Investigation of Disc/Pad Interface Temperatures in Friction Braking

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Abstract

Maintaining appropriate levels of disc-pad interface temperature is critical for the overall operating effectiveness of disc brakes and implicitly the safety of the vehicle. Measurement and prediction of the distribution and magnitude of brake friction interface temperatures are difficult. A thermocouple method with an exposed hot junction configuration is used for interface temperature measurement in this study. Factors influencing the magnitude and distribution of interface temperature are discussed. It is found that there is a strong correlation between the contact area ratio and the interface maximum temperature. Using a designed experiment approach, the factors affecting the interface temperature, including the number of braking applications, sliding speed, braking load and type of friction material were studied. It was found that the number of braking applications affects the interface temperature the most. The real contact area between the disc and pad, i.e. pad regions where the bulk of the kinetic energy is dissipated via friction, has significant effect on the braking interface temperature. For understanding the effect of real contact area on local interface temperatures and friction coefficient, Finite Element Analysis (FEA) is conducted. It is found that the maximum temperature at the friction interface does not increase linearly with decreasing contact area ratio. This finding is potentially significant in optimising the design and formulation of friction materials for stable friction and wear performance.

Key words

Friction, Interface, Contact, Temperature, Measurement, Finite Element, Thermal, Brake

NOMENCLATURE

μ	Coefficient of friction
V	Sliding speed (m/s)
F	Friction force (N)
P	Rate of thermal energy = FV (W)
r	Heat partition ratio
P_p	Rate of the heat transferred into the pad = $r\,P$ (W)
A_r	Real area of contact (mm^2)
A_n	Portion of the pad surface which is not in contact with the rotor surface, (mm^2)
A_a	Apparent contact area of the pad = $A_r + A_n$
A_s	Total surface area of the brake pad excluding the apparent contact area A_a (mm^2)
R	Ratio of the real area and the apparent contact area, A_r/A_a ($0 \le R \le 1$)
h	Surface heat transfer coefficient (W/m^{2}K)
k	Thermal conductivity of the friction material (W/mK)
T_{max}	Maximum temperature of the brake pad (°C)
T_{ave}	Average temperature of the brake pad body (°C)
Δ_R	Change of the ratio R
Δ_{Tmax}	Change of the maximum temperature of the brake pad (°C)
Δ_{Tave}	Change of average temperature of the brake pad body (°C)
DoE	A statistical design of experiments approach
emf	ElectroMotive Force
WIRE	An exposed wire thermocouple configuration for brake disc/pad interface temperature measurement
RUB	A rubbing thermocouple for brake disc surface temperature measurement

1. INTRODUCTION

Friction brakes are required to transform large amounts of kinetic energy into heat energy at the contact surfaces between the brake rotor and stator. The temperature distribution in the friction pair generated in the process is a complex phenomenon which directly affects the braking performance, and has been investigated by many researchers over many years [1 – 5]. The aim of the work presented here is to understand disc/pad

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interface temperature and friction in vehicle braking by means of an exposed thermocouple technique for interface temperature measurements and FE thermal modeling approach.

1.1 The technique of exposed thermocouple interface temperature measurement

It would be desirable to measure interface temperatures during actual friction braking tests so that the precise operating conditions were known for design purposes. However, measuring the interface temperature of a friction pair is a difficult task. Several methods have been reported [5 - 8], and these can be categorized into non-contact measurement, which includes the methods such as optical and infrared measurement, and contact measurement, which includes the methods such as thermocouple and temperature sensitive material coating (or paint). Those methods mentioned above are not very effective in measuring the interface temperatures. The exposed thermocouple technique for measuring grinding interface temperature developed by Qi and his colleague [9] was reported as an effective method for measuring interface temperature of a friction pair. The difference of the exposed thermocouple technique in comparison with conventional embedded thermocouple method is that the hot junction of the thermocouple is located directly at the friction interface. The validation work by Xu [10] shows that the exposed thermocouple technique can measure temperature as accurate as other common accepted methods, and confirms that directly contact of the hot junction tip of thermocouple to the interface of a friction pair has little effect on the quality of the temperature signal produced. The exposed thermocouple technique was adopted in measuring the brake disc/pad interface temperature in authors’ previous work [11]. In this study, the technique of the exposed thermocouple interface temperature measurement is used in the investigation of the factors that affect the friction braking interface temperatures under designed experimental conditions.

1.2 The FE modeling approach for analysis of pad temperature distribution

It is known that the real area of contact between a brake disc and pad in operation is much smaller than the apparent contact area [12, 13, 14]. This is implicit in classical theories of friction, but in brakes it is emphasised by:

- Material structure and composition of the pad and the disc materials;
- Surface texture patterns generated by the machining of the surfaces;

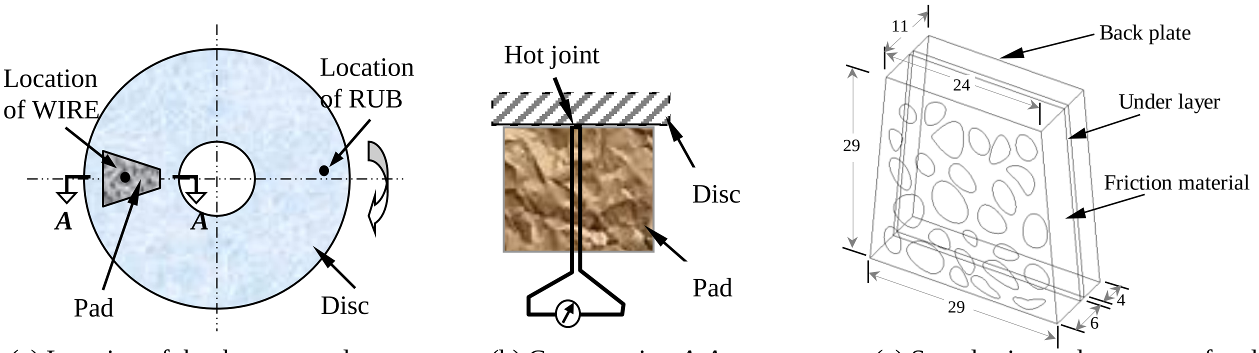
- The braking process itself, e.g. the uneven thermal expansion or thermal distortion of the disc and the pad or system vibration and dynamic effects.

By using a high definition thermal imaging system [8], for example, the surface and near-surface temperatures were monitored at various locations in a disc brake during drag-type testing. It showed that contact at the friction surface was not uniform, with contact areas constantly shifting due to non-uniform thermal expansion and wear. Eriksson & Jacobson [13] explained that primary and secondary contact plateaux are formed at the contact surface of a friction pad; the primary plateaux form first due to the lower removal rate of the mechanically stable and wear resistant ingredients of the pad, such as metal fibres. The secondary plateaux are formed by compacted wear debris initiated by the protruding hard phase, i.e. primary plateaux, to form nucleation sites for their growth. These contact plateaux form the surface contours, which define the real contact area of a brake pad. The shape and area of the real contact surface can be affected directly by wear in two ways: the growth of secondary plateaux and delamination of the contact plateaux. Such a real contact area distribution and its effect on interface temperature distribution cannot be measured directly. The only alternative means for analysis of effect of the real contact area distribution on the real contact temperature distribution is utilising Finite Element Analysis (FEA). FEA has been proved as an effective method to study the behaviour of friction brakes in terms of stress/strain, temperature, thermoelastic instability (TEI), vibration/noise and service life in vehicle braking [2, 14-18]. Different approaches and assumptions have been used for model simplification in different FE analyses. An FEA of the effect of disc/pad real contact area on the brake pad temperature distribution was carried out in authors' previous work [19]. In this investigation the FE modelling technique is used to simulate and visualize the maximum brake interface temperature and its distributions in friction braking under different contact scenario in terms of contact area ratio and real contact area distribution.

2 EXPERIMENT SET-UP

2.1 Measurement system

The thermocouple measurement method used was based on the dual assumption that the physical parameter measured by the thermocouple was an aggregate ElectroMotive Force (emf) generated at the thermocouple wire tip and the disc interface, and that the emf obtained corresponded to the local disc and pad interface temperatures. Fig. 1 shows the exposed wire thermocouple (WIRE) configuration used for the interface temperature measurement. The hot joint of the WIRE located at the interface within the disc-pad contact zone, as shown in Fig. 1b. In addition, a conventional rubbing thermocouple (RUB) setup, which is commonly used for monitoring disc temperature, was used for comparison as well as for system automatic control and failure diagnosis. The hot joint of the RUB contacted with the disc surface and located at a distance from the disc and pad contact zone (Fig.1a). All these thermocouples were installed on a small sample friction test dynamometer. This test rig is hydraulically actuated and computer controlled, with a sliding speed range from 5 – 16 m/s, and power dissipation in the range 0 – 10 MW/m^2. The control variables, e.g. normal load, sliding speed, cooling temperature and loading time were computer programmed/controlled. The output signals, e.g. friction force, interface temperature by WIRE and disc surface rubbing temperature by RUB, were acquired and processed by a computer aided data acquisition system. During each test, the cast iron disc, 125 mm diameter, is rotated at constant speed, and the pad specimen (approximately 29 mm x 24 mm x 11 mm, see Fig. 1c) was pressed in contact with the disc under a constant normal load over a sequence of successive 20-second application. In between each 20-second application the pad was released from contact allowing it to cool to 80 °C.



(a) Location of the thermocouples (b) Cross-section *A-A* (c) Sample size and structure of pad

Fig.1 Set up of the thermocouples used and the structure of the pad sample used

Table 1 Conditions used in the designed experiments

Factors	Level (-1)	Level (+1)	Assigned Letter
Load (N)	150	300	A
Speed (rpm)	1000	2500	B
Material	M	P	C
Disc/pad surface cond.	*SC1*	*SC2*	D

The following conditions were set throughout the experiments:
(1) loading time *20 s*;
(2) number of applications for each test 7;
(3) preheating load *450 N*;
(4) preheating temperature *80 ℃* and
(5) cooling temperature *80 ℃*.

Table 2 $L_{16}(2^4)$ Latin square used in the DoE

	1	2	3	4	5	6	7	8	9	10
Test No.	**A**	**B**	**A*B**	**C**	**A*C**	**B*C**	**D**	**A*D**	**B*D**	**C*D**
1	-1	-1	1	-1	1	1	-1	1	1	1
2	-1	-1	1	-1	1	1	1	-1	-1	-1
3	-1	-1	1	1	-1	-1	-1	1	1	-1
4	-1	-1	1	1	-1	-1	1	-1	-1	1
5	-1	1	-1	-1	1	-1	-1	1	-1	1
6	-1	1	-1	-1	1	-1	1	-1	1	-1
7	-1	1	-1	1	-1	1	-1	1	-1	-1
8	-1	1	-1	1	-1	1	1	-1	1	1
9	1	-1	-1	-1	-1	1	-1	-1	1	1
10	1	-1	-1	-1	-1	1	1	1	-1	-1
11	1	-1	-1	1	1	-1	-1	-1	1	-1
12	1	-1	-1	1	1	-1	1	1	-1	1
13	1	1	1	-1	-1	-1	-1	-1	-1	1
14	1	1	1	-1	-1	-1	1	1	1	-1
15	1	1	1	1	1	1	-1	-1	-1	-1
16	1	1	1	1	1	1	1	1	1	1

2.2 Design of experiments

A statistical design of experiments (DoE) was established to study the effects of braking factors on the braking performance. A four factor, two level full factorial orthogonal design was selected as summarized in Table 1. The pad materials used were *M*: a woven asbestos free friction lining, and *P*: a compression molded, heavy-duty brake lining for commercial vehicle drum brakes. Disc/pad surface conditions were controlled as *SC1* and *SC2*. Under the test condition *SC1* (braking applications 1 to 7) the pad and disc surfaces were prepared to the "new" condition before each test, using abrasive paper. Under the test condition *SC2* (braking applications 8 to 14) the pad surface was left in the "used" condition while the disc surface was prepared to the "new" condition before each test, using abrasive paper. The main outputs (or responses) from this DoE were the interface temperature, *T*, (measured by WIRE), and the friction coefficient, μ. An $L_{16}(2^4)$ Latin Square experimental design [20] was used for the full factorial experiments as shown in Table 2. The 10 columns in the table represent four factors and six interactions between any two factors in the table. The rows in the table are the 16 tests (T1 to T16) carried out in the experiments.

3 SET-UP FOR FINITE ELEMENT (FE) MODELLING

In the FE modeling, the pad's surface temperature variations in friction braking was studied with special interest in the non uniform contact at the brake friction interface and the real contact area ratio, $\boldsymbol{R} = \boldsymbol{A_r}/\boldsymbol{A_a}$. A constant thermal energy partition ratio, $\boldsymbol{r}$, was introduced under the assumption that $\boldsymbol{r}$ only affects the absolute pad temperature but has no significant effect on the temperature variation; such assumption helps to make the modelling manageable. The loading input conditions were the same as those used in the experimental investigation. The material property of the friction pad detailed in Table 3 was used in the FE modeling. Fig. 1c shows the 3-D model of the friction pad. The model had three layers; friction material (6 mm thick with an apparent rubbing surface area of $\boldsymbol{A_a}$ = 768.5 mm^2), an underlayer (1 mm thick) and a backplate (4 mm thick). To represent the different thermal behavior, the front face of the pad was partitioned into the real contact area, $\boldsymbol{A_r}$, and the non-contact area, $\boldsymbol{A_n}$, as shown in Fig. 1c. A free meshing technique was used with four-node tetrahedronal solid elements. The FE modelling and simulation was carried out using the commercially available I-DEAS CAE system [21].

The ratio of the real area and the apparent contact area, $\boldsymbol{R}$, was set up as a control factor in the FE modelling. Five models, labeled PSC1 to PSC5, were setup for simulating five contact scenarios. Table 4 details the contact area ratio and the real area distribution used, from which the effective heat flux acting on the real contact interface, $\boldsymbol{P_b}/\boldsymbol{A_r}$, was obtained. The friction heat source at the disc/pad interface was treated as a thermal load boundary for the pad. Two sequential heating and cooling phases were set up as the thermal load boundary condition as shown in Table 5. In the heating phase (20 second duration) a constant normal load was assumed to act on the pad with a constant sliding speed. In the cooling phase, the pad front surface was removed from the disc surface for 40 seconds, during which time surface heat transfer took place on all the pad surfaces including the pad front surface. The surface heat transfer from the pad front surface occurred via forced convection or free convection.

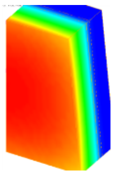
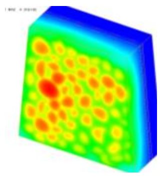
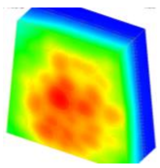
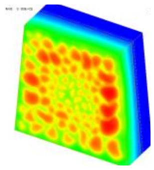
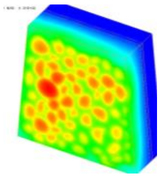
Table 3 Model conditions used in the FEA

Pad friction material properties used [15]	
Elastic modulus (GPa)	1.25
Poisson's ratio	0.3
Thermal conductivity (W/m K)	2.06
Heat capacity (J/kg K)	749
Density (kg/m)	14.3

Table 5 The transient boundary

Surfaces	Heating phase		Cooling phase	
	Heat flux	$\boldsymbol{h}$ (W/m^2°C)	Heat flux	$\boldsymbol{h}$ (W/m^2°C)
$\boldsymbol{A_r}$	$\boldsymbol{P_p}/\boldsymbol{A_r}$	0	0	10
$\boldsymbol{A_n}$	0	0	0	10
$\boldsymbol{A_s}$	0	10	0	10

Table 4 Conditions and assumptions used in the FE models

Model No:	**PSC1**	**PSC2**	**PSC3**	**PSC4**	**PSC5**
Models					
$\boldsymbol{R = A_r/A_a}$	1	0.487	0.386	0.554	0.536
$\boldsymbol{A_r}$ ($\times 10^{-4}$ m^2)	7.685	3.74	2.967	4.257	4.116
Heat flux $\boldsymbol{P_b/A_r}$ (W/m^2)	101496	208556	262892	183219	189523
Constants	Friction coefficient $\boldsymbol{\mu}$ = 0.4 Friction speed $\boldsymbol{V}$ = 6.5 m/s Normal load = 300 N Heat transfer coefficient $\boldsymbol{h}$ = 10 W/m^2°C Partition ratio $\boldsymbol{r}$ = 0.1				

4 RESULTS

4.1 Temperature signals assessment

Fig. 2 shows a general overview of the temperature signals obtained during the friction braking tests using the scale test rig. This shows that the signals responded to the brake loading phase and the cooling phase correctly. The reliability of the measuring system was assessed by analyzing the RUB signals; as shown in Fig.2, during individual braking applications the temperature indicated by RUB increased from start temperature of 80 °C. After each application, the temperature returned steadily to 80 °C and then increased again as the next application started. It was shown that the temperature signals measured from the different sensor set-ups indicated different signal response speeds. The temperature signals measured by WIRE reduced immediately as the braking load was removed. However, the temperature signal measured by RUB started to reduce with a delay of about 4 seconds from the end of the braking application [11]. This was mainly due to RUB being located a distance from the disc and pad contact interface, as indicated in Fig. 1a. The WIRE technique, therefore, is better than the RUB technique in responding to interface interactions between the disc and pad during braking applications.

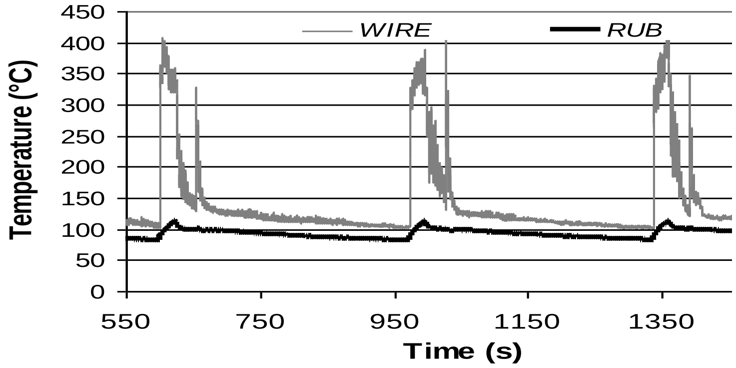


Fig. 2 Typical temperature-time signals from the experiments

4.2 Effects analysis

Based on the DoE, effects plots and Daniel plots were generated. Daniel plots, or half normal plots, are statistical plotting techniques used to analyse experimental data, especially to identify the more significant factors among those investigated [20]. In the Daniel plots of Fig. 3a, for example, the absolute values of the temperature contrasts were plotted, so that all the large temperature contrasts appeared on the right-hand side of the graph. As shown in Fig.3a, the factor D (the disc/pad surface conditions) and the interaction B*D (interaction between the disc/pad surface conditions and the braking load) appear on the right-hand side, which means that the disc/pad surface condition, or the number of braking applications, has the strongest effect on the interface temperature, either directly or indirectly (i.e. through interaction with other factors). Similarly, the Daniel plot for friction coefficient in Fig.3b shows that the most effect factor on friction is not the speed used, not the load applied, and not the friction material tested but the interaction B*D (interaction between the disc/pad surface conditions and the braking load) and the factor D, i.e. disc/pad interface contact condition. Further interpretation of this statistical result is given in the section 5.

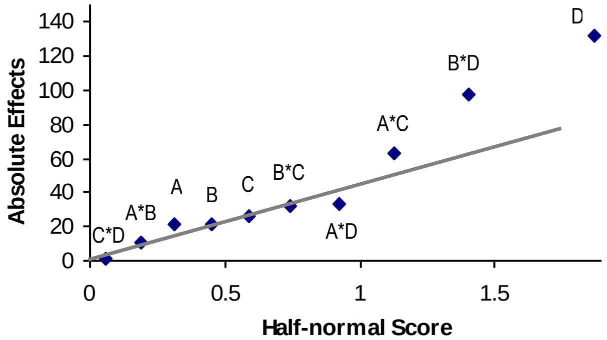


Fig. 3a Daniel plot for temperature

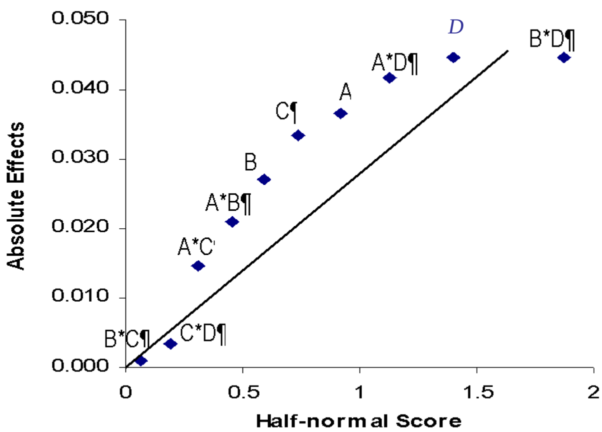


Fig. 3b Daniel plot for friction coefficient

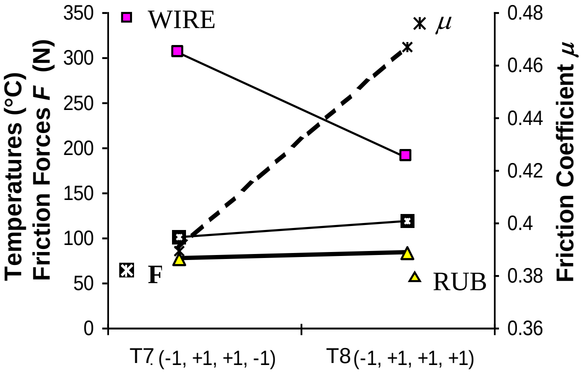


Fig. 4 Effects of the disc/pad surface conditions

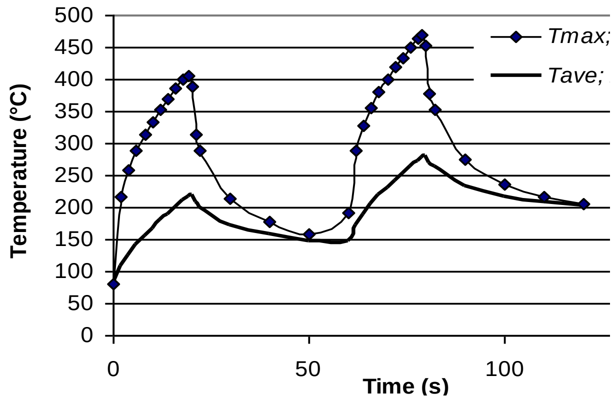


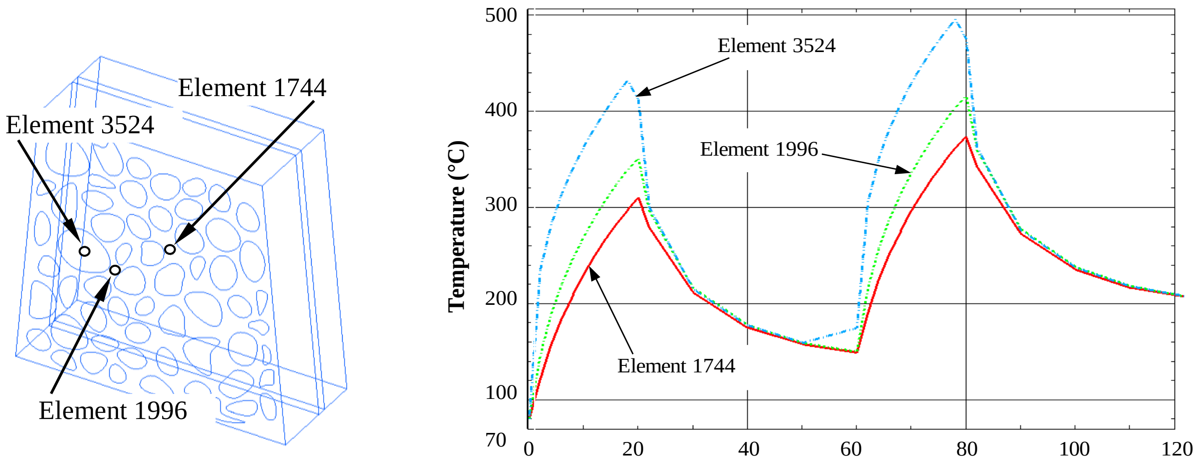
Fig.5 Typical maximum temperature vs time from the FE model PSC5

4.3 The effects of a number of braking applications

For further understanding why the number of braking applications (factor D) affected the most on the braking performance, the test results from T7 and T8 were compared, as shown in Fig. 4. The difference between test conditions T7 and T8 was Factor D, the number of braking applications. For T7, D = SC1: braking applications 1 to 7; and for T8, D = SC2: braking applications 8 to 14. Fig. 4 shows that the local interface temperatures, measured by WIRE, in test condition T7 were higher than that in test condition T8. In contrast, the disc surface temperatures, measured by RUB, in test condition T7 were slightly lower than those in test condition T8. At the same time, the average friction force, F, and friction coefficients μ under condition T7 were lower than that under condition T8. The above phenomenon can be summarized as: the increase in the braking application times from 1-7 to 8-14 causes an increase in friction coefficient μ and friction force, F, and consequently the disc temperature, measured by RUB, increases. Unexpectedly, the local disc/pad interface temperature, measured by WIRE, decreases instead of increases. This is a favorable scenario as far as vehicle braking is concerned, i.e. a higher friction torque combined with a lower disc/pad interface temperature. The mechanisms behind this phenomenon are discussed in section 5.

4.4 Effect analysis of contact area ratios with FE approach

The changing pattern of the maximum temperature of the uneven disc/pad contact during the heating and cooling phases were simulated by FEA. Fig. 5 and Fig. 6 are the typical temperature plots. Fig. 5 shows the graph of maximum temperature, T_{max}, vs time obtained from the simulation of the FE model PSC5. It shows that the maximum temperature increased rapidly during the 20 second heating period. During cooling, the maximum pad temperature reduced rapidly as no heat generation at the contact surface area plus heat dissipation from the surface. The change of the pad body average temperature, T_{ave}, is also shown in Fig. 5. The temperature pattern is similar to that obtained from the experiments, such as that shown in Fig. 2. The temperature history plots of some selected elements located on the pad front surface are given in Fig. 6. It found that the location of the maximum interface temperature, instead of being unchanged at the centre of the pad contact surface, might vary depending on the real pad/disc contact area distribution. For example, the temperature of element 1744, which was located near the centre of the pad surface, was much lower than the temperature of element 3524, which was located further away from the geometry centre of the pad surface, no matter in the heating phase or the cooling phase in a braking application.



(a) Location of the elements numbered 3524, 1996 and 1744

(b) The temperature plots for the elements

Fig. 6 The temperature plots of elements 1744, 1996 and 3524 in 120 second heating and cooling phases with the PSC2 model (R = 0.487)

Fig. 7 shows a general pattern on how the ratio of contact area affecting the surface temperatures. The maximum temperature, T_{max}, increases with the decrease in the contact area ratio. In contrast the average pad temperature, T_{ave} is independent of the contact area ratio. It also shows that the maximum temperature do not increase linearly with decreasing contact area ratio. For example, as a comparison in Table 6, the maximum temperature changed only by 50 °C when the contact area ratio changed by half (from 1 to 0.55), whereas it changed by 100 °C when the ratio changed by only 0.168 (from 0.55 to 0.39). In an actual friction braking condition, the effective disc/pad contact area would be expected to be less then half of its apparent contact area [12,13]. It can be expected therefore, from the above simulation results, that the change of the effective contact area during braking will have a significant effect on the local pad maximum temperature variation.

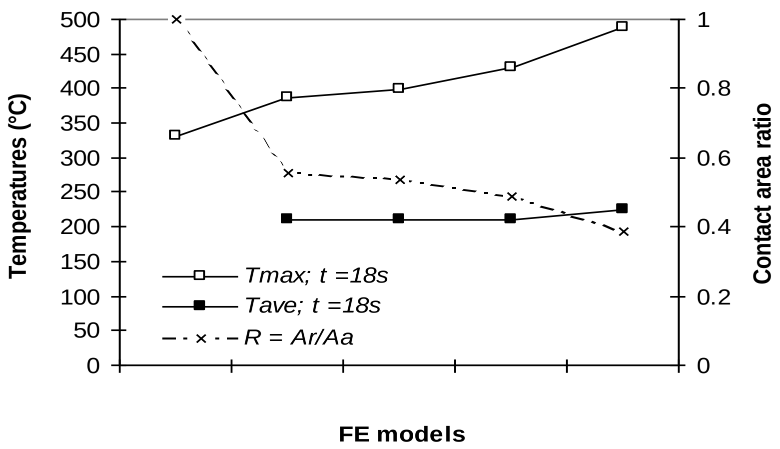


Fig. 7 The effect of the ratio of contact area on the pad surface temperatures

Table 6 Analysis of the change of contact area ratio on the change of interface temperature

	PSC1 and PSC4		PSC4 and PSC3		PSC5 and PSC2	
	18th Second	78th Second	18th Second	78th Second	18th Second	78th Second
Δ_R	0.446 (=1 - 0.554)		0.168 (= 0.554 – 0.386)		0.049 (= 0.536 – 0.487)	
Δ_{Tmax} (°C)	57	62.3	100	103.4	29.8	29.34
Δ_{Tave} (°C)	-	-	13	14.7	0.27	-0.8

5 DISCUSSIONS

5.1 The effects of the disc/pad interface variation

○ The local interface temperature

When two fresh pad and disc surfaces are in contact under braking load, the pad friction surface is only in partial contact at bulk/macro scale, as the two surfaces are not bedded in (i.e. geometric conformity is not achieved) [14]. Wear subsequently enables bedding-in to be completed. Furthermore, as the number of braking applications increases, the local real contact area distribution will change at macro/micro scale as well. Under fresh conditions the pad surface is in an open condition, i.e. only primary contact plateaux are in contact with the disc and there are pores or voids existing between the primary contact plateaux. This is another reason why the initial real contact area A_r is much smaller than the apparent contact area A_a. As the number of braking applications increases, however, the wear debris will fill in the "open" area between the primary contact plateaux, as a so-called 'secondary contact plateaux' formation [13]. This process may be belonging to the 'burnishing'. The 'geometric conformity' and 'secondary contact plateaux' formation are a physical increase in the real contact area A_r. If assuming that total normal load and friction coefficient are constant, then the increase in the real contact area A_r means a decrease in the average local normal pressure and friction/shear stress (F/A_r). As a result, the local heat flux and local temperature at the interface, which is proportional to the friction/shear stress ($\boldsymbol{T_{max}} \propto F/A_r$) will decrease. The bulk average temperature, which is the function of the apparent contact area ($\boldsymbol{T_{ave}} \propto F/A_a$), however, will not be affected by the change of the real contact area. This may explain the phenomena found in the experiments (in Fig. 4) and obtained in the FEA (in Fig. 7) regarding to the local disc/pad interface temperature (or $\boldsymbol{T_{max}}$) and the disc surface temperature (or $\boldsymbol{T_{ave}}$). It can be concluded that the change of the effective contact area during braking has a significant effect on the pad local maximum temperatures and has little effect on the pad average temperatures.

- **The friction coefficient**

Fig. 8 (and Fig.4) shows that the friction coefficient is affected by the number of braking applications and it changes significantly during the first few braking applications, which is consistent with the results reported by Borjesson [12]. The increase of the instantaneous real contact area (or the change of the ratio A_r/A_a), discussed previously, cannot be used to explain why the friction coefficient increases with the increase of the number of the braking applications, because the contact area has no direct effect on the friction coefficient according to the Dry Friction Law. It is known that the pad/disc contact surfaces will change their surface martial properties during the burnishing process due to friction, wear and other mechanical or chemical interaction at the interface in the friction braking process. The concept of an Interface Layer or Tribo-layer has been used to distinguish the difference between the surface layer and the main bodies of the pad and disc. The Interface Layer formation has became a common and wide accepted phenomenon in vehicle braking [2, 22-24] and has been considered in as an important element in modeling of the braking process. For example, Day [2] examined the role of interface contact resistance in the calculation of heat flow and temperature generated in brake friction pairs using FEA using a 'five-phase model', i.e. Phase 1: Virgin friction material; Phase 2: Reaction zone; Phase 3: Surface char layer; Phase 4: Interface layer; and Phase 5: Metal mating body. He postulated that frictional heat is generated in the friction material (the softer of the friction pair) and transferred across the interface to the rotor. The interface layer (Phase 4) plays an important role in predicting interface temperatures and heat flows.

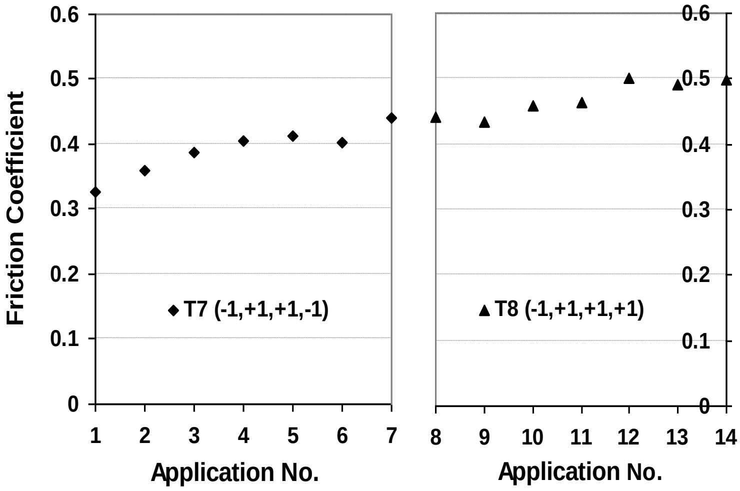


Fig 8 Effect of the number of braking applications on the friction coefficient

The friction coefficient, which is a function of the property of the interface layer as well as the friction pairs' properties, will change along with the burnishing process, i.e. a process of developing interface layer. If the metal wear debris or metal oxide fills in the "open" area on the pad surface by mechanical or chemical means during the process, the affinity between the two contact surfaces will increase. This will make the two contact surfaces disposed towards forming an adhesive bond (or weld junction), thus explaining the apparent increase in friction coefficient as shown in Fig. 4 and Fig. 8. In addition, some more free hard wear debris/particles behaving as abrasives may accumulate at the disc and pad interface along with braking process, which, consequently, will cause additional plastic shear deformation along the sliding direction, and apparently an increase in friction coefficient.

It is common practice that 'bedding-in and burnishing' is necessary for any new friction pair. The function of 'bedding-in and burnishing' is to enlarge and stabilise the actual contact area by secondary plateaux formation and the interface layer formation [13, 14].

The necessity of ‘bedding-in and burnishing’ a pad before being used can thus be explained by the mechanisms of the effect of the contact area ratio on the local pad maximum temperature and the effect of the interface layer formation on the friction coefficient.

5.2 Issues on future work

- **Further development in interface signal measurements**

The DoE experiments explored the phenomenon that the bedding-in and burnishing in braking process will provide a optimised braking condition, i.e. a combination of high friction torque and low maximum interface temperature. This phenomenon can only be detected by using interface temperature measurement techniques, e.g. using WIRE in this work. The conventional measurement techniques, e.g. RUB method commonly used in industry and in research/development, are unable to do so. Further development and improvement of the exposed thermocouple technique is necessary and is being carried out by authors.

- **Further development of the FE approach**

It is for the first time to simulate the uneven brake disc/pad contact scenario with FEA. The information obtained from FEA, such as the interface temperature under an uneven contact changing with the real area of contact in a non-linear manner, is unique and essential for understanding the interface temperature in braking processes. Our current FE models, where the size of real contact surface is fixed for each model, form a basic step toward much advanced FE models. One attempt for the further developments of the FE modeling is to make the size of real contact surface as a process variable, changing with time during the FE simulation. It is expected that the dynamic aspect of the contact variation on the thermal behavior of friction braking will be exposed by the advanced FE modeling approach.

6 CONCLUSIONS

1. Interface temperature measurements using an exposed thermocouple technique are applied in the vehicle brake disc/pad interface temperature measurement. By using the technique it unveils for the first time that the disc/pad interface temperature measured by WIRE, has different characteristic than that of the disc surface rubbing temperature, measured by RUB.

2. The results from the DoE experiments indicate that the number of braking applications has the strongest effect on the interface temperatures in comparison with other factors, i.e. friction loads, friction speeds and friction materials.
3. Finite Element analysis shows the correlation between the contact area ratio and the interface maximum temperature. It is found for the first time that the maximum temperature at the friction interface does not increase linearly with decreasing contact area ratio. In contrast, the contact area ratio has little effect on the average pad bulk temperature.
4. The findings from the experiments and FE simulations provide an insight of how the real contact area affects the friction interface temperature in friction disc/pad braking. The findings are used to interpret the well-known 'Bedding-in' and 'Burnishing' mechanisms in friction braking.
5. For a better understanding of vehicle braking process and disc/pad friction/thermal behaviors during the process, a direct pad/disc interface temperature measurement, such as the exposed thermocouple technique used in this study, should be conducted. The conventional rubbing thermocouple technique, which is still commonly used, is inadequate for detecting and interpreting the phenomenon of the braking disc/pad interface temperature. The FEA approach used in this study is a very effective method for insight of the disc/pad interaction during a friction braking process.

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