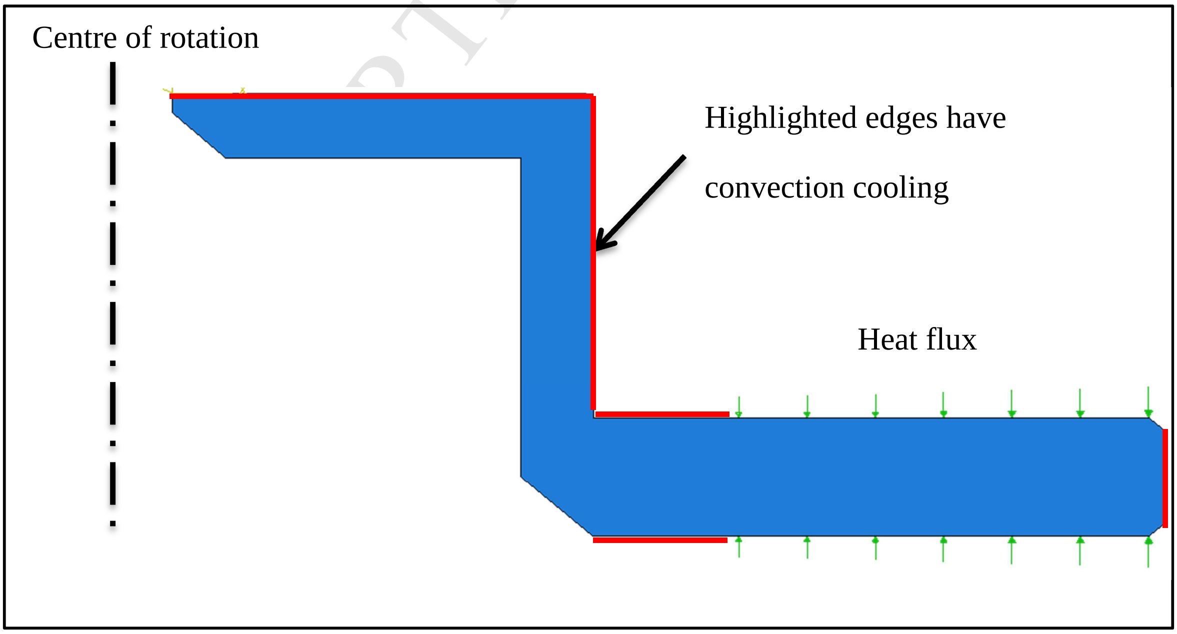


Figure 9: Two dimensional axisymmetric finite element model of full scale disc

The convective heat transfer coefficient was calculated using equations (24, 25). It was used in each model and was assumed constant in each case.

For laminar flow, the convective heat transfer coefficient, h_R, is defined by:

$$h_R = 0.7\left(\frac{k_a}{d_o}\right)\mathrm{Re}^{0.55} \qquad For\ \ \mathrm{Re} <= 2.4\times10^5 \tag{24}$$

and for turbulent flow, it takes on the value:

$$h_R = 0.04\left(\frac{k_a}{d_o}\right)\mathrm{Re}^{0.8} \qquad For\ \ \mathrm{Re} > 2.4\times10^5 \tag{25}$$

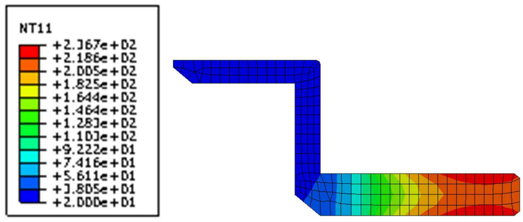
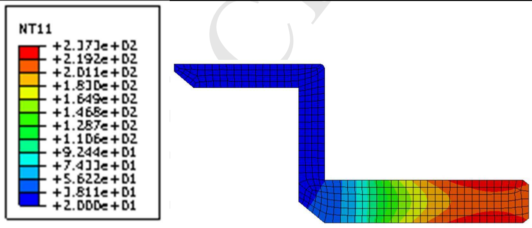
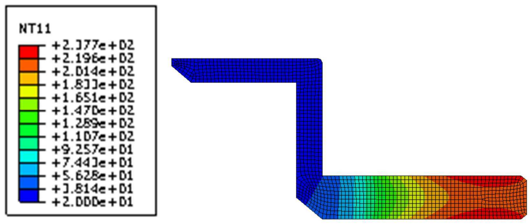
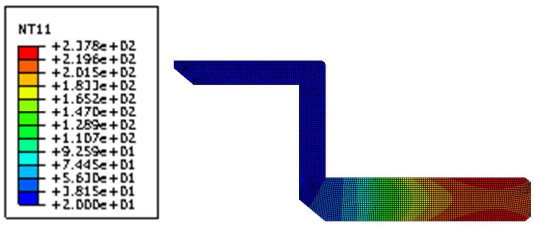
where d_o is the outer diameter of the disc, k_a is the thermal conductivity of air in $W/m.K$ and Re is the Reynolds number:

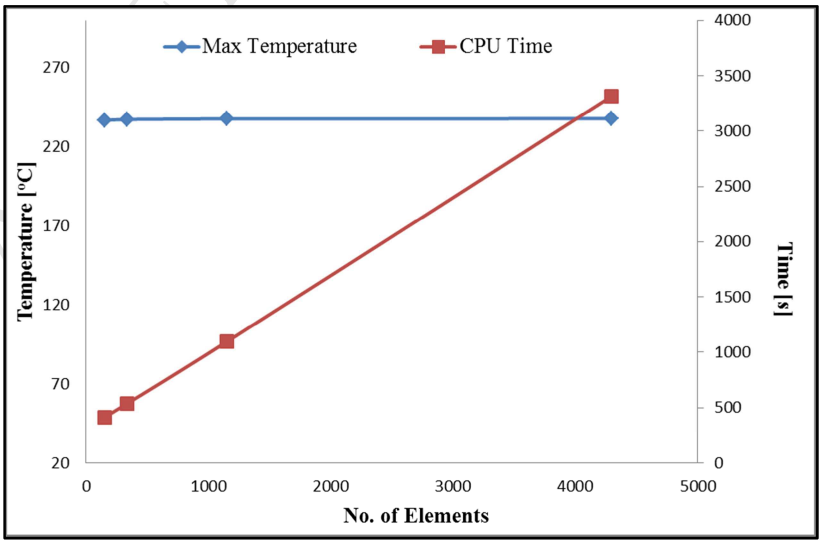
$$\mathrm{Re} = \frac{V\rho_a l}{\mu_a} \tag{26}$$

in which V is the vehicle speed in m/s, ρ_a is the density of the air in kg/m^3, l is the characteristic surface length (assumed to be the diameter of the disc in m) and μ_a is the viscosity of the air in $kg/m.s$. All the air properties are assumed to be at ambient temperature in the present work.

A mesh sensitivity analysis was carried out using a constant heat flux in order to optimise the number of elements in the 2D model. The rotational speed of disc was 1225 rpm and brake torque was assumed to be a constant 80 Nm. The simulation time for the analysis was 10 s. The results show that the global element size has a small effect on the temperature distribution and maximum surface temperature as shown in Table 2. On the other hand, there is a trade-off between the number of elements (element size) and the simulation time. Consequently, a 0.0019 m element, representing a compromise between CPU time and solution accuracy, was chosen for all the subsequent 2D axisymmetric simulations. Although a smaller element size offers slightly more accurate results, this takes around twice as long to run which was deemed inappropriate for the long drag brake event. The same procedure was used to investigate the mesh sensitivity for the small scale disc brake model and showed that 220 elements gave accurate results for reasonable run times.

Table 2: Mesh sensitivity analysis for full scale disc

Global element size [m]	No. of elements	T_{max} [^oC]	CPU Time [s]	Model
0.0025	153	236.8	415	
0.0019	335	237.3	537	
0.001	1153	237.7	1097	
0.0005	4293	237.8	3311	



The experimental results were first used to validate the full scale Abaqus model. The heat flux was calculated based on the experimental data using equation (24):

$$Q = \tau\omega \tag{24}$$

where τ is the measured braking torque (as for example shown in Figure 6) and ω is the rig rotational speed (assumed constant). However, only a certain proportion of this heat flux is transferred into the disc, the remainder being transferred to the pad friction material. The proportion of heat transferred to the disc (the partition ratio) can theoretically be calculated from the respective thermal properties of the disc and pad materials using equation (25) [13, 14]:

$$\gamma = 1 - \left(\frac{1}{1 + \sqrt{\frac{\rho_p c_p k_p}{\rho_d c_d k_d}}} \right) \tag{25}$$

From the published properties of the grey cast iron discs and estimated property of the pad materials, Equation (25) predicts γ to be 0.87. However this equation takes no account of the transfer layer that can form on both rubbing surfaces and in practice a higher proportion of heat is often found to flow to the disc than suggested by equation (25). In the present case good agreement was found between the measured and predicted surface temperatures for a partition ratio of 0.95 which is similar to the value typically used by other researchers [14, 20]. For example, Figure 10 shows the predicted and measured surface temperature for the full scale disc when subjected to the applied torque time history shown in Figure 6 assuming a heat partition ratio of 0.95. It can be seen that very good agreement is achieved throughout the duration of the test.

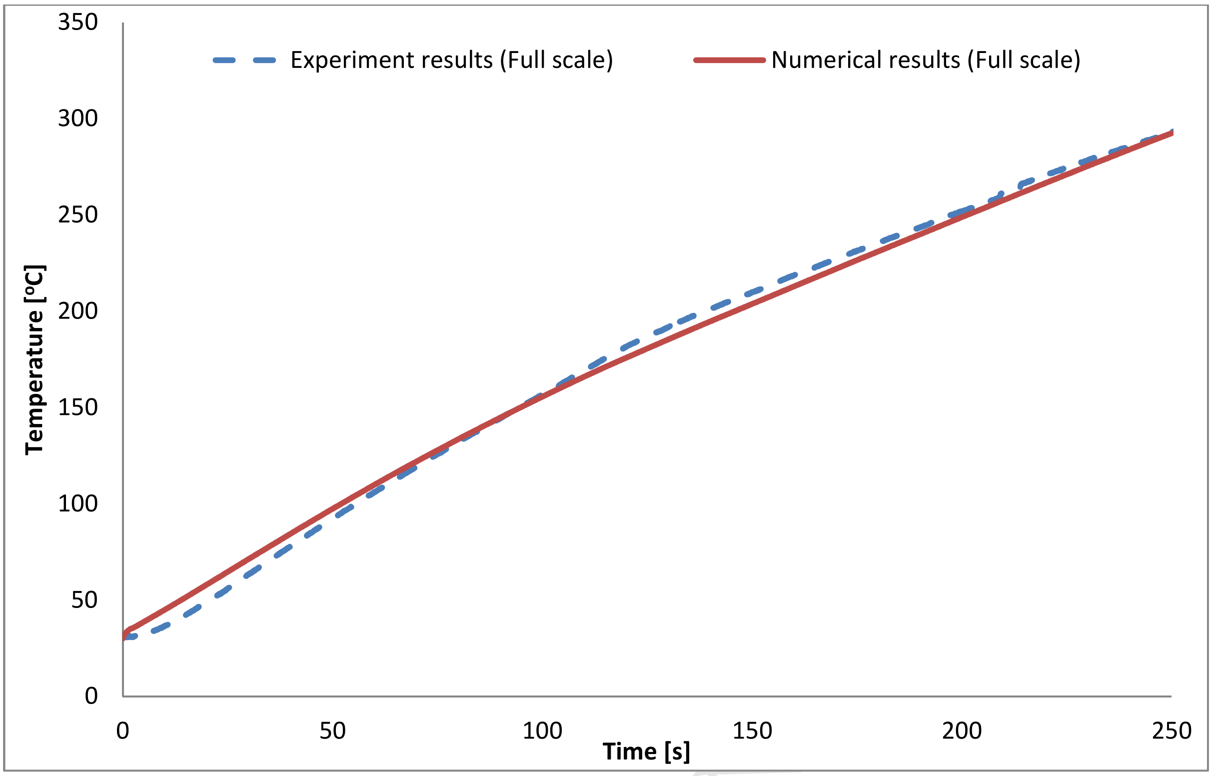


Figure 10: Comparison of the experimental and numerical full disc surface temperature response

Having validated the FE model of the full size brake against the experimental results, the 2D axisymmetric models were used to simulate very high speed drag braking in order to further investigate the thermal performance of both small and full scale brakes that would otherwise be beyond the capacity of the available brake dynamometer. Figure 11 shows the predicted surface temperature at the mean rubbing radius for both small and full scale brakes. The rotational speed of the full scale brake was 1225 rpm and brake torque was assumed to be a constant 80 Nm. Using the scaling methodology defined above the corresponding rotational speed of the small scale brake was 2170 rpm and the brake torque was 15 Nm. The braking simulated is very aggressive (equivalent to braking a 1000 kg vehicle travelling at a constant 140 km/h down a 5% gradient) but it can be seen from Figure 11 that the difference in the surface temperature at mean rubbing radius between the full and small scale disc brake at the end of the simulation is only of the order of 10 ^{o}C. This difference is considered acceptable given that there are differences in the brake geometry which were not accounted for in the scaling methodology.

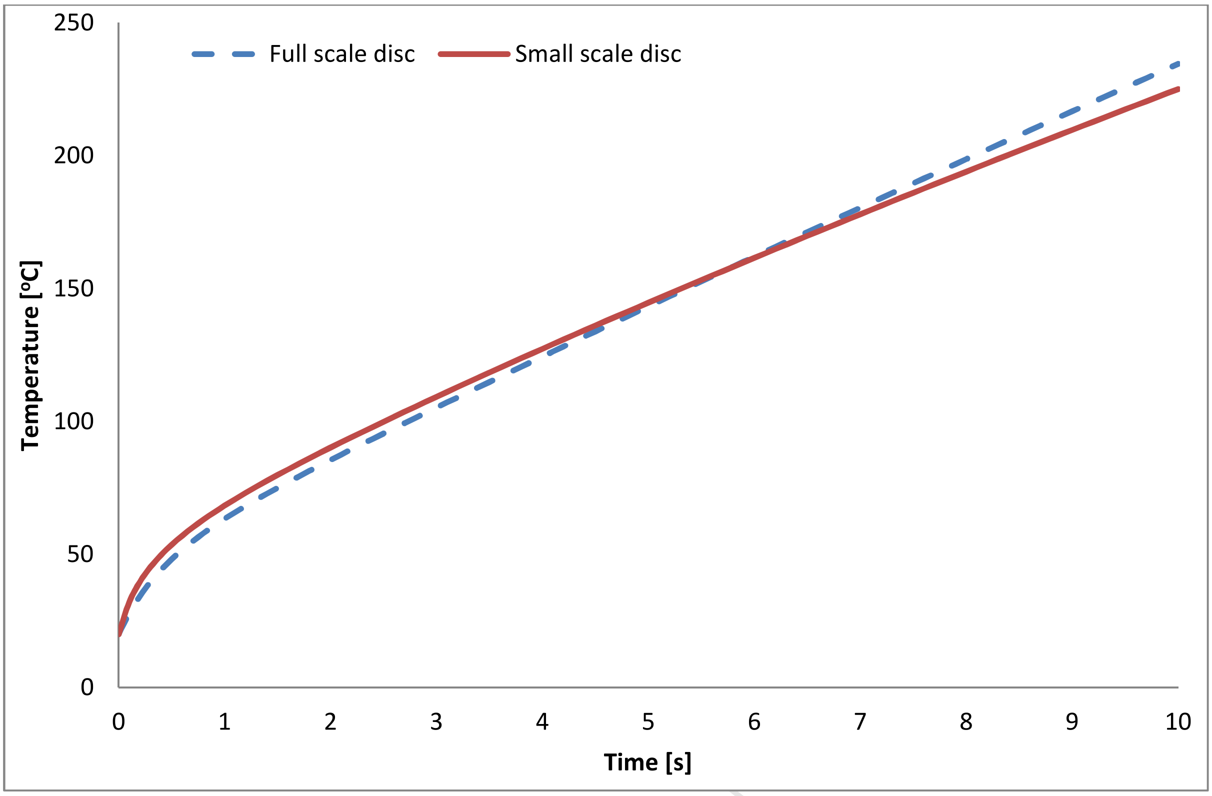


Figure 11: Numerical disc surface temperature for small scale and full scale discs during high drag braking

The temperature distribution within both the full and small scale discs at different time steps for this extreme drag braking event was investigated further as shown in the contour plots of Figures 12 and 13 respectively. Despite the difference in geometry, as shown in Figure 3 and 4, the temperature distribution for the two discs is very similar. This is because, in the scaling methodology, the thermal mass of the disc is considered to be the more important parameter over the geometry. In addition, the convection cooling effect will have an influence on the temperature distribution between the full and small scale discs: the full scale disc has a larger surface area exposed to the environment than the small scale disc brake even though the same convective heat transfer coefficient was assumed throughout. Overall the detailed temperature distributions further demonstrate the validity of the scaling methodology.

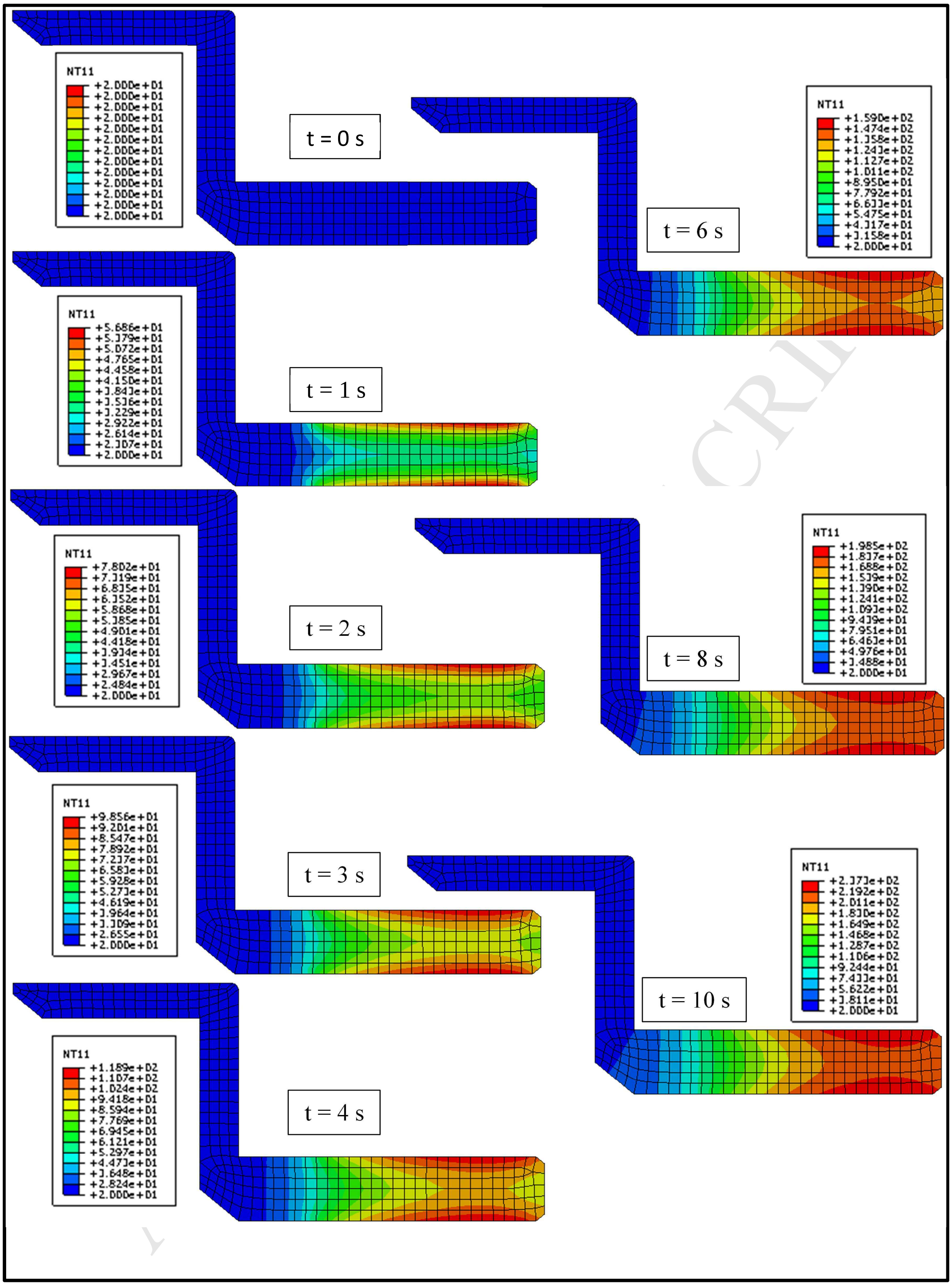


Figure 12: Temperature distribution of the full scale disc at different time steps

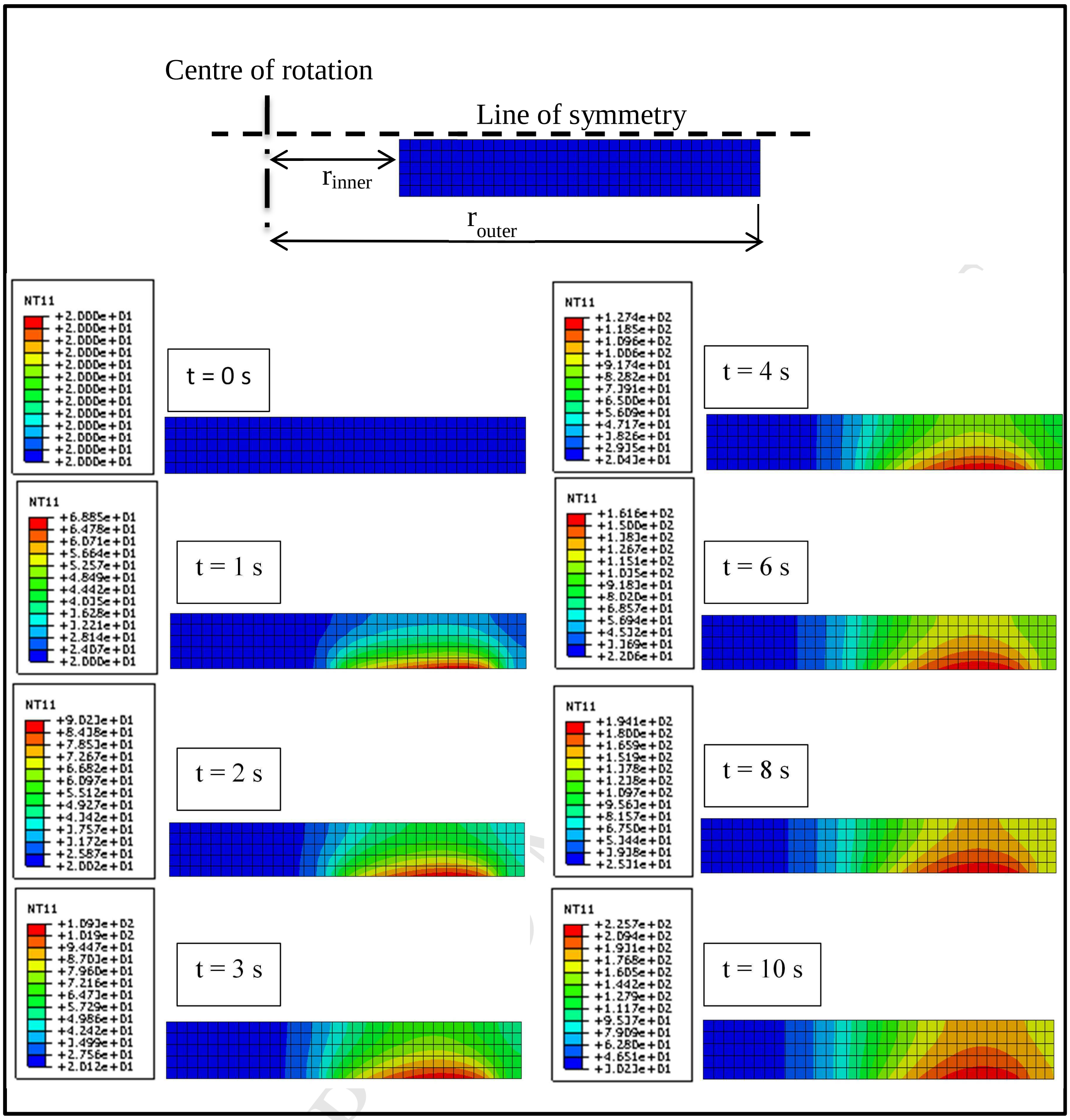


Figure 13: Temperature distribution of the small scale disc at different time steps

7. Conclusions

A theoretical and experimental basis for representing the thermal performance of automotive brake discs at a reduced scale has been presented. Experimental data from the reduced scale disc was compared with the corresponding full scale disc data obtained from a laboratory brake dynamometer. The results obtained showed generally very good agreement between the measured sliding surface temperatures for both the full and small scale discs, thereby demonstrating the validity of the scaling approach.

The thermal response of the full and small scale disc brakes was also investigated using a two dimensional axisymmetric finite element model and the results confirmed that both discs experienced almost the same maximum surface temperatures. Although there are detailed differences in the temperature distributions because of the effect of different convective cooling and the different disc geometry, overall the results demonstrated that the scaling methodology can be used with confidence for the design and development of automotive disc brake systems.

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Highlights

- One dimensional thermal model of solid disc brake was derived numerically.
- A scaling methodology is proposed to evaluate the thermal performance of a disc brake.
- The scaling methodology was validated experimentally and numerically.
- The results demonstrated that the scaling methodology can be used with confidence.
- 2D axisymmetric finite element model was used to investigate the thermal performance of disc brake.