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A NEW PARADIGM FOR DISC-PAD INTERFACE MODELS IN FRICTION BRAKE SYSTEM

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KEYWORDS – finite element method, heat transfer, interface conductance, partition ratio, pressure distribution

ABSTRACT
In this paper a 2D coupled thermal-stress finite element model is established and used to predict thermal phenomena at the disc-pad interface of a disc brake system. The importance of certain critical settings and parameters for the 2D FE model has been identified (such as, a limited degree of freedom for a brake pad in place of accepted practice that considers uniform contact), here a non-uniform pressure distribution resulting from friction bending moment effects due to the introduction of a pivot point. These parameters affect the distributions of both interface temperature and pressure. The simulation results show that when the interface conductance h is 10^6 W/m^{2}K or higher, the interface temperature distribution is no longer sensitive to friction bending moment effects. However, when h is 30000 W/m^{2}K or lower, the interface temperature distribution and heat partition ratio are significantly affected by the setting used for the rotational degree of freedom of the pad. The simulation results provide a useful reference for a better design of a disc brake system for different applications.

1. INTRODUCTION

With the development of science and technology in automotive engineering, a general trend is that the maximum speed of vehicles is increasing. A consequence of this is how to maintain the ability to reduce the speed to a safe level within an effective time and therefore proper better the braking system becomes imperative. In a disc braking event, sliding contact between pads and disc is generated by physical interaction, and consequently friction heat is generated at the pad/disc contact interface. High deceleration of a typical passenger vehicle is known to generate interface temperatures as high as 900 ˚C in a fraction of a second (1). This high temperature during braking can cause lots of problems, such as premature wear, brake fluid vaporization, thermal cracks, thermally-excited vibration, thermo-elastic instability, and even brake failure. In order to study the temperatures it is necessary to determine the thermal dissipation, heat partition ratio, and the effects of an interface tribo-layer (ITL) developed in the friction braking system. The ITL existing at the disc-pad interface makes the two contact surfaces not contact directly, which results in a non-infinite thermal contact conductance, h, at the disc-pad interface. The magnitude of contact conductance is affected by the nature of the ITL, the interface gap/clearance, as well as the actual contact area and actual contact pressure distribution at the disc-pad interface. In the literature, h value of 10k-30k W/m^{2}K are mentioned or used (2).
In this work the effects of several critical parameters used in the 2D FE model have been studied and we consider the effect of different assumptions in terms of the rotational degrees of freedom of the pad on the contact pressure and temperature distribution.

2. BASIC 2D FE THERMAL-STRESS COUPLED MODEL

2.1 <u>2D FE model setup</u>

Figure 1 illustrates the geometry and key features of the two-dimensional dynamic model including the pad and disc. The pad consists of two components: a back plate and a pad lining. Material properties of the three components used in the models are listed in Table 1(3). As a 2D FE model, every solid part has three degrees of freedom. U1 (horizontal, along the x-axis), U2 (vertical, along the y-axis), and UR3 (rotation about the z-axis) and some degrees of freedom for each part are constrained or eliminated at different modelling steps as detailed in Section 2.2.

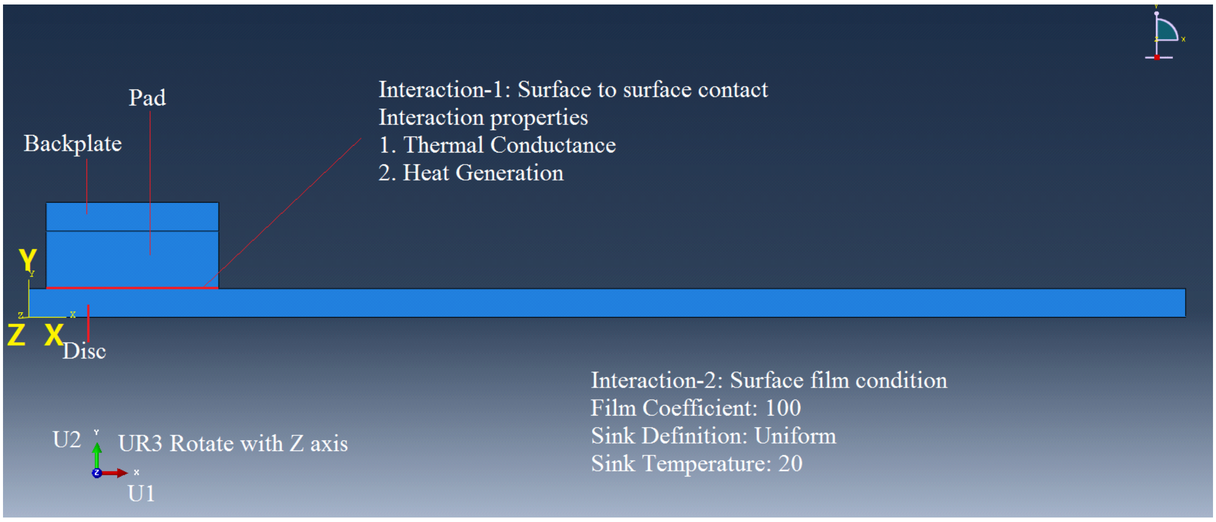


Figure 1 Basic 2D FE model setup

A plane-strain type of element, CPE4T, is used for the present 2D analysis, which assumes that the pad and disc have infinite length in U3 or z-axis direction. CPE4T is a 4-node plane strain element in ABAQUS (thermally coupled quadrilateral, bilinear displacement and temperature) (4). The element size of the basic model used is 1×1 mm initially, and the effect of element size is discussed in Section 3.4. As a result a suitable element size setting is recommended.

There are two interactions settings in the 2D models. The first one is the interaction between the surrounding environment and the two solid part (disc and pad). The convection coefficient of 100 W/m^{2}K) is used to represent the surface film condition and a sink temperature of 20 °C is used to represent the surrounding environment temperature and the initial temperature of the pad and the disc (2).
The second interaction is 'surface to surface' friction contact interaction between the lower surface of the pad and the upper surface of the disc, as shown in Figure 1, which is realized by four settings; normal behavior, tangential behavior, thermal contact conductance, and heat generation.

- Normal behavior: normal behavior is defined by a type of 'contact'. The 'Hard' contact pressure-over-closure is selected, which means that the classical Lagrange multipliers method of constraint enforcement is used in an ABAQUS/Standard analysis (and penalty contact enforcement in an ABAQUS/Explicit analysis) (4).

- Tangential behavior: tangential behavior is defined by a friction formulation, in which a "penalty" method is used, that permits some relative motion of the surfaces when they are slipping (4). When the surfaces are sticking, the magnitude of sliding is limited to the elastic slip. A friction coefficient of 0.4 is used.

	Disc	Pad	Backplate
Thermal Conductivity *k* (W/mK)	48	2.06	41.5
Density ρ (kg/m^3)	7800	2580	7800
Young's Modulus (Pa)	2.09E11	1.25E9	2.1E11
Poisson's Ratio	0.3	0.3	0.34
Thermal Expansion (m/mK)	1.1E-5	1.43E-5	1.1E-5
Specific Heat c (J/kgK)	452	749	480

Table 1 Material properties (3)

Conductance (W/m^{2}K)	Clearance (μm)
h (nominal value)	0
100	5
0	10

Table 2 Setting up interface conductance and clearance

- Interface thermal conductance and clearance: interface thermal conductance is defined as the heat conduction between the lower surface of the pad and the upper surface of the disc as shown in Figure 1. Clearance (δ) in ABAQUS is defined as the distance separating two contact surfaces. The thermal conductance can be defined as a function of the clearance, i.e. as the clearance changes from 0 to δ, the corresponding thermal conductance changes from its nominal value h to 0. It is known that the main factor influencing the existence of a clearance (or gap) between the pad and disc contact surfaces is the surface roughness of the disc and pad contact surfaces, R_a. R_a is normally around 5 to 20 μm (5). In this study, therefore, the effective clearance between the pad and disc contact surfaces is assumed to lie between 0 and 10μm, as shown in Table 2. When the gap is 0, the thermal conductance at the two contact surfaces reaches its nominal value h (W/m^{2}K); if the gap is larger than 10μm, there is no heat transfer, and consequently the conductance approaches zero (and the thermal resistance becomes infinite). When the gap is larger than 5μm, the heat is assumed to dissipate to the surrounding air, and the convective coefficient of 100 W/m^{2}K is applied (6). The effect of the thermal conductance *h* on simulation results is discussed in detailed in Section 3.1.

- Heat generation: the heat generation factor, *f*, in ABAQUS is used to define the fraction of friction induced heat energy initiated at the slave surface (the lower surface of the pad in the present 2D model). When f=1, 100% of the heat energy is initiated on the pad side; when f =0, all heat energy is initiated on the disc (master) side; and when f=0.5 the friction heat energy is initiated equally on both sides. Normally f=0.5 is used. The effect of the heat generation factor on FE simulation results is discussed in detail in Section 3.3.

2.2 <u>Setting up the time sequence</u>

The model is defined by a sequence of three analysis steps:

- Initial Step: model boundary conditions, pre-defined fields, and interactions applicable at the beginning of the simulation are defined, e.g. the pad has one translational degree of freedom (i.e. the pad is allowed to move in the U2 direction) and the disc has no degree of freedom.

- Step One (static): the normal load is applied, i.e. a uniform 10 MPa pressure is applied to the top of the pad's back plate, while other boundary conditions are propagated from the Initial Step.

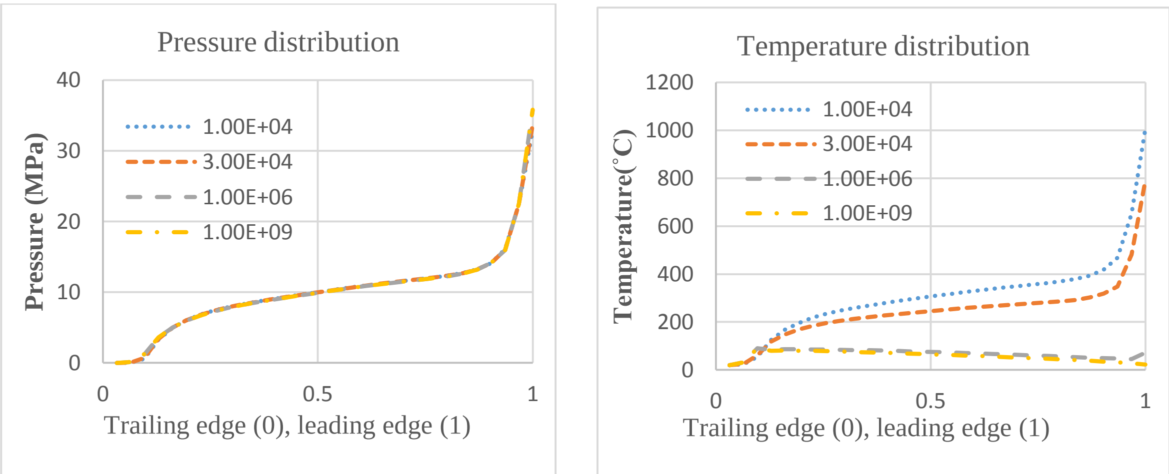
- Step Two (coupled temperature-displacement action): both of the boundary conditions on the disc and the pad are changed from Step One. The disc is allowed one translational degree of freedom in U1 (x-axis) and a velocity of −5m/s is applied to the disc. The time period for Step Two is 33ms, during which the disc is sliding from right to left with respect to the pad, as shown in Figure 1. The time increment for simulation is fixed at 0.0001s. As far as the pad is concerned, the pad is allowed to freely rotate (FR) about its top left corner, i.e. the rotation pivot A (RP-A). How this rotational degree of freedom affects simulation results is discussed in detail in Section 3.2.

3. TECHNICAL ISSUES IN BETTER DESIGN OF 2D FR MODEL

During the development of the 2D FE coupled thermal-stress model, several issues/factors are encountered, that affect the quality of the model simulation results. The main issues investigated here are the interface thermal conductance, the pad's rotational degree of freedom, the heat generation, and the element size. Consequently, some recommendations are provided for the better design of the 2D FE coupled thermal-stress model. All results presented in Section 3 are measured at the time instant t = 33 ms (i.e. at the end time of the simulation).

3.1 Interface thermal conductance

The contact thermal conductance, *h*, due to the existence of an ITL at the disc-pad interface and its effect on the interface heat transfer are investigated. It is difficult to actually measure or quantify this factor due to its complex nature. Lee (2) and Loizou (6) used value of *h* equal to $3\text{x}10^4$ W/m^{2}K and 10^{15} W/m^{2}K, respectively, in their disc-pad thermal analyses. Values of 10^4 W/m^{2}K, $3\text{x}10^4$ W/m^{2}K, 10^6 W/m^{2}K, and 10^9 W/m^{2}K are selected for this factor analysis. The other factors/settings, which are kept constant, are: pad is FR at RP-A, f=0.5, and the size of the element is 1×1 mm.



(a) (b)

Figure 2 (a) Pressure distributions and (b) temperature distributions on the pad contact surface (with four different thermal conductance values)

Conductance (W/m^{2}K)	Average Temperature (˚C)	Maximum Temperature (˚C)	Average Pressure (MPa)	Maximum Pressure (MPa)
10^4	313.10	1006.00	10.19	32.86
$3\text{x}10^4$	248.88	790.50	10.21	33.70
10^6	67.82	86.11	10.26	35.73
10^9	57.50	81.35	10.26	35.82

Table 3 Average/maximum temperature and pressure values at pad's contact surface under different conductance values

Figure 2 shows the pressure and temperature distributions on the pad's contact surface with different conductance values, and Table 3 shows the average/maximum temperature and pressure values at the pad's contact surface. It is clear that changes in the thermal conductance has little effect on the contact pressure, in terms of average/maximum pressure and pressure distribution. The average pressure is around 10MPa, consistent with the uniform pressure applied to the pad's upper surface.

The pad contact surface temperature, however, is significantly affected by the thermal conductance, in terms of average/maximum value as well as distribution. When the conductance values are 10^4 W/m^{2}K and 3x10^4 W/m^{2}K, as shown in Figure 2(b), the temperature increases dramatically from the pad's trailing edge to its leading edge. When the conductance values are at high values of 10^6 W/m^{2}K and 10^9 W/m^{2}K, however, the temperature tends to decrease slightly from the pad's trailing edge to its leading edge. The average/maximum temperatures are shown in Table 3, and the distributions become less sensitive to the conductance value at high conductance conditions. Modeling incorporating higher conductance requires much smaller time increments, which in turn means longer and unnecessary computing time. It is, therefore, recommended to use 10^6 W/m^{2}K to represent a high conductance scenario, and 10^4 W/m^{2}K to represent normal friction braking conditions (a 'benchmark' value that has been used in the literature (2)) for this type of disc-pad FE thermal analysis. It is clear that proper consideration of the interface thermal conductance in FE modelling/simulation of friction braking process is very necessary. It is not acceptable to either assume no thermal contact resistance (where h=∞) or assume no heat exchange (where h=0) at disc-pad interface.

3.2 Rotational degree of freedom of the pad

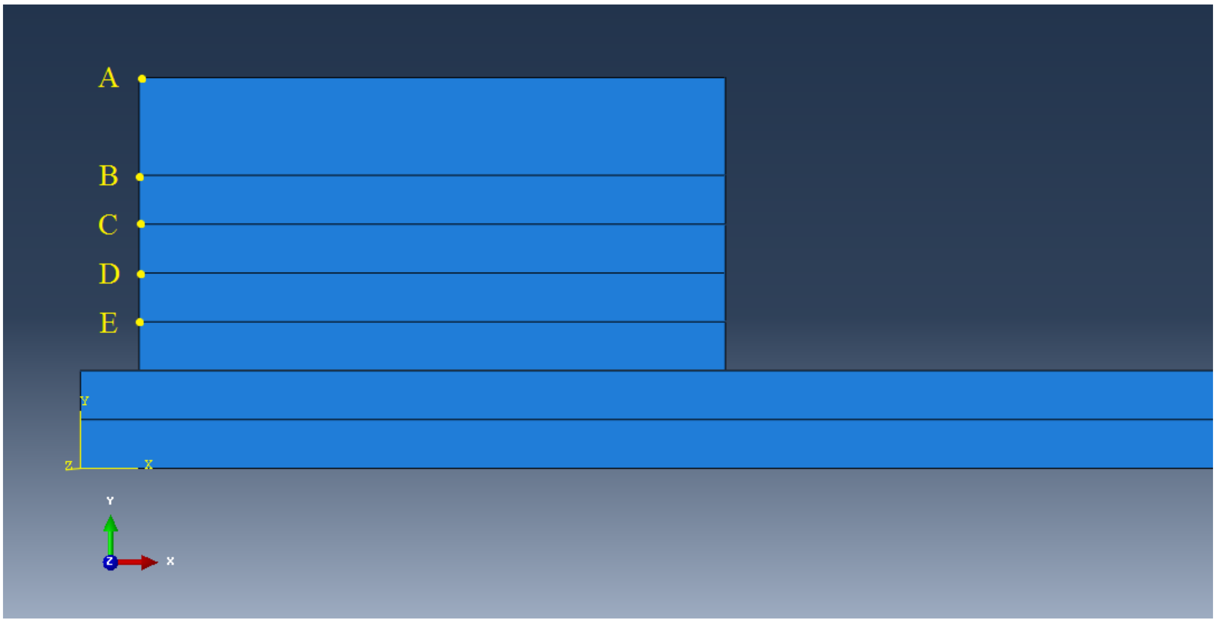
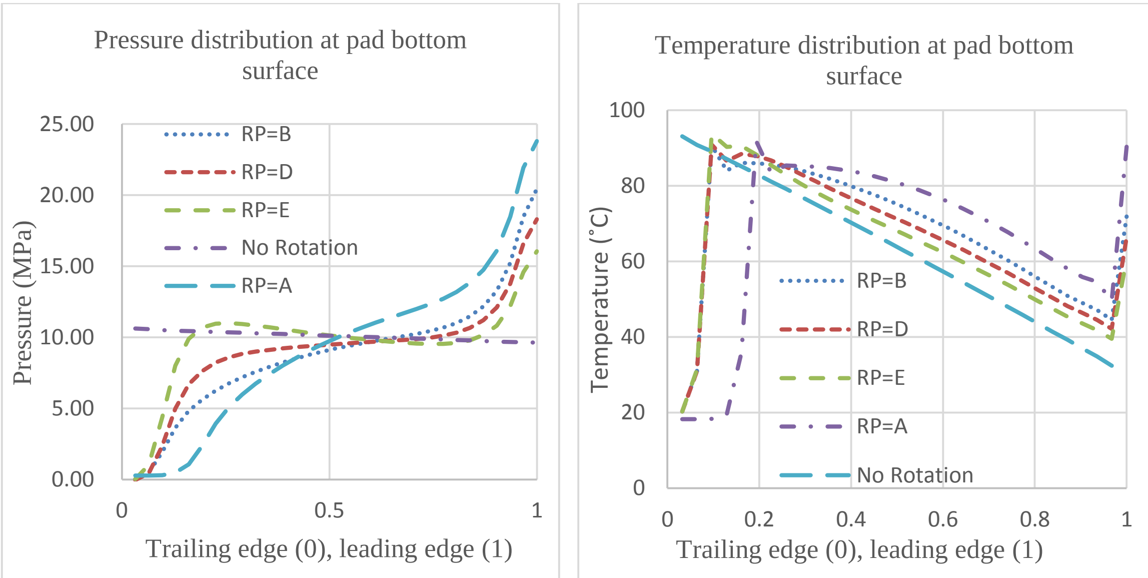


Figure 3 Basic 2D model with pivots A to E along the pad's trailing edge

In the literature there are two different boundary conditions commonly used in FE modelling of the friction pad. One is No-Rotation (NR), i.e. keeping the pad with no rotation degree of freedom about the z-axis, which ignores the friction induced bending moment effect. The other is Free-Rotation (FR), i.e. allowing the pad to rotate freely about the z-axis. A knowledge limitation is that arguably no of these two conditions represent the dynamic behavior of brake pads in a real braking event. The pad normally has limited rotational degree of freedom, i.e. it can rotate slightly about the z-axis due to the friction/reaction forces and clearance fit between the pad and the caliper in a braking event, and consequently the disc-pad interface contact pressure/temperature distributions could be affected. To study this effect, different pivot points are selected (A, B, C, D, E) along the pad's trailing edge, as shown in Figure 3, respectively, to represent the different levels of the friction induced moment effects upon the FE simulation. Other factors/settings kept constant are: f=0.5, size of the element is 1×1 mm, and the interface conductance values of 10^4 W/m^{2}K and 10^6 W/m^{2}K.

Figure 4 shows the pressure and temperature distributions on the pad's contact surface when the RP of the pad is at point A, B, D or E respectively, as defined in Figure 3.



(a) (b)

Figure 4 (a) Pressure distribution and (b) temperature distribution at the pad contact surface under different pad boundary conditions, when 10^6 W/m^{2}K (RP=A,B,D,E, and No Rotation)

The contact pressure, shown in Figure 4 (a), is significantly affected by the RP point, in terms of maximum/minimum value as well as distribution. Under RP-A, the contact pressure increases dramatically from 0 MPa at the pad's trailing edge to 25 MPa at the pad's leading edge. Under NR conditions, however, the contact pressure is uniformly distributed along the disc-pad contact interface. The average pressure for all cases is similar at around 10 MPa, consisted with the uniform pressure applied on the pad's upper surface.

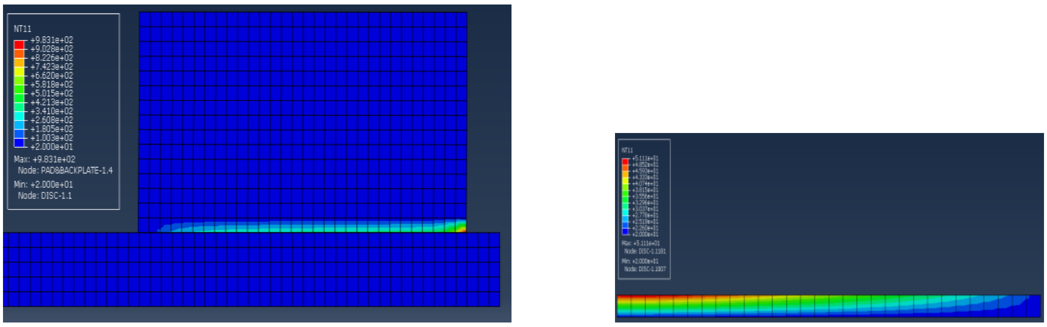
The change of the RP point, however, has a limited effect on the temperature in terms of average/maximum temperature and temperature distribution, as shown in Figure 4 (b).

It is clear that proper setting of this boundary condition in FE modelling/simulation of friction braking process is very necessary. It is not acceptable to either assume NR or assume FR. It is suggested to use RP-D to represent a limited friction induced moment in disc-pad FE thermal-stress analysis.

3.3 <u>Heat generation *f*</u>

It is known that friction induced heat is generated mainly at the pad side instead of the disc side due to the fact that the pad is softer and deforms in a more plastic/elastic mode than the disc during a braking event (3). Thus it is expected that the f value should be larger than 0.5 and smaller than 1. In addition, the existence of an ITL between the two contact surfaces means that it is possible for the contact surface temperature at the pad side to be not equal to that at the disc side. The f value is expected to affect the contact surface temperature of the disc (and that of the pad), and the heat partition ratio. In this analysis, values *f*=0.5, 0.75, and 1 are used.

To magnify the effect of *f* value on the contact surface temperatures at the disc-pad interface, the thermal conductance value h is set to 0 W/m^{2}K, which means that there is no heat exchange at the interface. The pad boundary condition is FR-A, and the element size is 1×1 mm.

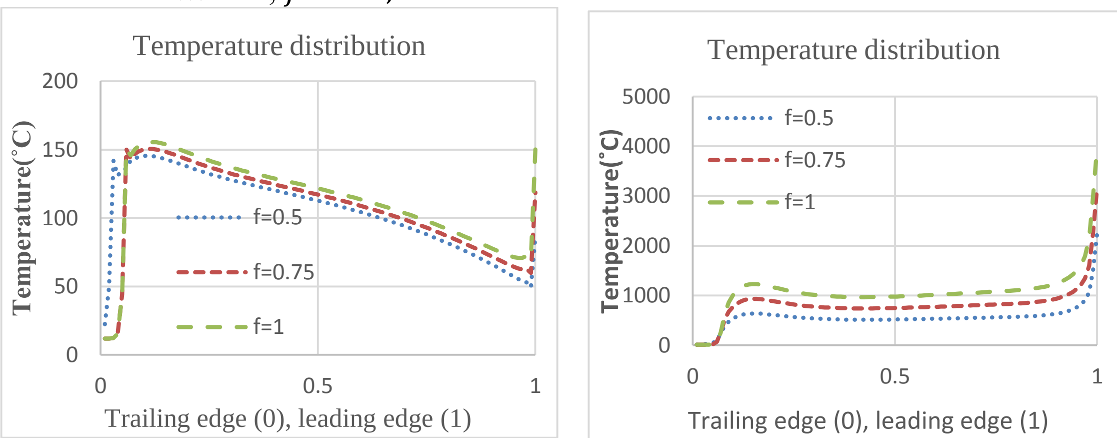


(a) (b)

Figure 5 (a) Results of 2D model with *f*=0.5, *h*=0; (b) Temperature distribution at the disc upper surface

Figure 5 shows that, since *f*=0.5 implies that half of the friction heat is generated at the pad side and half at the disc side, and as expected, a temperature increase is observed on both pad and disc contact surfaces during the friction braking event. Approximately the pad maximum temperature increases from 20°C to 900°C, and the disc's maximum temperature increases from 20 °C to 50°C. For further analysis of the combined effect of *f* and *h*, $h = 10^4$ W/m^{2}K and $h = 10^6$ W/m^{2}K are used. Results are shown in Figure 6 and Table 4.

- $h = 10^6$ W/m^{2}K, $f = 0.5, 0.75$ and 1



(a) (b)

Figure 6 Pad contact surface temperature distribution $f = 0.5$, 0.75, and 1, (a) $h = 10^6$W/m^{2}K, (b) $h = 10^4$W/m^{2}K

Conductance (W/m^{2}K)	Temperature		
	$f = 0.5$	$f = 0.75$	$f = 1$
10^6	113.09	117.54	121.98
10^4	517.08	747.11	977.18

Table 4 Temperature at the mid-point of the pad's contact surface

Figure 6 (a) shows the difference in heat generation with different values of f. The temperature is low, with a maximum of 160 °C at the pad's trailing edge (around 0 to 0.1 at the x-axis). For the models with $f = 0.75$ and $f = 1$ the temperature distributions are similar to those for $f = 0.5$ however, the temperature is slightly lower. Excluding the trailing edge part, the temperature distributions of all three models are similar and the difference between them is only around 3.8% as shown in Table 4.

- $h = 10^4$ W/m^{2}K, $f = 0.5$, 0.75, and 1

Figure 6 (b) shows that when the conductance is lower than 10^4 W/m^{2}K, the temperature at the pad's lower surface becomes very high (over 500 °C) and the temperature difference between them is always over 30%, as shown in Table 4.

It is clear that properly setting the *f* value in FE modelling/simulation of friction braking is very necessary. The *f* value of 0.75 is probably the most appropriate representation of the heat generation situation at the disc-pad interface in braking processes and, therefore, is recommended for this type of disc-pad FE thermal-stress analysis.

3.4 <u>Element size</u>

It is known that smaller element sizes normally mean more accurate simulation results but with longer computing time. An appropriate element size can provide reliable results in a reasonable time. For realizing an effective FE analysis, the effect of the element size is studied. Different element sizes, i.e. 1×1 mm, 0.5×0.5 mm, 0.25×0.25 mm and 0.2×0.2 mm are used in the study.

Other factors/settings kept as constant are: pad FR at RP-A, h = 10^6 W/m^{2}k, and *f* = 0.5.

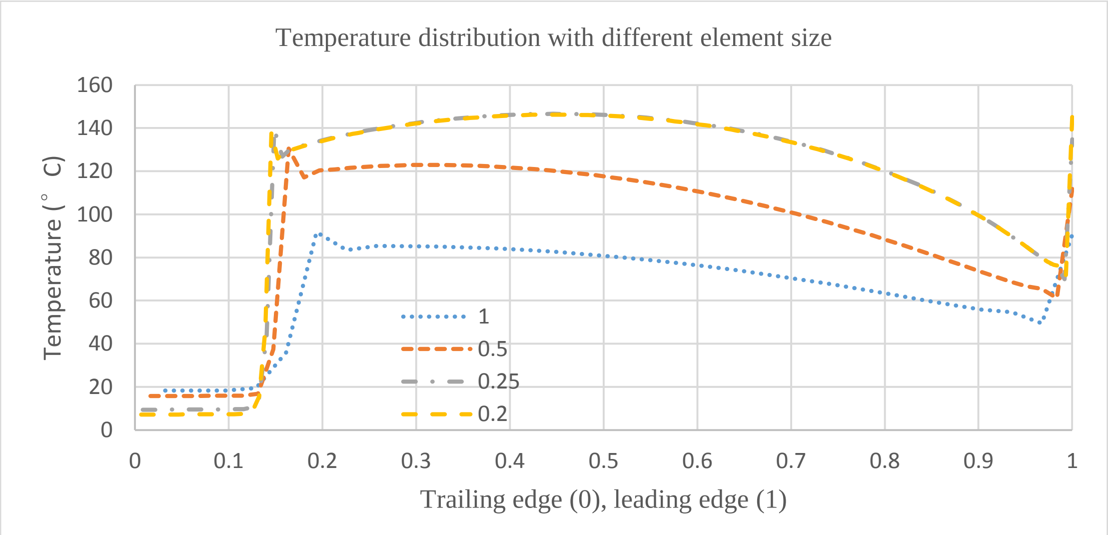


Figure 7 Simulation results of the temperature distributions on the pad contact surface using different element size respectively (i.e. 1mm, 0.5mm, 0.25mm and 0.2mm)

Figure 7 shows plots of the pad contact surface temperature distribution from the pad's trailing edge to its leading edge as a function of the element size. This figure shows that the element size does affect temperature distribution predictions, i.e. the highest temperature region moves from the trailing edge to the middle area as the element size decrease. It also affects the value of the maximum temperature. It clear that when the element size changes from 0.25mm to 0.2mm, the distribution curve is no longer changing significantly. An element size of around 0.25mm is, therefore, preferred. In terms of computing time, one simulation with 0.25mm element takes of the order of eight hours.

Element Size (mm)	1	0.4	0.3	0.25	0.25-1
Computational Time	t	12t	15.6t	24t	10.8t

Table 5 Estimated computing time consumed when using different element size

To reduce the computing time, a non-uniform meshing strategy is adopted, which is based on the understanding that the high variation in terms of the values of stress/strain and temperature is expected to occur only at location adjacent/close to the interfaces. Therefore a fine mesh is created only in regions close to the interfaces. A non-uniform 0.25-1 mm mesh is set up, in which the element size next to the interface is 0.25mm and the element size next to the upper side of the pad (as well as the lower side of the disc) is 1mm. Some additional tests are carried out using element sizes of 0.4mm, 0.3mm, 0.25mm, and non-uniform 0.25-1 mm, respectively. Table 5 shows the comparative computing times. The computational time under non-uniform 0.25-1 mm meshing is less than 4 hours for a single simulation, reduced from 8 hours under uniform 0.25 mm meshing. A non-uniform 0.25-1 mm mesh is, therefore, recommended for this type of 2D FE thermal analysis.

4. ANALYSIS OF HEAT PARTITION RATIO

The heat partition ratio, a term introduced in 1937 by Blok (7), is an important measure used in friction induced heat transfer problems, which indicate how the friction heat is dissipated into the contact pair (as well as to its surrounding environment). Pereverzevam and Balakin (8) and Yevtushenko *et al*. (9, 10) pointed out that one of the input parameters for FEM temperature calculations in pad/disc brake systems is the heat partition ratio. The friction induced heat power per unit area of friction surface is

$$q = q_1 + q_2 = \mu v p \qquad [1]$$

μ denotes the coefficient of friction, v is the relative velocity of the sliding bodies, p is the contact pressure, q_1 is the heat dissipated into the pad part and q_2 is the heat dissipated into the disc part. The heat partition ratio γ is defined as

$$q_1 = \gamma q \text{ and } q_2 = (1-\gamma)q \qquad [2]$$

One of the typical formulas used to calculate the heat partition ratio in braking system is Charron's formula, shown in Equation [3]:

$$\gamma = \frac{\sqrt{K_1\rho_1 c_1}}{\sqrt{K_1\rho_1 c_1}+\sqrt{K_2\rho_2 c_2}} \qquad [3]$$

Based on Equation [3], the heat partition ratio is a constant if the material thermal properties for the friction pair (i.e. density ρ, specific heat capacity c, and thermal conductivity K) are constant. As pointed out by Loizou (3), however, it is debatable whether a constant heat partition ratio uniformly distributed along the interface should be used in a thermal-stress analysis of a conventional braking event. By using the 2D models developed in this research, the heat partition ratio at the pad/disc interface during a friction braking is simulated and analysed.

4.1 Model conditions for the heat partition ratio analysis

Based on the analysis results in Section 3, the conditions selected for analysis of the heat partition ratio are summarized in Table 6.

Setup	Content
Rotate Pivot	Point D
Element Size (mm)	0.25-1(non-uniform)
***h* (W/m²K)**	10^4, 10^6
f	0.75

Table 6 2D model settings for the heat partition analysis

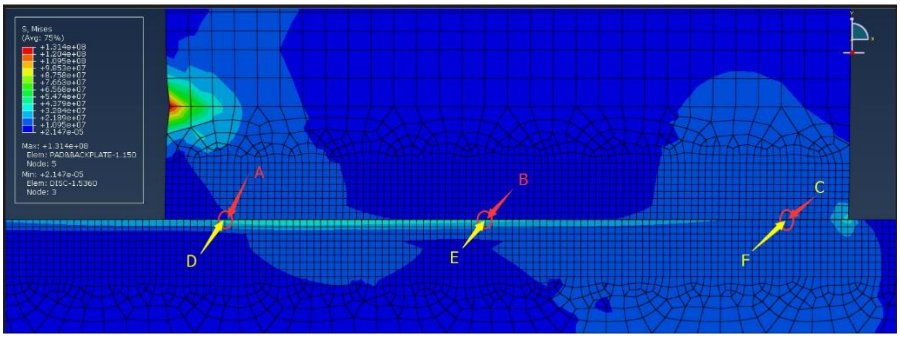


Figure 8 Six single nodes for calculating partition ratio

For calculation of the heat partition ratios at the contact interface, three contact pairs of nodes are selected: nodes A&D, B&E, and C&F, as shown in Figure 8. The three nodes A, B and C are at the contact surface of the pad and the three nodes D, E and F are at the contact surface of the disc. At the start of the simulation, there is no deformation in the x-direction, so the nodes D, E and F on the disc surface have the same coordinates as nodes A, B, and C on the counterpart of the pad surface, respectively. During the simulation, however, the nodes on the disc surface do not have exactly the same coordinates as the corresponding nodes on the pad surface because of elastic deformations of the pad and the disc, in the x-direction. Therefore, the nearest nodes (D, E and F) are chosen on the disc's contact surface to match nodes A, B, and C, respectively. ABAQUS/Standard provides the heat flux per unit area (HFL) across the thermal gap elements as an output. For each unique nodal point, there are three choices of HFL output: Magnitude, HFL1 and HFL2. Magnitude means heat flux per unit area leaving the surface; HFL1 means heat flux in the x-direction and HFL2 means heat flux in the y-direction (4). For the calculation of heat partition, the HFL2 is used as the output. Furthermore, HFL2 at different time instants during the simulation are selected and analysed. Each simulation takes 3400 sub-steps, and six instants have been chosen for the calculation and analysis, as shown in Table 7.

Time Step	**Description (Increment)**	**Time (ms)**
500	498	5.98
1000	998	10.98
1500	1498	15.98
2000	1998	20.98
2500	2498	25.98
3000	2998	30.98

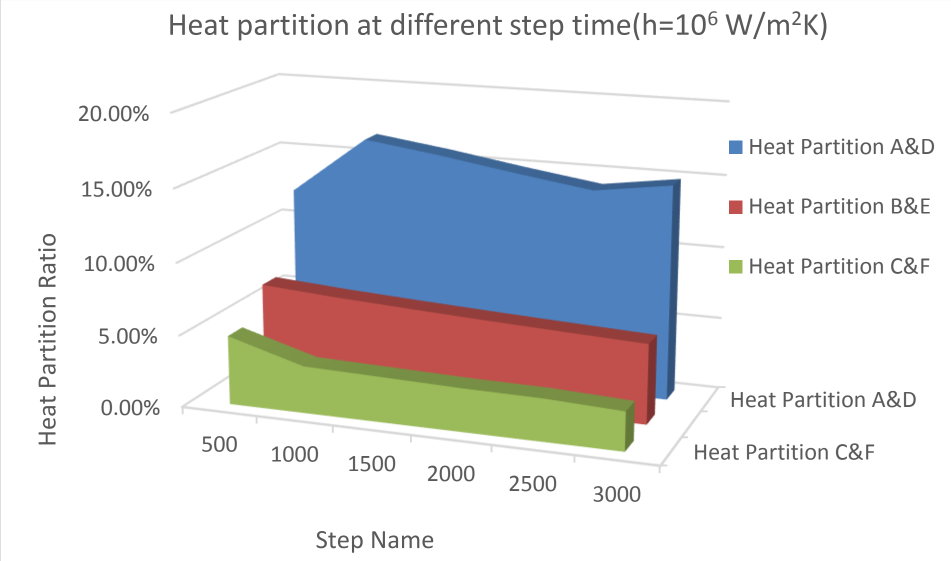
Table 7 Six time instants used for heat partition calculation

4.2 Results and analysis

We calculate the heat partition by using Equation [3], and the heat partition ratio is 13.3% based on the material thermal properties in Table 1.

4.2.1 Heat partition ratio when $h = 10^6$ W/m^{2}K

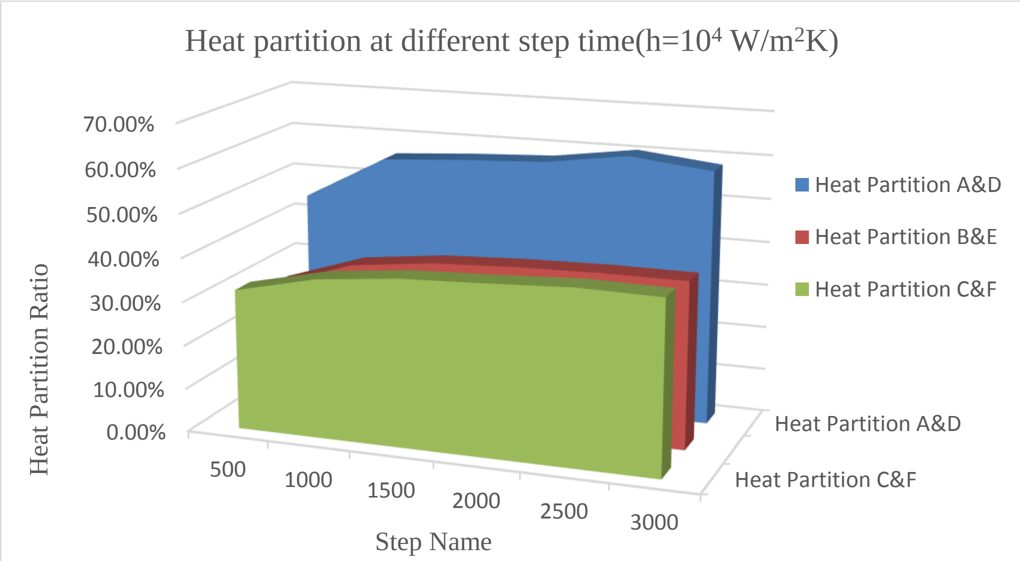
Figure 9 shows that when thermal conductance is at a high level, the heat partition decreases from the trailing edge to the leading edge, as defined in Figure 2. During the braking event, the heat partition ratio at the middle node group B&E decreases from 6.92% to 5.51% and that at the leading edges node group C&F decreases from 4.78% to 2.68%. The heat partition ratio of the trailing edge node group A&D, however, fluctuates between 12.45% and 16.47% with time, which is closer to the theoretical value of 13.3% based on Equation [3].



	500	1000	1500	2000	2500	3000
Heat Partition A&D	12.45%	16.47%	15.73%	14.88%	14.10%	14.86%
Heat Partition B&E	6.92%	6.55%	6.25%	5.98%	5.73%	5.51%
Heat Partition C&F	4.78%	3.26%	3.12%	2.98%	2.92%	2.68%

Figure 9 Comparison of heat partition of the three pairs of nodes at six time instant ($h = 10^6$ W/m^{2}K)

4.2.2 Heat partition ratio when $h = 10^4$ W/m^{2}K



	500	1000	1500	2000	2500	3000
Heat Partition A&D	45.48%	55.59%	56.81%	57.69%	60.37%	58.28%
Heat Partition B&E	28.71%	34.86%	37.01%	37.86%	38.23%	38.15%
Heat Partition C&F	32.14%	36.38%	38.27%	38.99%	39.89%	39.55%

Figure 10 Comparison of heat partition of the three pairs of nodes at six time instant (h=10^4 W/m^{2}K)

Figure 10 shows that when thermal conductance is at a level of 10^4 W/m^{2}K, the heat partition ratio is much higher than the theoretical value of 13.3%. During the braking event, the heat partition ratios increase, instead of decrease as observed under high thermal conductance conditions. Specifically, the heat partition ratio at the middle node group B&E increases from 28.71% to 38.15%, the ratio at the leading edges node group C&F increases from 32.14% to 39.55%, and the ratio at the trailing edge node group A&D increases from 45.48% and 58.28%, respectively.

In addition, the behaviour at middle and leading edges are similar, in terms of heat partition, in comparison with that at the trailing edge, no matter what *h* value is used, which indicates the indirect combined effect of the contact pressure and the interface clearance on the heat transfer at the disc/pad interface during a braking event.

Simulation results show that the heat partition ratio is not constant. It is normally high near the trailing side and low near the leading side of the pad during a friction braking process. It is also changing with contact time. Furthermore, the interface contact conductance and the contact pressure affect the heat partition ratio, in terms of its magnitude.

5. CONCLUSION AND FUTURE WORK

5.1 Technical consideration for better design of 2D FE model

It is shown clearly that some technical issues have to be addressed properly if a model developed is to provide reliable and high quality simulation results.

It is recommended that the following values should be used in the present type of FE analysis:

1) Contact thermal conductance: 10^4 to 10^6 should be considered, where 10^6 for fresh/new disc-pad contact condition, and 10^4 for well bedded-in and burnished disc-pad pair.
2) Heat generation f = 0.75 should be considered.
3) Constraint of pad: the effect of the friction induced moment has to be considered. One way to represent this effect is to control the rotation pivot point location. In this study RP-D, a pivot point located at middle on the trailing pad edge (and perpendicular to the contact interface), is recommended.
4) Finite element with 0.25-1mm non-uniform meshing and plane-strain type is recommended.

5.2 Heat partition at disc-pad interface

Simulation results from the 2D FE thermal-stress coupling analysis show that

5) The heat partition ratio is not constant. It is normally high near the trailing side and low near the leading side of the pad during a friction braking process. It is also changing with contact time.
6) Interface contact conductance affects the heat partition ratio in terms of its magnitude and distribution, which indicates that ITL has a noticeable effect on disc-pad interface temperatures.

5.3 Future work

Based on the 2D FE model established in this work, the disc-pad interface temperature distributions will be studied. A 3-D FE model based on the configuration of a scale brake test rig (in the Brake Research Lab., University of Bradford, UK) will be developed. All recommended settings from this work will be used in the construction of the 3D FE thermal-stress coupled model. And corresponding experimental investigations using the scale brake test rig will be carried out for the 3-D FE model validation and further understanding of friction braking processes.

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