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A NUMERICAL AND EXPERIMENTAL STUDY OF THE FACTORS THAT INFLUENCE HEAT PARTITIONING IN DISC BRAKES

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ABSTRACT

To investigate the heat partition on a vehicle disc brake, a small scale test rig with one contact interface was used. This allowed the disc/pad contact temperatures to be measured with fast-response foil thermocouples and a rubbing thermocouple. Based on the experimental conditions a 3D symmetric disc brake FE model has been created. Frictional heat generation was modelled using the ABAQUS finite element analysis software. The interface tribo-layer which affects heat partitioning was modelled using an equivalent thermal conductance value obtained from the authors' previous work (1). A 10 second drag braking was simulated and the history and distribution of temperature, heat flux multiplied by the nodal contact area, heat flux leaving the surface and contact pressure was recorded. Test rig and FE model temperatures were compared to evaluate the two methods. Results show that heat partitioning varies in space and time, and at the same time contact interface temperatures do not match. It is affected by the instantaneous contact pressure distribution, which tends to be higher on the pad leading edge at the inner radius side. They are also affected by the thermal contact resistance at the components contact interface.

INTRODUCTION

The Heat Partition ratio at the friction interface of a brake pad and disc is an important parameter in the study of brake disc/pad interface temperatures. Improved knowledge of causes and effects of heat partition in friction brakes will help in the understanding of brake performance in terms of disc thermal crack formation and fatigue failure, pad fade, and thermally induced friction instability effects.

In conventional braking thermal analysis the heat partition ratio is often assumed to be constant and/or uniform on the contact interface (2, 3), even though it is reported as varying in transient thermal states (4). Factors that can affect its variation include the tribo-layer formation at the interface, the contact pressure distribution and real contact area distribution. Conventional theory can assume matched contact temperatures exist across the interface by varying the heat partitioning or that the contact temperatures are not matched by having a uniform the heat partitioning. These fundamental theories have been developed by Blok (5) and Jaeger (6) in 1930's and has been followed by other researchers since then (7, 8).

In this work none of the fundamental theories is taken as grounded and the behaviour of heat generation and partitioning on disc brakes is examined.

EXPERIMENTATION

The test rig consisted one contact interface and the configuration and experimental conditions are presented in this section.

Test Rig configuration

An existing test rig facility has been modified for this project. The system configuration layout can be seen in Figure 1. These modifications consist of embedding a foil thermocouple (T1) in the rotating disc, opposed the pad surface in order to capture the disc surface temperature. A conventional K-type thermocouple (T2) was used to measure the disc back side temperature. The foil thermocouple used, was able to provide response times of the order of 1 to 5ms. A rubbing thermocouple (T3) is used to measure the disc surface temperature away from the contact interface. The ambient temperature near the disc surface is measured with a conventional K type thermocouple (T4). Three conventional K type thermocouples were embedded in the pad. This was to measure the pad leading edge (T5), trailing edge (T6) interface temperatures, and back side (T7) temperature.

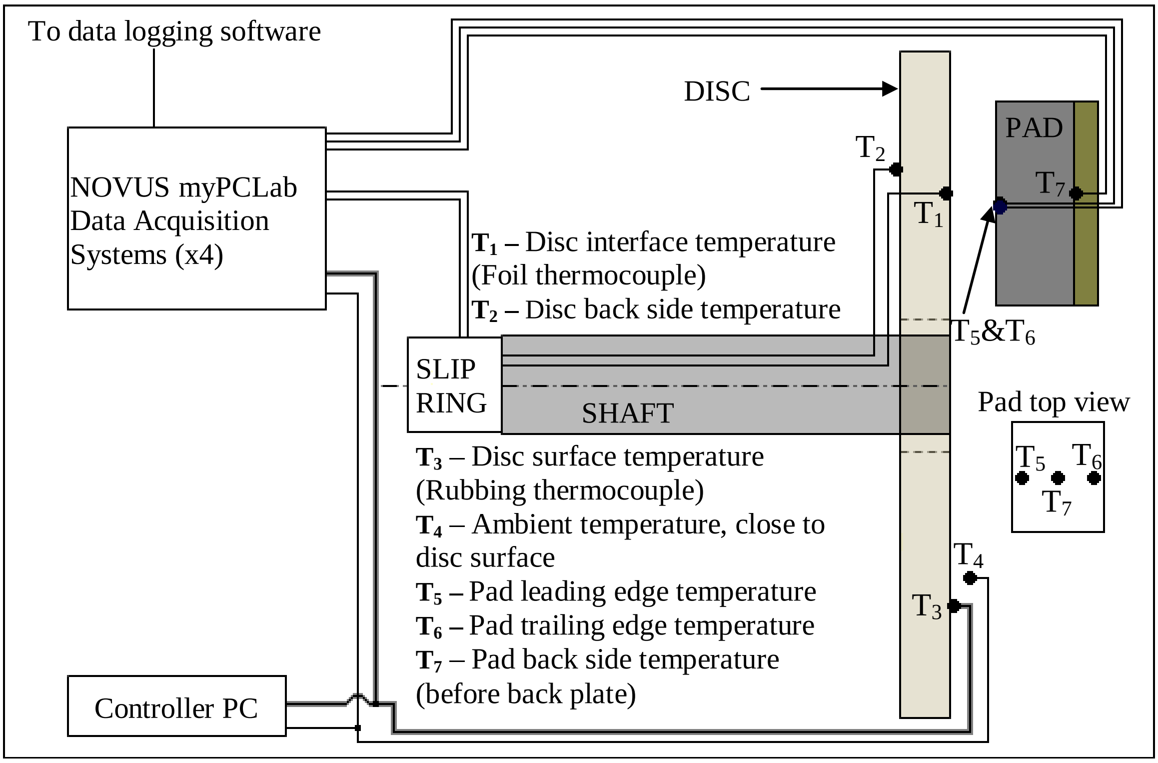

Figure 1: Test Rig Configuration

The entire system is controlled by the controller PC, which is an existing system developed previously in the university. The controller PC can control the disc rotating speed, the pad load and the actual time of application. This is a drag braking scenario as the disc speed is not reducing when the brakes are applied (thus the vehicle is not decelerating). The controller PC can also control a pre-heat and maximum temperature for the disc surface. A rubbing thermocouple and an ambient temperature thermocouple close to the disc are used by the controller for condition monitoring and control of the testing process.

Experiment Conditions

The following conditions were set for the experiment.

- Disc rotating speed: 800 RPM. A mid-range family car with a 195/55 R15 (595mm diameter) tyre rotating at 800RPM will result in the vehicle travelling at 90kph (≈56mph).
- Preheat Load: 300N. The minimum load allowed by the test rig was used for preheating the disc. This was to ensure that the disc was not heated fast at the contact interface while cold on other places, but to allow the best possible uniformity in temperature distribution across the components. The uniformity in temperature distribution is required for proper association of the test rig with the FE model.
- Pad Load: 500N. This results to a pressure of 0.64 MPa on an area of 776mm^2 on the backplate. Restrictions in the test rig hydraulics did not allow further load increase.
- Preheat temperature 150^{o}C and cooling temperature 120^{o}C. Slow cooling will be allowed (no cooling fans) to achieve the desired uniformity in temperature distribution before the actual application takes place. Measurements for preheat and cooling temperature for the controller are taken from the rubbing thermocouple (T3).

Bedding-in procedure

Bedding in was carried out to ensure the best contact at the interface. During bedding-in three regions have been identified (see Figure 2(a), (b) and (c)). From these it was observed that the highest pressure, based on material wear occurred at the leading side of the pad. Although numerical values for the experimental contact pressure could not be measured, the wear of the pad surface can suggest its distribution in terms of high and low points. This will be useful when comparing it with the FE model.

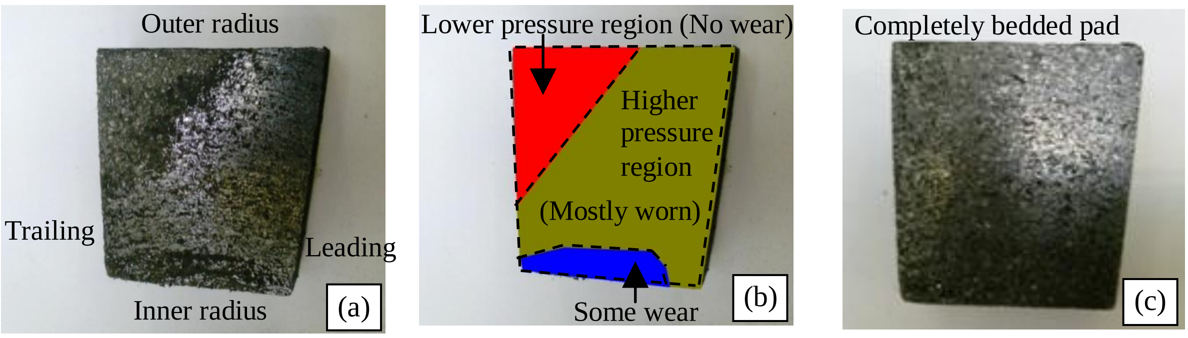

Figure 2: (a) Pad wear (b) Three identifiable regions and (c) completely bedded pad

TEST-RIG BASED FEA MODEL

A test rig-based finite element model with identical boundary conditions was attempted. The model was composed of the disc, the pad and the back plate. Material properties for the disc where taken from (9) and for the pad and back plate from (10).

Simulation Set-Up and Conditions

The FE model set-up was composed of four steps as summarised in Table 1. Two different FE models are used. The first is used to predict the preheat temperature (This will help in associating the experimental with the numerical approach). Then, the result of the first model (temperature distribution) is used as a starting point for the second model.

Table 1: Steps composing the simulation set-up

	Step	Information/Action
Model 1		
1	Components Preheat	• Known temperatures are set as boundary conditions. • FE model runs until a steady-state condition is reached. • Temperature distributions for disc and pad are predicted
Model 2		
2	Initial (Model 2)	• Import temperature distributions from model 1
3	Apply Pressure	• Cooling coefficients • Initial disc Conditions • Disc/Pad contact definition • Apply load
4	Rotate	• Modify disc conditions • Back plate rotational conditions • Rotational disc displacement (shaft effect) • Disc rotation

Components Pre-Heating (In step 1)

Before the application begins, the disc and pad components on the test rig have been pre-heated and cooled at certain temperatures. Computing power and software limitations does not allow running the FE model until these temperatures are reached. For this reason, the temperature distribution at the point where the cooling ends and the application initiates had to be estimated for the FE model.

The estimation of the component temperatures was done by taking the temperatures of known points from the test rig. For the pad component, reference points were taken from the thermocouples, as seen in Figure 1. The temperatures at these points were taken in ABAQUS as boundary conditions (see Figure 3), and the temperatures in between were interpolated until the model reached a quasi-static situation. Slow preheating and cooling on the test rig allowed getting as near as possible to the quasi-static situation. The same was repeated for the disc components.

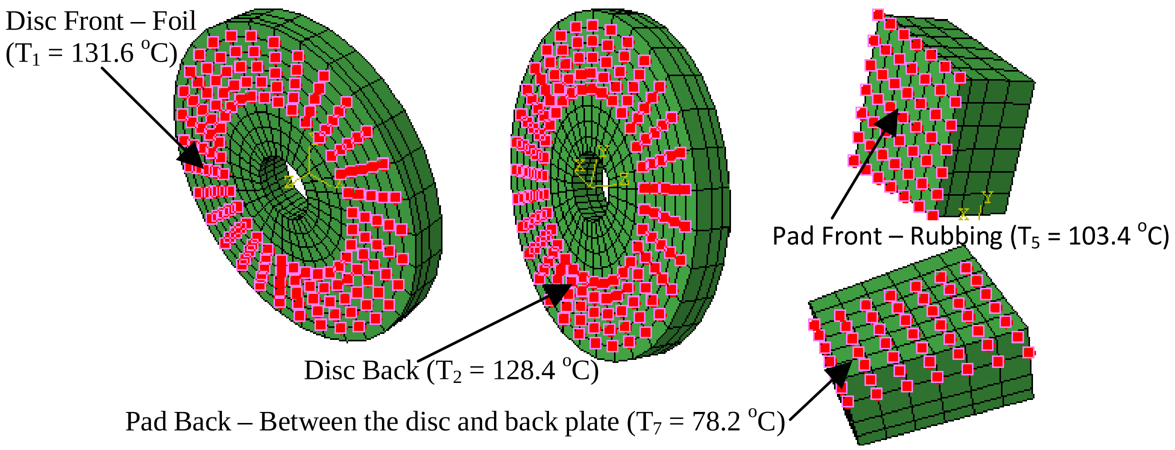

Figure 3: Reference temperatures for the prediction of temperature distribution

Disc Cooling (In step 3)

Cooling was allowed only at the front side of the disc and at the outer radius where the disc is exposed to the environment. The back side of the disc is treated as isolated. A thermal gasket is placed on the disc in test rig to be able to associate the two methods. Cooling on these surfaces was forced because the disc was rotating. Since the cooling provided was based on the disc rotational speed, no cooling was applied on the pad, as it is stationary. The calculation of cooling coefficients on the disc rubbing surface were made from the empirical equations (11). These were based on air properties, the rotor's dimensions and rotating speed.

Disc/Pad Contact Definition and Load (In step 3)

An average coefficient of friction of 0.4 has been used. An equivalent thermal conductance value for the effects of the interface tribo-layer of 2.064E6 W/m^{2}K has been used. This replicates the effects of a 5μm ITL for the materials used, having an 80% pad and 20% disc composition within the layer (shown in previous work (1). A load of 0.64MPa was applied on the entire back plate surface (as in the test rig) before proceeding to step 4 (as in Table 1).

RESULTS

In this section the results of both methods will be presented and compared.

<u>Test Rig</u>

Figure 4 shows the temperature readings taken from the test rig for the specific application (reference temperatures) as shown in Figure 3. There was an interruption of signal for the pad front thermocouple (T5) just before the application started, and the signal came back a few seconds after the application had finished. For this reason the starting temperature for T5 is known, but the temperature history during the application is unknown at this point. More details can be found in the discussion section.

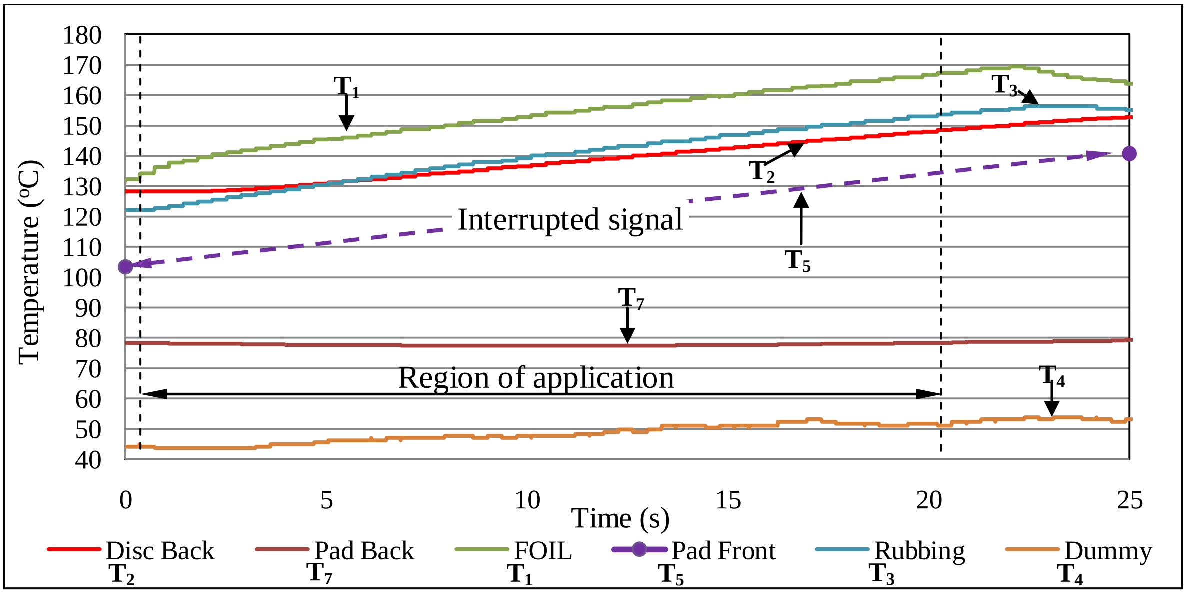

Figure 4: Temperature readings during the lab application

<u>FE Model</u>

Figure 5 shows the FE model results for the same positions as shown in Figure 4 for the test rig. Measurements on the FE model are taken on nodes on the same physical locations as the thermocouples on the test rig. The foil and disc back nodes are rotating with the disc.

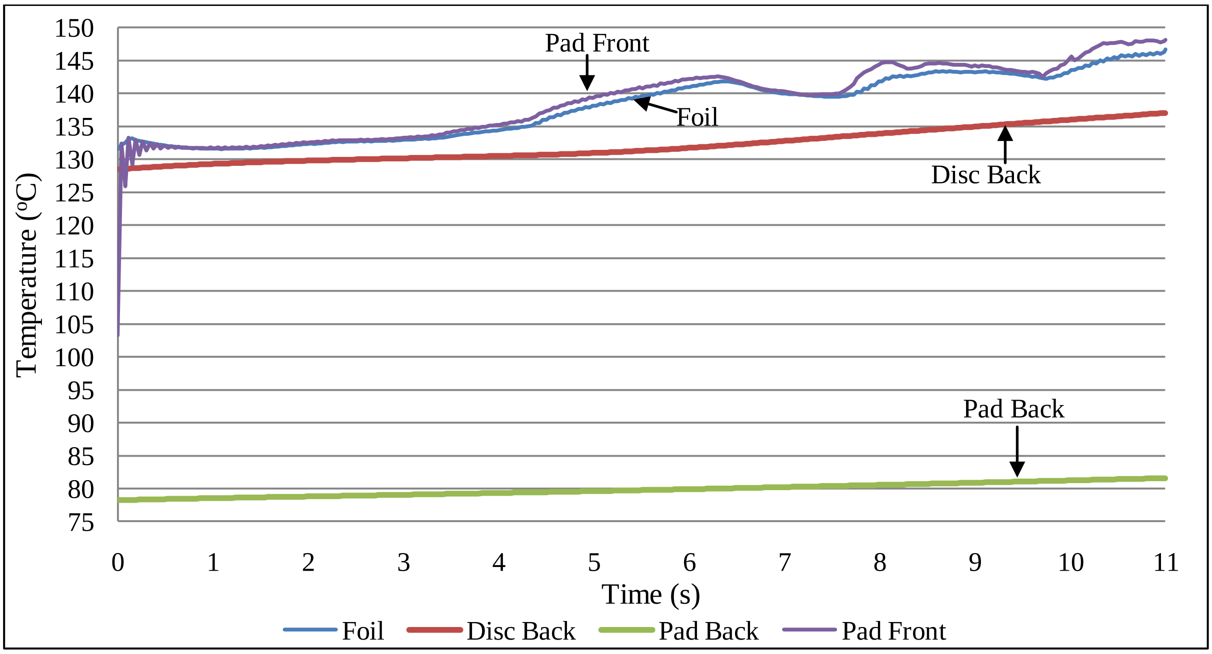

Figure 5: FE model temperature results

Table 2 compares the temperature results for the two methods at the end of the 10 second application as shown in Figure 4 and Figure 5.

Table 2: Results Comparison

	Temperature (°C)				
	T_1 Foil	T_2 Disc back	T_3 Rubbing	T_5 Pad Lead	T_7 Pad Back
Experimental (Test rig) (T_E)	154.2	137.5	140.6	-	77.4
Numerical (FE method) (T_N)	143.6	136.0	145.0	145.0	81.2
% Difference $\left[\left(\frac{T_E - T_N}{T_E}\right) \times 100\right]$	6.87	1.09	-3.13	-	-4.91

Figure 6 shows the instantaneous contact pressure distribution, heat flux multiplied with the nodal area (HFLA) on the contact surfaces, and temperature distribution at 5 seconds. A positive HFLA sign indicates that heat is leaving the surface, and a negative sign that heat enters the surface. By comparing Figure 6(a, b) with Figure 6(c, d) the effect of the instantaneous contact pressure on heat generation is apparent.

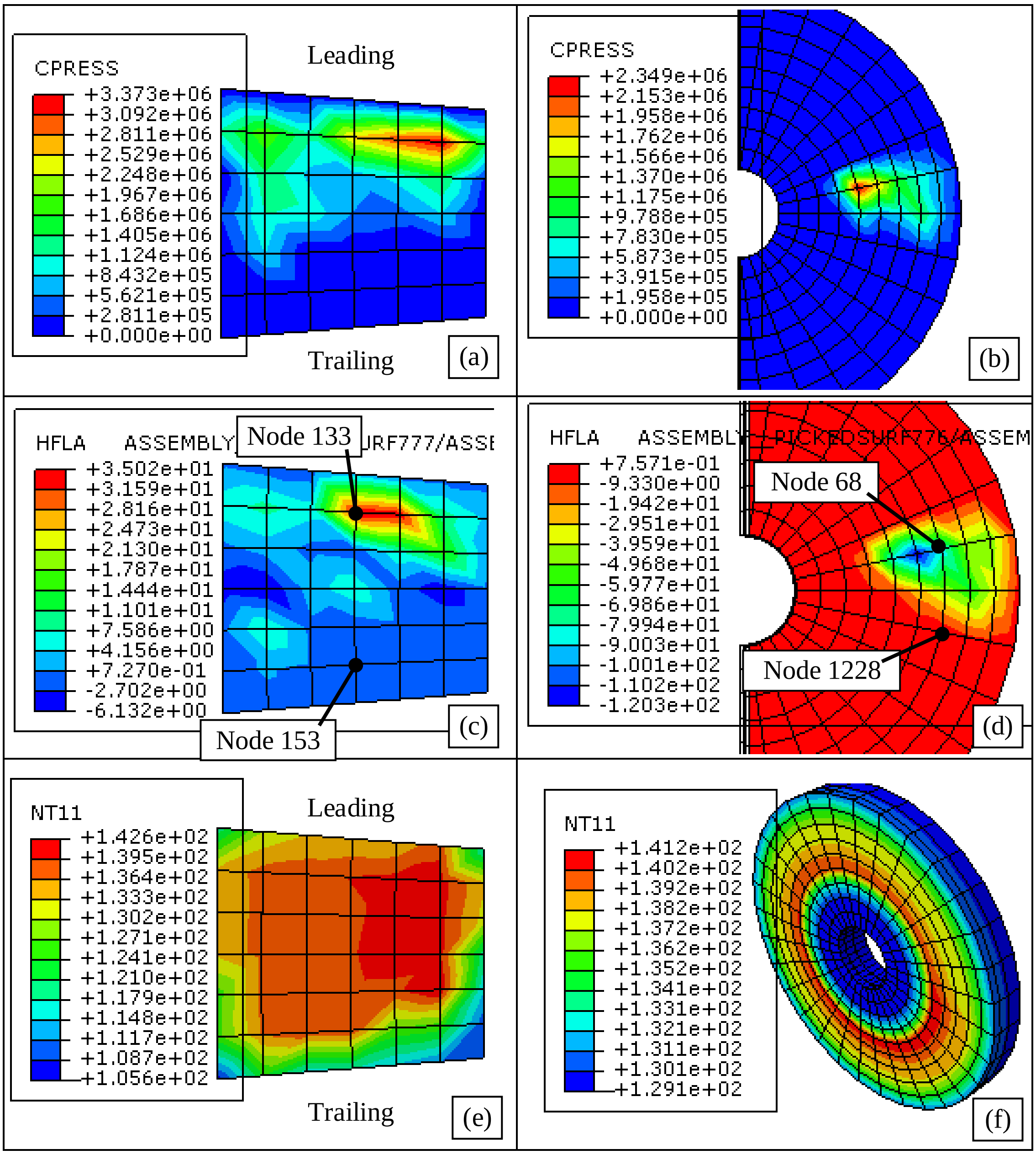

Figure 6: (a) Pad and (b) disc contact pressure, pad (c) and disc (d) heat flux multiplied by the nodal area and pad (e) and disc (f) temperature distribution for the same time instant (t = 5 seconds)

Table 3 shows the heat generation and partition between the leading and trailing edge of the pad, at nodes 68, 133, 1228 and 153 as shown in Figure 6.

Table 3: Temperature and heat partition/generation at the leading and trailing edges

	Component	Node	Pressure (MPa)	HFLA (J/s)	Absolute HFLA (J/s)	Total HFLA (Absolute) (J/s)	Partition (%)
Lead	Disc	68	1.127	-85.76	85.76	120.78	28.99
	Pad	133	2.587	35.02	35.02		
Trail	Disc	1228	0.003	0.76	0.76	1.33	43.12
	Pad	153	0	-0.57	0.57		

DISCUSSION

There were some intermittent signalling problems with the conventional type thermocouples. The data extracted from the experiments will help in addressing the problems, and to also see which the strengths and weaknesses of the apparatus are. When there was no intermitted signalling problem, it has been observed that the data extracted was consistent in a range of about ±5 ^{o}C. This shows that when the signalling problem is solved the results will be consistent.

The results of the FE method seem to have a good agreement with the experiments having a temperature difference in the region of 1.09 to 6.87%, but improvements will still be made. These will include the addition of bedding-in in the modelling. Refinements on the pad mesh can be made, so that when heat partitioning is measured the nodes will meet precisely in the same position. In this paper, the closest nodes were chosen for measuring (between the two surfaces), which in some cases are up to 1.5 millimetres apart. All these are a part of future work. The FE simulations were able to demonstrate the distribution of contact pressure, heat generation and temperature. The FE models show higher contact pressure on the leading inner side of the pad, which agrees with the bedding-in procedure observations. This makes the inclusion of bedding-in in the FE model important, in terms of making the results more realistic. After including the bedding in, it is expected that the instantaneous contact pressure distribution will change. It will spread in a bigger area, rather than concentrating on the inner leading edge of the pad, which can have an effect on the real contact area. The dependency of heat generation on instantaneous contact pressure is apparent in Figure 6.

CONCLUSIONS

1. Increased contact pressure results in an increased frictional heat generation and to a reduction in heat partitioning. It tends to be higher to the pad leading edge.
2. Heat partitioning is found to be varying significantly from the leading to the trailing edge (circumferential direction) on the contact interface. It is also varying in the radial direction as the contact pressure varies from the inner to the outer radius of the disc.
3. Even though the higher temperature points between the disc and pad are close the temperatures at the contact interface do not match. A temperature difference on the interface of up to 30^{o}C has been observed at some points on the same time increment (as in Figure 6).

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