

Thermal Performance of Monolithic and Coated Disc Brakes Using Abaqus and Matlab Software

Abdulwahab Alnaqi, David Barton, and Peter Brooks

Department of Mechanical engineering, University of Leeds, LS2 9JT, UK

Abstract: The brake system is a critical item of safety equipment in a vehicle and provides the driver with adequate control over the speed of the vehicle and is classified as one of the most crucial parts in any vehicle. Rotor reliability and the thermal behaviour of various small scale disc brake rotors were investigated using a small scale brake dynamometer rig and a number of different numerical approaches. Abaqus and Matlab software were firstly used to construct a one dimensional (1D) thermal model to investigate the thermal performance of disc brakes. The 1D model was validated with literature and experimental results. An axisymmetric thermal model was then developed using Abaqus in order to investigate the temperature distribution through the brake rotor. The effect of a coating layer on a disc brake rotor was investigated using Abaqus implicit analyses which included an enhancement of the modelling of the coating layer and as assessment of the results accuracy. Matlab code was constructed to communicate with the Abaqus input file in order to perform a repetitive braking events and optimisation analysis. This code was developed in order to save time and to utilise the optimisation toolbox in Matlab. The numerical results showed good agreement with the experimental results obtained from the literature.

Keywords: Heat transfer, Disc brake, Small scale, Abaqus and Matlab.

1. Introduction

A disc brake must undergo extensive testing to make sure it functions as desired over a varied range of braking conditions. Furthermore, testing is an essential stage in order to meet the legislative requirements and make sure that the vehicle meets all the safety criteria. Many researchers have investigated the thermal performance of disc brakes by predicting the temperature distribution using different methods such as lumped analysis, a one dimensional analytical method, a two or three dimensional numerical method with different assumptions in each case comparing the numerical results with experimental results (Abbas et al., 1969-70, Amin et al., 2007, Jun et al., 2008, Mcphee and Johnson, 2008, Newcomb, 1960, Talati and Jalalifar, 2009). In this research, a small scale test rig was used instead of a full scale dynamometer. The major advantages of reduced scale testing are highlighted by (Kermc et al., 2005). They found that it is more cost efficient than full scale tests and thus, leads to an increase in the practicality of fast back-to-back testing.

Thermal modelling was introduced by various automotive brake designers as a method of evaluating the system performance early in the development stages which decrease the level of complexity and time consuming investigation (Kim et al., 2007). Different approaches have been used in the modelling of brake systems in order to achieve the best accuracy, but there is a trade-

off between the accuracy of the results and run time (Sheridan et al., 1988). Ideally, the thermal model needs to account for the effect of varying friction coefficient, vehicle geometry and inertia along with different heat flows (El-Sharkawy, 2008). Different studies in the literature combine a collection of software such as Abaqus to create complex, resource-heavy simulations (Adamowicz and Grzes, 2011, Alsaif et al., 2010, Bozic et al., 2012, Grieve et al., 1997).

In addition, other research has been undertaken in the area of lightweight disc design. Most research makes use of aluminium alloy and composites as the material for the brake rotor. The reason for using aluminium alloy is because of its tremendously encouraging properties such as high specific heat capacity, low density and high thermal conductivity which make it a good candidate for many engineering applications. On the other hand, aluminium alloys have disadvantages, namely low maximum operating temperature and low wear resistance which can limit their application (Dahm et al., 2009, Shrestha and Dunn, 2007, Shrestha et al., 2003).

In order to achieve the main aim of replacing the cast iron disc with a light weight disc (coated aluminium alloy), there are many issues to resolve. One of the issues is modelling the thin coating layer in Abaqus software; this issue was investigated by using the thermal resistance approach which is explained in this paper. Furthermore, a novel approach to execute disc brake finite element models using both Abaqus and Matlab software for different braking events is presented. This approach gives the user the power of using the Matlab optimisation toolbox in conjunction with the Abaqus solver. Also it reduces the computation time and memory required for such an analysis.

2. Comparison between Matlab and Abaqus one dimensional models

In this section one-dimensional thermal models were developed in Abaqus and Matlab in order to investigate the thermal performance of a disc brake. The one dimensional Matlab model was firstly validated with experimental results from the literature and then with the Abaqus one dimensional model. The one dimensional model was used to develop the interaction code between Abaqus and Matlab.

2.1 Matlab One Dimensional Model

In this study the thermal analysis was carried out according to the following assumptions: the kinetic energy changes to thermal energy without any other form of energy loss during braking, the heat flux generated through friction between the pad and the disc is transferred from the friction interface to brake parts by conduction; heat transfer by radiation from the disc is included in this study along with heat transfer by convection and conduction. All brake parts are assumed to be at steady state conditions and constant temperature before braking. The one dimensional finite difference model of the solid disc brake is illustrated in Figure 1 in which L_s is the thickness of the disc substrate and L_c is the thickness of the coating.

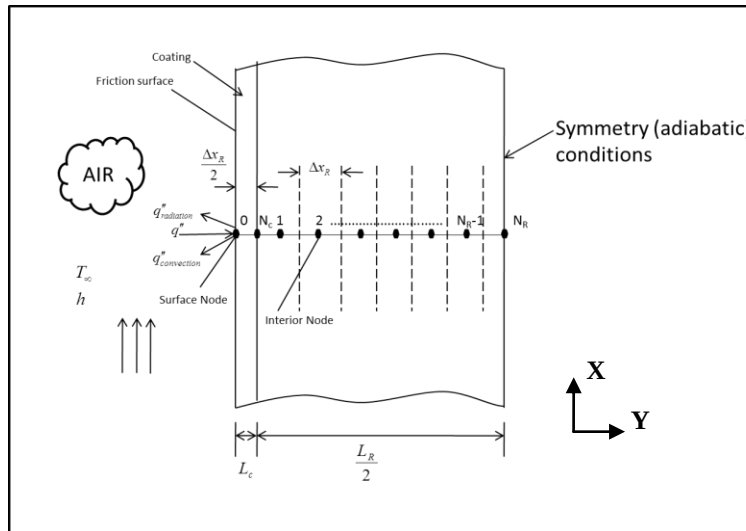


Figure 1. One dimensional thermal model for brake disc

Based on Figure 1, the finite difference equations for calculating the temperature were derived as shown in the literature (Bozic et al., 2012, Limpert, 1999) and embedded within a bespoke Matlab m-file to calculate the temperature through the disc brake. The overall Matlab model structure is shown in Figure 2. Furthermore, this model was modified to calculate the thermal performance of a coated disc brake.

The Matlab model was used to investigate the thermal performance of various small scale disc brake materials. A sensitivity analysis was carried out to find the optimum number of nodes representing the disc and the coating as there is a trade-off between the number of nodes inside the coating and the model running time. The one-dimensional Matlab model was validated with the results from the literature (Newcomb, 1960). The material properties used in this simulation are for steel, as shown in the first line of Table 1. In addition, the specification of the disc brake modelled is presented in Table 2.

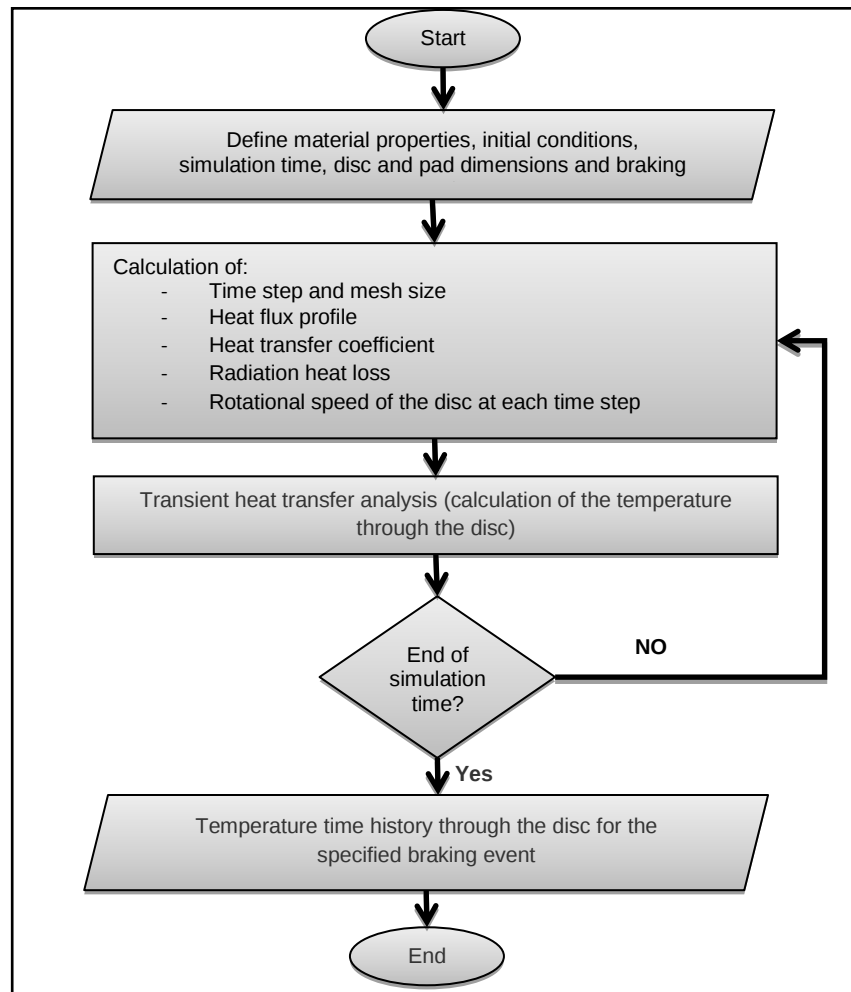


Figure 2. The overall Matlab model structure

Table 1. Thermal properties (Alsaif et al., 2010, Curran and Clyne, 2005, Jun et al., 2008).

Materials	Density ρ [kg/m ³]	Specific heat c_p [J/kg.K]	Conductivity k [W/m.K]
Steel	7854	434	60.5
Grey Cast Iron (GCI)	7100	500	51.5
Al-MMC (AMC640XA)	2900	800	130
Alumina coating	3030	828	1.6
Pad Material	2596	827	0.736

Table 2: Parameters of automotive brake application (Newcomb, 1960).

Inner disc diameter	132 mm
Outer disc diameter	227 mm
Disc thickness	11mm
Mean sliding radius	94.5 mm
Convection coefficient	$60 \text{ Wm}^{-2}\text{k}^{-1}$
Vehicle mass	1800 kg

Using the data in Tables 1 and 2, the disc surface temperature was evaluated using the one dimensional thermal model, based on a braking duration of 4 seconds, an initial vehicle speed of 100km/h and a deceleration of 7m/s^2 . The results were compared with the published results (Newcomb 1960) as shown in Figure 3. The calculated results show very good agreement with the published experimental results.

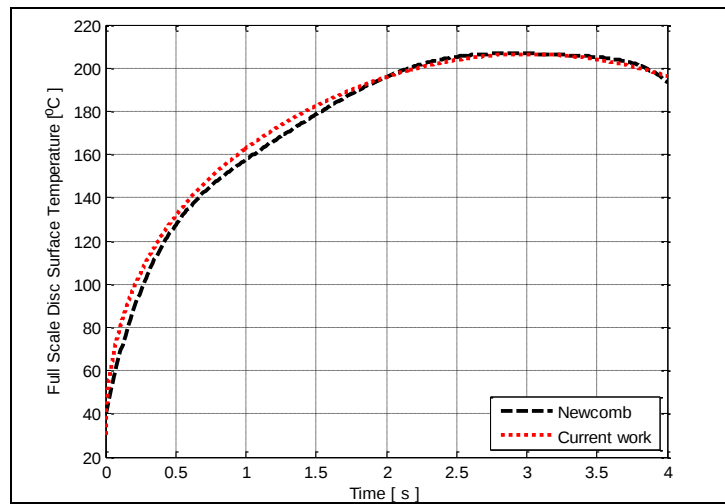


Figure 3. Validation of the current model with published results (Newcomb, 1960).

2.2 Abaqus One Dimensional Model

The performance of the small scale disc brake was also analysed using Abaqus finite element analysis software. A 1D model was first validated against the Matlab model and then used as a guide to develop the 2D and 3D FEA models. Also the 1D model was used to investigate different approaches to include the effect of a coating layer with or without explicitly modelling it. The 1D transient heat transfer model of small scale disc brake was meshed using a 4-node linear axisymmetric heat transfer quadrilateral elements (DCAX4). The same boundary conditions of the 1D Matlab model were applied in the 1D Abaqus model as shown in Figure 4. The overall Abaqus model structure is shown in Figure 5 .

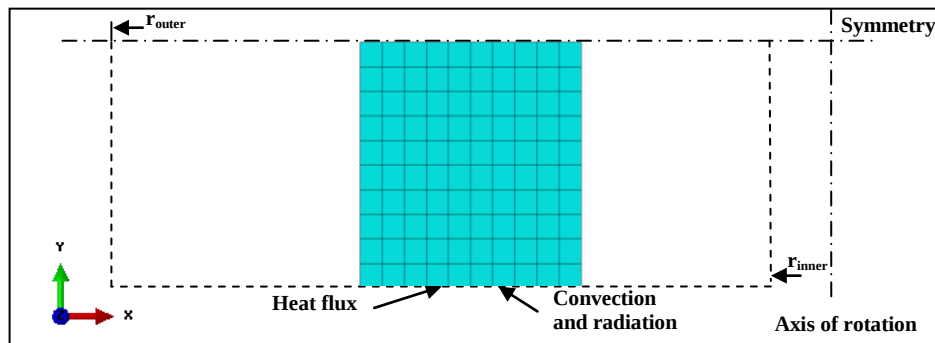


Figure 4. One dimensional FE model in Abaqus

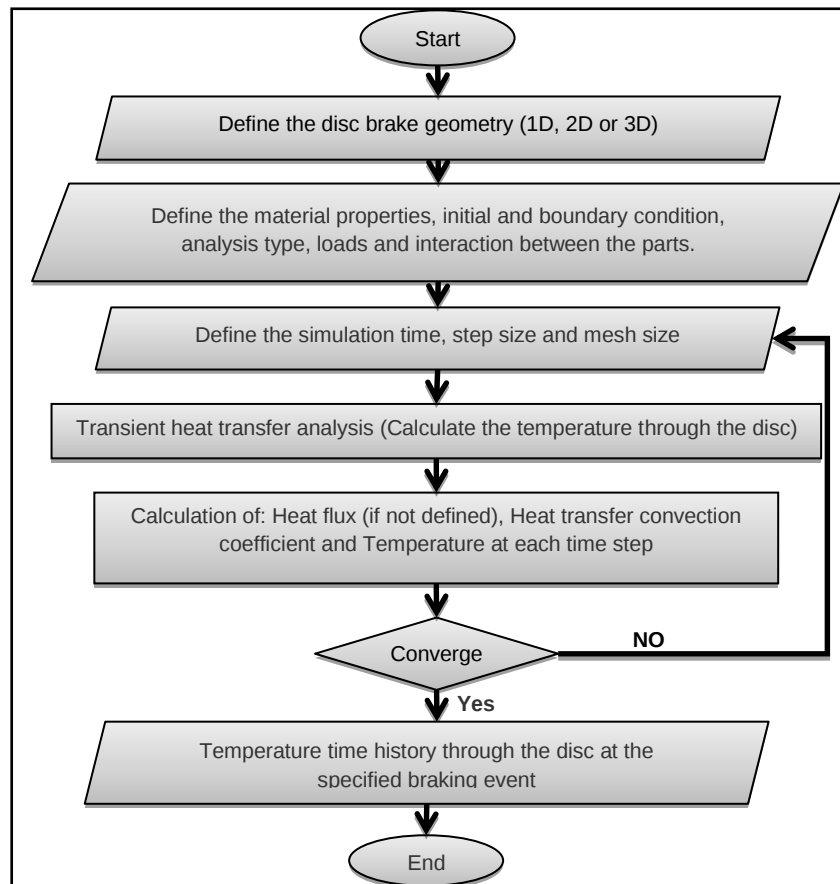


Figure 5. The overall Abaqus model structure

The 1D Abaqus model was validated against the 1D Matlab cast iron disc model. The material properties, initial conditions and braking parameters are shown in Table 1 and 3. The results derived from the Abaqus model shows very good agreement with the results obtained from Matlab as shown in Figure 6.

Table 3: Initial conditions and parameters used for the simulation (Jun et al., 2008).

Parameter	Value	Parameter	Value
Disc thickness, t_d	0.014m	Proportion of braking at front axle, x_f	0.7
Rate of deceleration, d	0.7g	Pad area, A	575 mm ²
Brake disc diameter, d_o	125mm	Pad arc angle, θ_p	60°
Partition ratio, p	0.17	Initial speed, v_i	100km/h
Surroundings temperature, T_∞	20°C	Final speed, v_f	0km/h
Vehicle mass, m	163kg	Vehicle wheel diameter, d_t	0.292m

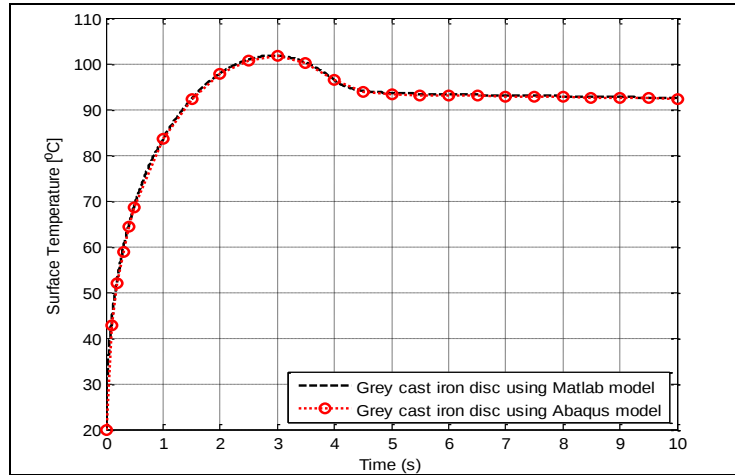


Figure 6. Validation of the numerical Matlab model with Abaqus model.

3. Comparison between explicit and thermal resistance modelling of coating within Abaqus

The main problem of modelling a thin coating layer on the surface of the brake disc in Abaqus using an explicit or implicit algorithm is the thickness of the coating which is typically in the order of μm ; this requires a much denser mesh than the substrate and causes high computation time. The thermal performance of a coated disc brake was therefore investigated using two different approaches. The Abaqus implicit algorithm was used to solve two transient one dimensional heat transfer models. The first model consists of two bodies directly in contact i.e. an aluminium metal matrix composite (Al-MMC) disc and pad brake modelled using 4 node heat transfer elements as shown in Figure 7a. The second model consists of the same two bodies but with an alumina coating layer explicitly modelled on the Al-MMC disc brake as shown in Figure 7b. The axial

thickness of the disc is 7 mm which is the same as the radius in the 1D model. The radial width of the disc and pad included in the model is also 7 mm although this is arbitrarily chosen as no heat flow occurs in the radial direction in this 1D model. In reality the coating thickness is in the scale of μm (Linck et al., 2005) but, in order to investigate the heat flow and temperature distribution through the coating layer, it was assumed that the coating thickness was $250 \mu m$. This coating thickness was chosen to reduce the model size and computational time. The material properties for the disc, pad and coating used in this analysis are shown in Table 1.

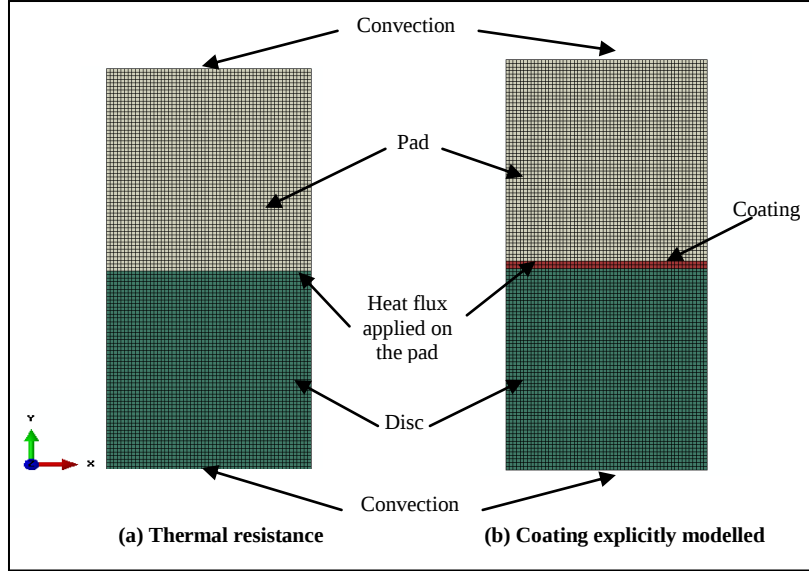


Figure 7. One dimensional static model.

The two transient static models were constructed and solved using Abaqus implicit, one with the coating modelled as a thermal resistance on the disc rubbing surface (Figure 7a) and one with the coating modelled explicitly with finite elements (Figure 7b). A 4 second transient heat transfer analysis with 0.01 second time step was performed in both cases, using 4 node heat transfer element (DC2D4). At the contact interfaces, surface contact with small sliding was chosen. The mesh size was chosen so that the distance between the nodes is 0.125 mm in the heat flow direction. A heat flux of either $1 \times 10^5 \text{ W/m}^2\text{K}$ or $5 \times 10^5 \text{ W/m}^2\text{K}$ was applied on the rubbing surface. Heat transfer by convection was considered in the models with heat transfer convection coefficient of $30 \text{ W/m}^2\text{K}$ for both models, and the component initial temperature was set to 20°C . The thermal resistance for the implicit coating model shown in Figure 7a was obtained using the thermal conductivity, thickness and cross-sectional area of the coating as follow:

$$R_{cond} = \frac{\delta x}{kA} = \frac{0.00025}{1.6} = 0.000156 \text{ m}^2\text{K/W} \quad (1)$$

Abaqus uses a thermal conductance rather than a resistance which is equal to $1/R_{cond} = 6400 \text{ W/m}^2\text{K}$ and this was applied to the rubbing surface of the brake disc. A heat flux of $5 \times 10^5 \text{ W/m}^2\text{K}$ was used to evaluate the temperature across the disc and pad for both models which is shown at the end of the simulation ($t=4 \text{ sec}$) in Figure 8. The x-axis shows the distance from the bottom block (disc) to the top block (pad). In order to compare both results the temperature of the thermal resistance model should be shifted by 0.25 mm to the right (coating thickness).

The results in Figure 8 show that the thermal resistance approach can be used to model the effect of the coating layer with some limitations, the maximum surface temperatures for both models are similar but there is a relatively small difference in the temperature across the disc itself with the temperatures being around 5°C higher for the thermal resistance model. This is because the higher thermal inertia (ρc_p) of the coating is not included in the thermal resistance model. The effect of this is to cause a lower proportion of the frictionally-generated heat to be transferred to the disc as can be seen from the simple heat partition equation:

$$\gamma = \frac{q_p''}{q_p'' + q_R''} = 1 - \frac{1}{1 + \sqrt{\frac{\rho_p c_p k_p}{\rho_R c_R k_R}}} \quad (2)$$

where γ is the ratio of heat flux into the pad to the total heat flux, q_p'' is the pad heat flux, q_R'' is the disc heat flux, c_p is the specific heat of the pad, c_R is the specific heat of the disc, k_p is the thermal conductivity of the pad, k_R is the thermal conductivity of the disc and ρ_p is the pad density and ρ_R is the disc density.

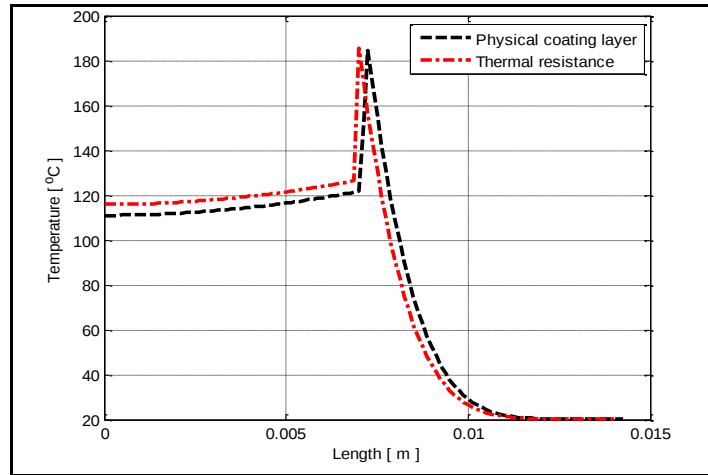


Figure 8. Temperature distribution across the disc and pad without tuning.

The thermal resistance model was then tuned through a trial and error process by raising the convection coefficient on the disc exposed surface. This increased the amount of heat that is transferred to the disc. Figure 9 shows the temperature across the disc, coating and pad brake for the two proposed models after tuning of the thermal resistance model at the end of simulation ($t=4$ sec). The x-axis shows the distance from the bottom block (disc) to the top block (pad). Note that in order to compare both results the temperature of the thermal resistance model should be shifted by 0.25 mm to the right to account for the coating thickness. The results show that the use of the thermal resistance model instead of explicitly modelling the coating layer with finite elements can give good accuracy.

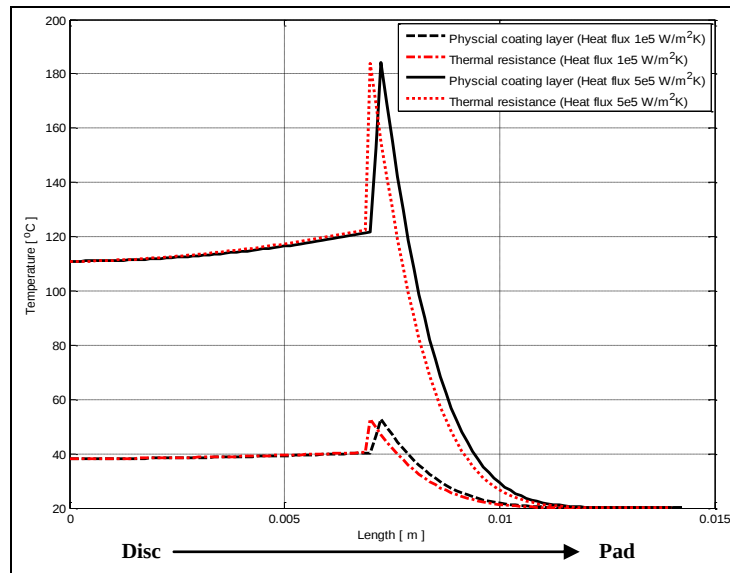


Figure 9. Temperature distribution across the disc and pad with convection tuning.

4. Axisymmetric modeling of coated brake disc

Axisymmetric models were also developed using Abaqus in order to evaluate more realistic finite element models in comparison with the experimental method. Uncoupled axisymmetric transient heat transfer model of small scale disc brake was developed to investigate the temperature in two directions. The model was meshed using 4-node linear axisymmetric heat transfer quadrilateral elements (DC2D4) as shown in Figure 10. A similar coating analysis as explained in the previous section was implemented for the model with the same disc and pad brake material. The disc brake dimension was $7\text{ mm} \times 52.5\text{ mm}$ and the pad block dimension was $7\text{ mm} \times 16\text{ mm}$ as shown in Figure 10. It was assumed that the coating thickness was 0.25 mm as before. A heat flux of $5 \times 10^5\text{ W/m}^2\text{K}$ was applied on the pad contact surface. Heat transfer by convection was considered in the models with heat transfer convection coefficient of $30\text{ W/m}^2\text{K}$. Cooling was applied on

the upper and lower exposed surfaces of the disc and pad, considering the sides as thermally insulated. The same convection analysis used in the one dimensional model was carried out in modelling the 2D thermal model.

Figure 11 shows the temperature in the centre of the rubbing surface. It can be seen that the results from the model with the thermal resistance showed good agreement with the model for which the coating was explicitly modeled. Figure 12 shows the temperature across the centre of the disc and pad at the end of the simulation ($t=4$ s). This has the same trend as the one dimensional model where there is a difference in the disc temperature which can be tuned using the convection coefficient.

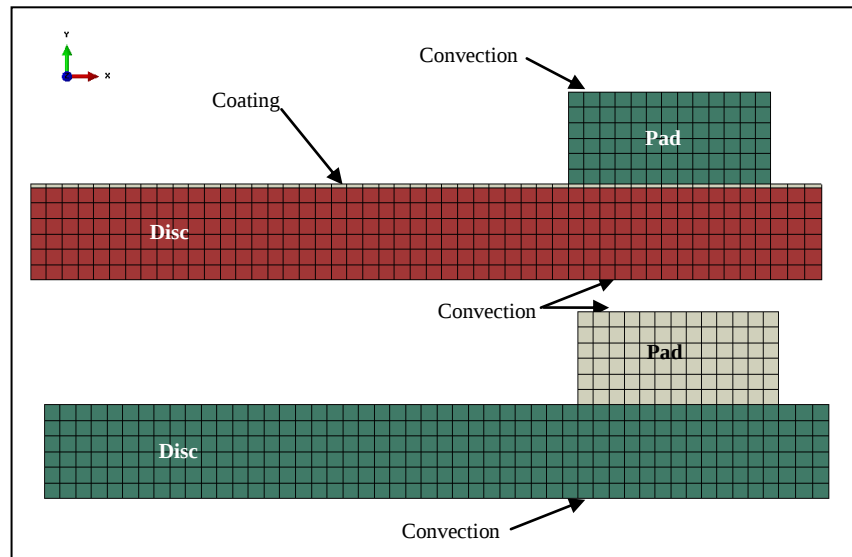


Figure 10. Axisymmetric transient models.

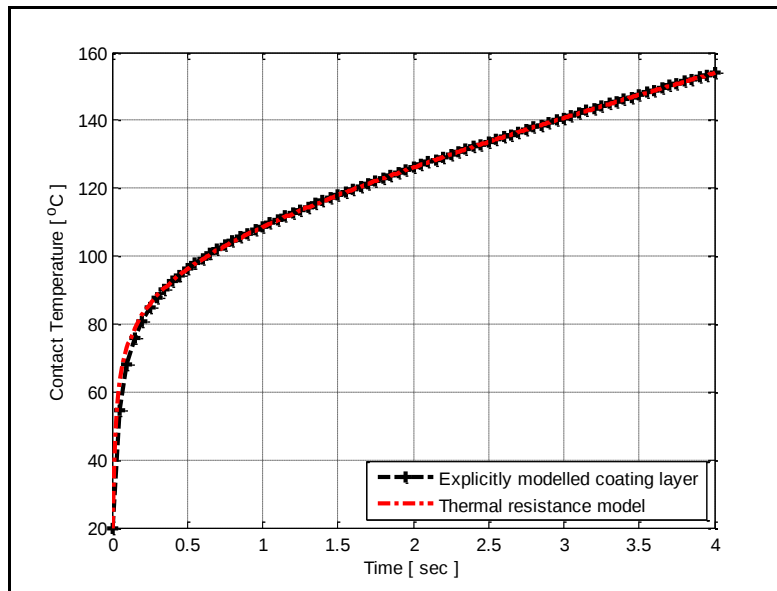


Figure 11. Contact temperature for the two dimensional models.

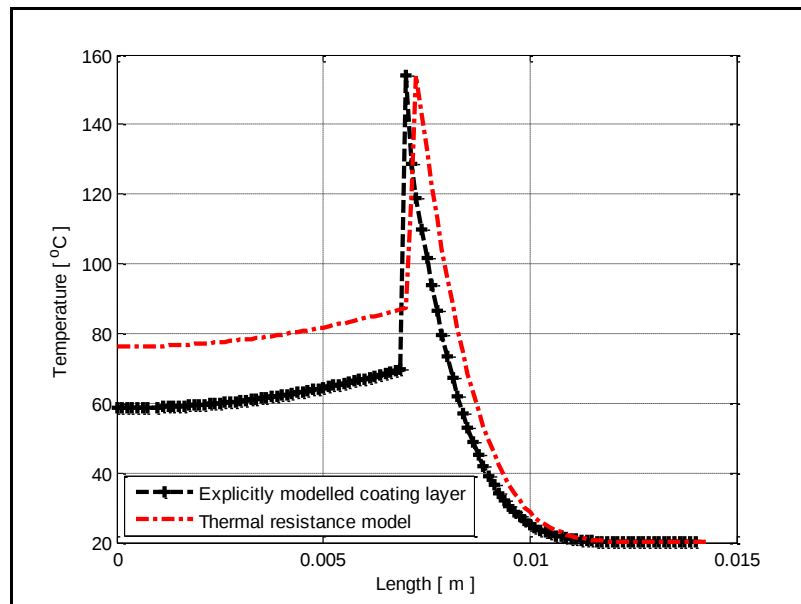


Figure 12. Temperature distribution across the centre of the disc and pad.

5. Abaqus/Matlab interaction interface

Both Abaqus and Matlab have been used to investigate the thermal performance of a solid disc brake rotor under different braking scenarios. The main problem associated with Abaqus was the time and memory usage for the coupled thermal analysis. This problem was solved by conducting an uncoupled analysis with the help of Matlab code to calculate the heat flux profile. Figure 13 shows the overall structure of the code used to permit interaction between Abaqus and Matlab. Another advantage of using this code is to utilise the optimisation toolbox in Matlab with Abaqus modelling capabilities.

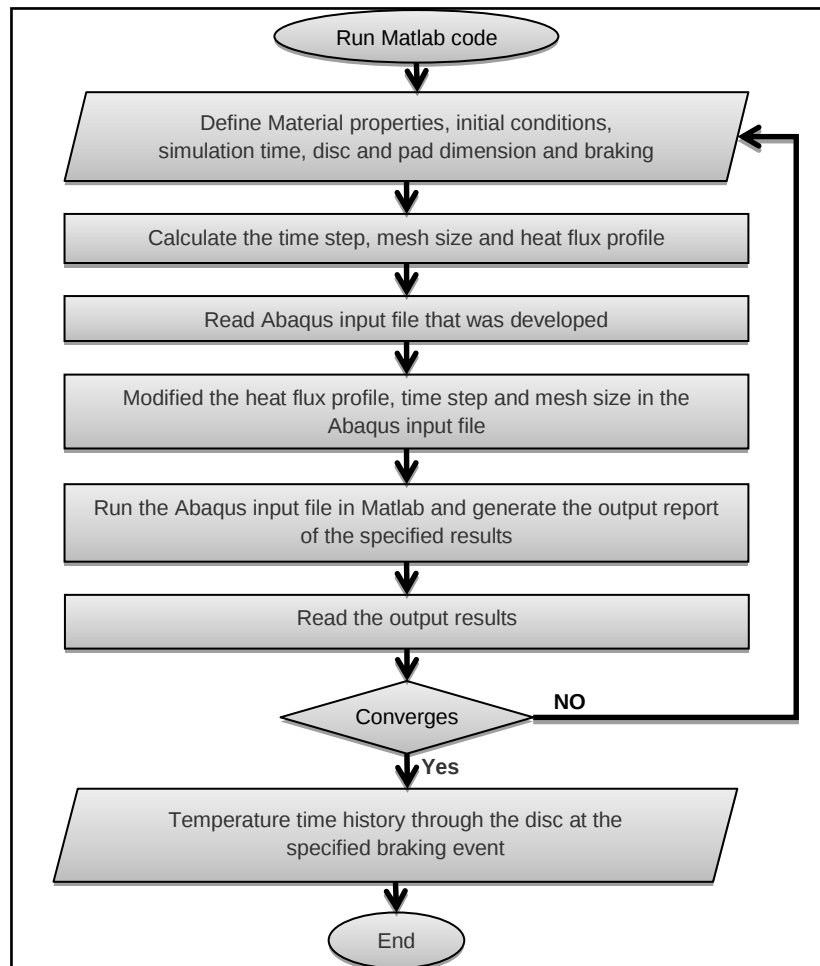


Figure 13. The overall Matlab/Abaqus model structure.

The scaling methodology (Bozic et al., 2012) was used to investigate the thermal performance of several different small scale disc brake using the data in Table 1 and 3. Four repetitive braking stops were performed using the algorithm described in Figure 13 for different small scale disc brake materials to investigate the surface temperature of the disc with typical results as shown in Figure 14. These results showed that for the same braking conditions the surface temperature of the Al-MMC rotor is higher than the surface temperature for the steel and grey cast iron rotors which was as expected.

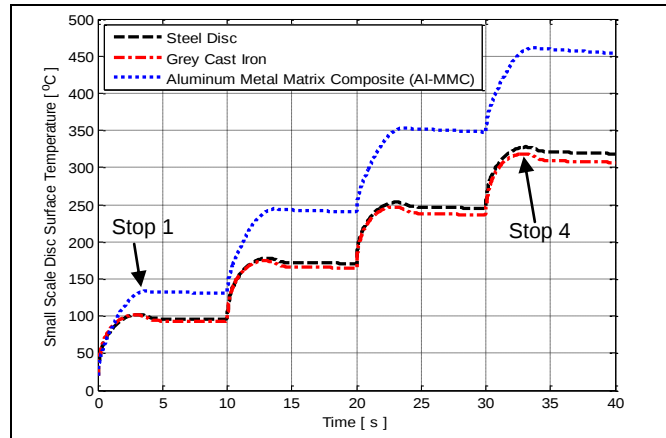


Figure 14. Surface disc temperature for different small scale disc brake materials.

6. Conclusion and future works

One dimensional thermal models of an automotive disc brake were developed using both Matlab and Abaqus. The 1D Matlab thermal models were validated and compared with the results obtained from the literature and Abaqus. Overall it was found that the Matlab results showed good agreement with Abaqus and the literature results. Abaqus models were used to investigate the coating layer modelling. The thermal resistance modelling approach without physically modelling the thin coating layer reduces the analysis time up to 20-25% and also reduces the memory needed for such analysis. This approach will of importance when modelling a coated three dimensional in Abaqus because it will save much time and reduce the model size. Despite the thermal resistance approach to modelling a thin coating layer giving very good results, particularly when using convection tuning, it needs more investigation before it can be implemented.

Furthermore, the combined Abaqus\Matlab interaction approach for solving the disc brake finite element models has several advantages:

- The new approach analysis used is significantly faster. The time to perform a typical brake disc analysis has been reduced, by up to 60-65%.
- The capabilities of using various braking events such as: drag brake, repetitive brake, alpine test... etc, are apparent with the combined approach

In addition the interaction approach will be used in future to perform optimisation analysis utilising the optimisation toolbox in Matlab. Furthermore, coupled braking analysis will be carried out to investigate the thermal performance of different disc and pad materials in order to investigate the possibility of replacing the conventional cast iron with lighter disc material. Finally, a DOE approach will be used in conjunction with a sensitivity analysis to investigate the effect of various parameters on the thermal performance of a solid disc brake using the proposed interaction approach.

7. References

1. Abaqus Users Manual, Version 6.8. Dassault Systemes Simulia Corp., Providence, RI.
2. *Aluminium Alloy // Commercial Alloy // 6082 - T6~T651 datasheet* [Online]. Available: http://www.aalco.co.uk/datasheets/Aluminium-Alloy_6082-T6~T651_148.ashx [Accessed 04/10 2012].
3. *AMC640xa data sheet* [Online]. Available: http://www.amc-mmc.co.uk/640xa_data.htm [Accessed 04/10 2012].
4. *Federal-Mogul eCatalog* [Online]. Available: <http://www.fmecat.eu/index-car.asp> [Accessed 04/10 2012].
5. ABBAS, S. A., CUBITT, N. J. & HOOKE, C. J. 1969-70. Temperature distributions in disc brakes. *Proc. Instn Mech. Engrs*, **184**(2A)(9), 185-194.
6. ADAMOWICZ, A. & GRZES, P. 2011. Analysis of disc brake temperature distribution during single braking under non-axisymmetric load. *Applied Thermal Engineering*, **31**, 1003-1012.
7. ALSAIF, M. A., DAHM, K. L., SHRESTHA, S., DEARNLEY, P. A. & BARTON, D. C. Year. Plasma Electrolytic Oxidation (PEO) treated aluminium metal matrix composite rotors for lightweight automotive brakes. *In: 6th European Conference on Braking*, 2010 France.
8. AMIN, A. M., SERI, S. M. & RAGHAVAN, V. R. Year. Study of the temperature distribution in disc brakes by the method of order-of-magnitude analysis *In: Regional Conference on Engineering Mathematics, Mechanics, Manufacturing & Architecture*, 2007.
9. BOZIC, G., ALNAQI, A., BROOKS, P. & BARTON, D. Year. Thermal Characterisation of Lightweight Brake Rotors for Passenger Car Application Using A small Scale Brake Dynamometer and One Dimensional Model. *In: Eurobrake 2012*, 2012 Germany.
10. CURRAN, J. A. & CLYNE, T. W. 2005. The thermal conductivity of plasma electrolytic oxide coatings on aluminium and magnesium. *Surface & Coatings Technology*, **199**, 177-183.
11. DAHM, K. L., BLACK, A. J., SHRESTHA, S. & DEARNLEY, P. A. Year. Plasma Electrolytic Oxidation treatment of aluminium alloys for lightweight disc brake rotors. *In: IMechE Conference on Braking*, 2009 York (UK). 53-60.
12. EL-SHARKAWY, A. E. Year. Transient One-dimensional Thermal Analysis of Automotive Components for Determination of Thermal Protection Requirements. *In: SAE Technical paper*, 2008-01-0733, 2008.
13. GRIEVE, D. G., BARTON, D. C., CROLLA, D. A. & BUCKINGHAM, J. T. 1997. Design of a lightweight automotive brake disc using finite element and Taguchi techniques. *Proceedings of the Institution of Mechanical Engineers. Part D: Journal of Automobile Engineering*, **212**(4), 245-254.

14. JUN, K. J., PARK, T. W., JUNG, S. P., LEE, S. H. & YOON, J. W. 2008. Development of a numerical method to predict the alpine test result. *Proceedings of the Institution of Mechanical Engineers Part D-Journal of Automobile Engineering*, 222, 1841-1849.
15. KERMC, M., KALIN, M., VIZINTIN, J. & STADLER, Z. 2005. A reduced-scale testing machine for tribological evaluation of brake materials. *Life Cycle Tribology*, 799-806.
16. KIM, M. R., AHN, B. K., LEE, J. M. & JUNG, Y. K. Year. Numerical Investigation of Thermal Behavior in Brake Assembly During the ALPINE braking Mode. In: SAE Technical paper, 2007-01-1021, 2007.
17. LIMPET, R. 1999. *Brake design and safety*, Warrendale, Penn, Society of Automotive Engineers.
18. LINCK, V., BAYADA, G., BAILLET, L., SASSI, T. & SABIL, J. 2005. Finite element analysis of a contact with friction between an elastic body and a thin soft layer. *Journal of Tribology-Transactions of the Asme*, 127, 461-468.
19. MCPHEE, A. D. & JOHNSON, D. A. 2008. Experimental heat transfer and flow analysis of a vented brake rotor. *International Journal of Thermal Sciences*, 47, 458-467.
20. NEWCOMB, T. P. 1960. Temperatures reached in disc brakes *J. Mech. Engng Sci*, 2 (3), 167-177.
21. SHERIDAN, D. C., KUTCHEY, J. A. & SAMIE, F. Year. Approaches to the Thermal Modelling of Disc Brakes. In: SAE Technical Paper, 880256, 1988.
22. SHRESTHA, S. & DUNN, B. D. 2007. Advanced plasma electrolytic oxidation treatment for protection of light weight materials and structures in a space environment *Advanced Surface Treatment*, 40-44.
23. SHRESTHA, S., MERSTALLINGER, A., SICKERT, D. & DUNN, B. D. 2003. Some preliminary evaluations of black coating on aluminium AA2219 alloy produced by plasma electrolytic oxidation (PEO) process for space applications. In: FLETCHER, K. (ed.) *Proceedings of the 9th International Symposium on Materials in a Space Environment*.
24. TALATI, F. & JALALIFAR, S. 2009. Analysis of heat conduction in a disk brake system. *Heat and Mass Transfer*, 45, 1047-1059.