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Interface temperatures in friction braking

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Synopsis
Results and analysis from investigations into the behaviour of the interfacial layer (Tribo-layer) at the friction interface of a brake friction pair (resin bonded composite friction material and cast iron rotor) are presented in which the disc/pad interface temperature has been measured using thermocouple methods. Using a designed experiment approach, the interface temperature is shown to be affected by factors including the number of braking applications, the friction coefficient, sliding speed, braking load and friction material. The time-dependent nature of the Tribo-layer formation and the real contact area distribution are shown to be causes of variation in interface temperatures in friction braking. The work extends the scientific understanding of interface contact and temperature during friction braking.

1. INTRODUCTION

Friction brakes are required to transform large amounts of kinetic energy to heat at the contact surfaces between the disc and the pad. The temperature distribution in the friction pair caused by this heat is a complex phenomenon which affects the braking performance directly, and has been investigated by many researchers over many years [1 – 5].

1.1 Multiple levels of contact and local interface temperature

It has been found that measured interface temperatures are generally greater than those which are predicted by single contact models at very low Peclet numbers, due to interacting effects of the multiple heat sources [3]. Using a nominal contact area in an approximate single contact temperature expression allows an average surface temperature rise due to the multiple heat sources to be derived. A similar approach has been employed in surface temperature calculation at high Peclet numbers for temperature-wear maps. In the multiple contact case, which includes most sliding situations involving bodies of finite thickness, there is an additional surface temperature rise (macro-level) besides the localized flash surface temperature found at the contacting asperities (micro-level). Such macro-level temperature increases can affect localized regions over the entire nominal contact area and could be considered to be the nominal contact temperature since these are the regions where sliding contact actually occurs, usually represented as a "hot spot". The temperature at the hot spots can be well above the bulk temperature of the contacting bodies, and, being a nominal contact temperature rise due to frictional heating, is significant in many dry sliding systems, yet is seldom considered in surface temperature calculations. The temperature rise due to local contact at a hot spot has been analysed [3]. At and near the real contact region, a sharp, non-linear temperature drop is caused by the 'small scale' heat flow restriction, which can be modelled as a moving heat source over a semi-infinite medium. The strength of the moving source is the frictional heat entering the body over the real area of contact. Further away from the contact interface a temperature drop is caused by the 'large scale' heat flow restriction. Fourier's heat conduction law governs that temperature drop, with the heat flux in this case being the frictional heat divided by the entire nominal area swept by the moving source. The total, or maximum local, contact temperature was expressed as

$$T_{local} = \Delta T_{local} + T_{nominal} = \Delta T_{local} + \Delta T_{nominal} + T_{background} \quad (1)$$

This shows that local temperature is always higher than the nominal temperature and the background temperature.

1.2 Local interface temperature measurement

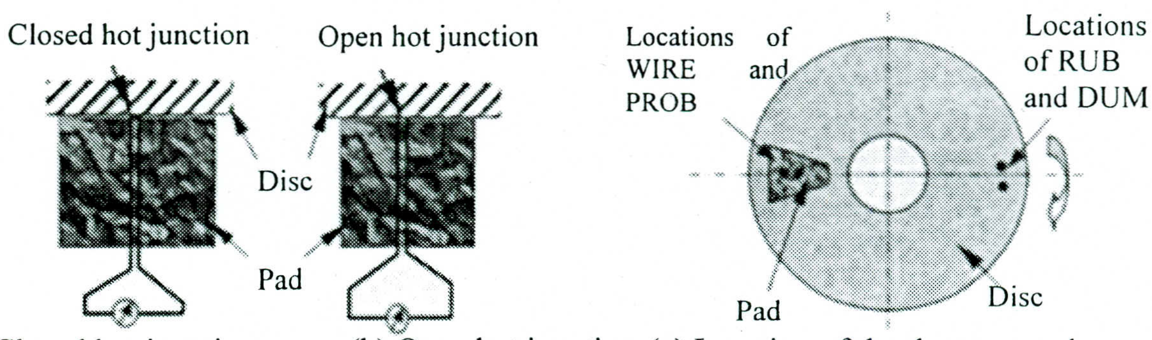
It would be desirable to make interface temperature measurements during actual friction braking tests. However, measuring the interface temperature of friction pair is a difficult task. Several methods have been used to monitor the disc/pad interface temperature [6 - 8], and these can be categorized into two groups: non-contact measurement including methods such as optical and infrared measurement, and contact measurement including methods such as thermocouple and temperature sensitive material coating (or paint). For example, the embedded thermocouple method has been used to investigate thermoelastic instability in an automotive disc brake system experimentally under drag braking conditions [6]. The onset of instability was clearly identifiable through the observation of non-uniformities of temperature measured. The disadvantages of embedded thermocouples include low signal-to-noise ratio due to steep temperature gradients in the pad, and relatively poor time response. A high definition thermal imaging system (Agema Infrared Systems) was used by Allied Signal to assist in the development of new friction materials for brake pads in the automotive, aerospace and rail industries [8]. Surface and near-surface temperatures were monitored at various locations in a disc brake during drag-type testing. The recorded transient temperature distributions in the friction pads and infrared photographs of the rotor disc surface both showed that contact at the friction surface was not uniform, with contact areas constantly shifting due to non-uniform thermal expansion and wear. The disadvantage of the infrared method is its sensitivity to surface emissivity, and affected for example by deposits of wear dust.

In the study presented in this paper, an exposed thermocouple method has been used, which is believed to be a more practical method for measuring the disc-pad interface temperature. The friction braking process is very complex, as many factors affect the frictional forces generated. The complexity is further increased when the interaction between disc and pad at both macro and micro level is included. The main objective of this work is to understand which factors affect most significantly friction braking performance, especially local interface temperature. A statistical design of experiments (DoE) approach was used for the effect analysis in this investigation.

2 EXPERIMENTS

2.1 Measurement system

The thermocouple measurement method, as shown in Fig. 1, is based on the assumptions that the quantity measured by the thermocouple is an aggregate ElectroMotive Force (emf) generated at the thermocouple wire tip and the disc interface, and that the emf obtained by this method corresponds to the local disc and pad interface temperatures. Fig. 1a and 1b show the wire thermocouple (WIRE) configurations used; the thermocouple wire used was 1 mm in diameter. To assess the temperature signals obtained by WIRE, an exposed probe thermocouple (PROB) was used at same time. The WIRE and the PROB thermocouples were located within the disc-pad contact zone, as shown in Fig. 1c. In addition, a conventional rubbing thermocouple (RUB) setup, which is commonly used for monitoring disc temperature, and a dummy thermocouples (DUM) were used for system automatic control and failure diagnosis. The RUB and the DUM were located at a distance from the apparent disc and pad contact zone, as shown in Fig.1c. All these thermocouples, i.e. WIRE, PROB, RUB and DUM, 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 ms^{-1}, and power dissipation in the range 0 – 10 MWm^{-2}. The rotor is a 125 mm diameter disc, and the friction material test sample is approximately 25 mm square. The signals were acquired and processed by a computer aided data acquisition system.



(a) Closed hot junction (b) Open hot junction (c) Location of the thermocouples

Fig.1 Thermocouple temperature measurement

2.2 Design of experiments

After a series of preliminary tests, 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 main outputs (or responses) from this DoE were the interface temperature, T, (measured by WIRE), and the friction coefficient, μ.

2.2.1 Factors and levels

The control factors and factor levels used in this investigation were:

1. Load applied (A): *150* and *300 N*
2. Speed (B): *6.5* and *16.5* ms^{-1} (*1000* and *2500 Rev/min*)
3. Pad materials (C): *M* and *P*
4. Disc/pad surface conditions (D): *SC1* and *SC2*

The pad materials used were *M*: a woven asbestos free friction lining, and *P*: a compression moulded, heavy-duty brake lining for commercial vehicle drum brakes. Disc/pad surface conditions were controlled as *SC1* and *SC2*. Under the test condition *SC1*, where the number of braking applications is from 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*, where the number of braking applications is from 8 to 14, the pad surface was "used" but the disc surface was prepared to the "new" condition before each test, using abrasive paper. The following conditions were set throughout the experiments: (1) loading time is *20 s*; (2) number of applications for each test is 7; (3) preheating load is *450 N*; (4) preheating temperature is *80 ℃* and (5) cooling temperature is *80 ℃*.

2.2.2 *Experimental design*

An $L_{16}(2^4)$ Latin Square [9] was used for the four factor and two level full factorial experiments as shown in Table 2. The 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.

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

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

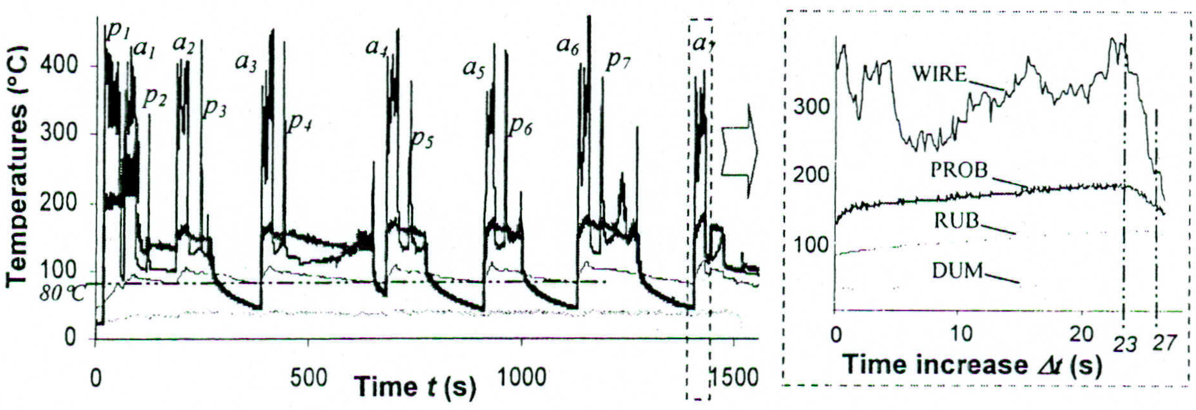
	1	2	3	4	5	6	7	8	9	10
Test No.	A	B	A*B	C	A*C	B*C	D	A*D	B*D	C*D
1	-1	-1	1	-1	1	1	-1	1	1	1
2	-1	-1	1	-1	1	1	1	-1	-1	-1
3	-1	-1	1	1	-1	-1	-1	1	1	-1
4	-1	-1	1	1	-1	-1	1	-1	-1	1
5	-1	1	-1	-1	1	-1	-1	1	-1	1
6	-1	1	-1	-1	1	-1	1	-1	1	-1
7	-1	1	-1	1	-1	1	-1	1	-1	-1
8	-1	1	-1	1	-1	1	1	-1	1	1
9	1	-1	-1	-1	-1	1	-1	-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

3 EXPERIMENTAL RESULTS

3.1 Typical temperature signals and reliability assessment of the measuring system

Fig. 2 shows a general picture of the temperature signals obtained during the friction braking tests using the scale test rig. This shows that the signals responded to the preheating periods

(p_i), brake loading periods (a_i), and the cooling periods correctly. The reliability of the measuring system was assessed by analyzing the RUB signals, which has been commonly used for monitoring disc temperature during braking. As shown in Fig.2a, during individual braking application the temperature indicated by RUB increased from 80 °C, which was the preheating temperature and cooling temperature. After each application, the temperature fell steadily to 80 °C and then increased again as the next application started. At the 7th application under the test T7 (-1, +1, +1, -1) in Fig. 1b, for example, the RUB temperature increased from 80 °C to 120 °C and then fell after the braking load is removed. The RUB temperature increased when the braking load or sliding speed increased as shown in Fig. 3a. The temperature increased by about 30°C, for example, when the speed increases from 1000 rpm (T3) to 2500 rpm (T15) in Fig. 3a. The results based on the RUB temperature measurement indicated that the measuring system was reliable and correctly representative of the test conditions and the braking process. Fig. 3b shows that using different temperature sensors, i.e. WIRE, PROB, DUM and RUB, under the same test condition can result in the different temperature outputs. The signals from different methods responded to the friction braking process showing particular characteristics.



(a) The signals in T7: *ai, pi* the ith application period (b) In the 7th application period

Fig. 2 A typical signal recorded from the test T7 (-1, +1, +1, -1)

Speed is 1000 rpm (A = -1); Load is 300 N (B = +1);
Material is *P* (C = +1); Disc/pad surface condition is *SC1* (D = -1)

3.2 Temperature signal comparisons

3.2.1 Signal response speed

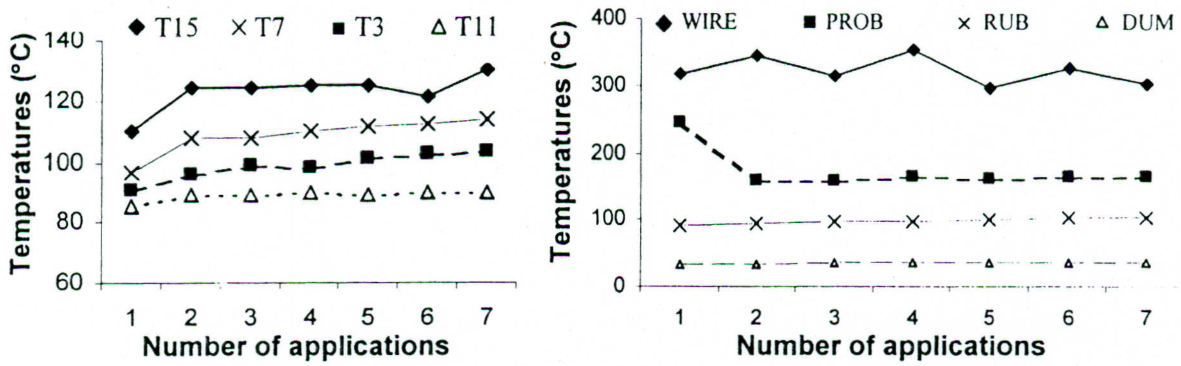
It was found that the temperature signals measured by different measuring methods indicated different behaviour in terms of the signal response speed as shown in Fig. 2b. The temperature signals measured by WIRE and PROB felled immediately at about the 23rd second as the braking load was removed. However, the temperature signal measured by RUB reduced at about the 27th second, which meant a delay of about 4 seconds when the braking load was removed. This was mainly due to RUB being located a distance from the disc and pad contact interface. The WIRE, therefore, is better than the RUB in responding to the braking change between disc and pad during braking process.

3.2.2 Fluctuation of the signals sensed by the WIRE

Fig. 2, and Fig. 3b show that the level of variation of the temperature measured was affected by the measuring methods used. The temperature measured by the WIRE is fluctuated more than those measured by other methods. If it is assumed true that during friction braking the level of contact at local contact area is time dependent, which consequently causes fluctuation in the temperature measured, then this type of information can only sensed by the WIRE.

3.2.3 Magnitude of the temperatures sensed by the WIRE

It was found that the magnitude of the temperature measured was also dependent of the measuring methods used. Under all test conditions, the temperature measured by WIRE was always higher than those measured by other methods under the same test conditions. Under the test T7 (-1, +1, +1, -1) in Fig. 3b, for example, the temperature measured using WIRE was about 225 °C higher than that using RUB. This kind of difference may truly reflect the difference between the disc-pad interface temperature and the disc temperature at some distance outside of the contact zone. The difference between WIRE temperature and PROB temperature is about 150°C. It may be reasonable to assume that WIRE represents the local maximum temperature, T_{local}, the PROB represents the interface nominal temperature, $T_{nominal}$, and the RUB represents the disc background temperature, $T_{background}$, as represented in Equation 1, a relationship of those temperatures. As the result, the temperatures measured by the WIRE were selected to represent the local interface temperatures.



(a) By the RUB under different test conditions (b) By different methods under T7

Fig. 3 Temperatures measured under different test conditions with different methods

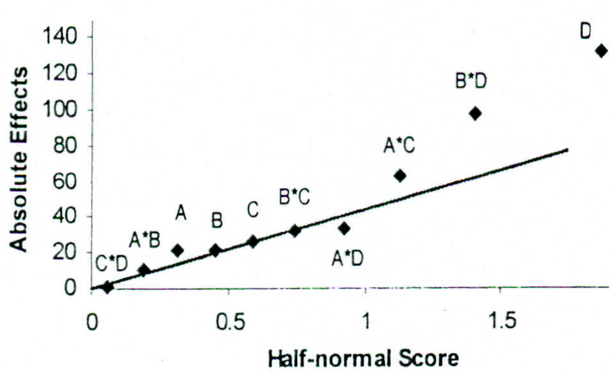


Fig. 4 Daniel plot for temperature

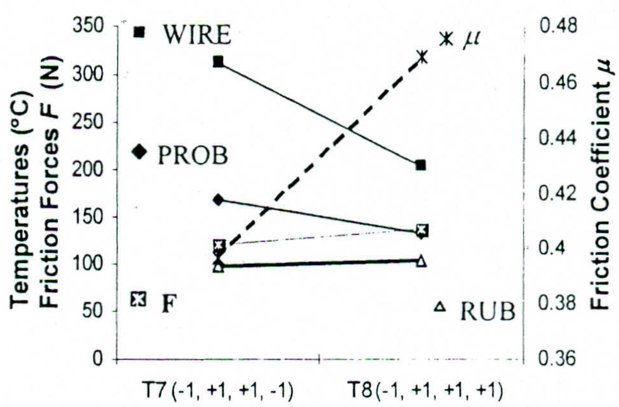


Fig. 5 Effects of the disc/pad surface conditions

3.3 Effects analysis

The temperature measured by the WIRE was used for the effect 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 [9]. In the Daniel plots in Fig. 4, 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. The factor D (the disc/pad surface conditions) and the interaction B*D (interaction between the disc/pad surface conditions and the braking load) appeared on the right-hand side in Fig.4. It appears 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). Daniel plots for friction coefficient also show that the disc/pad surface condition has a strong effect on friction. Further discussion relating to this is shown in the next section.

4 DISCUSSION

4.1 Effects of the disc/pad surface conditions on the interface temperature

A similar observation on the effect of number of braking applications on the friction was reported by Borjesson [10], which showed that friction coefficient was affected by the number of braking applications. And it changed significantly during the first few braking applications. Further analysis of the effects of a number of braking applications on the braking performance indicated that, as shown in Fig. 5 for example, the average local temperatures measured by the WIRE in test condition T7 (i.e. the braking application one to seven) were higher than that in test condition T8 (i.e. the braking application eight to fourteen). In contrast, however, the average background temperatures measured by the RUB in test condition T7 were lower than that in test condition T8. At same time the average friction forces in test condition T7 were lower than that in test condition T8 in Fig. 5. It seems that as the number of application increases, the pad and disc braking friction approach an optimized condition, i.e. generating higher friction torque (or a higher μ) but lower local interface temperature, T_{local}, which is supported by other research [10,11]. To interpret these observations, the change of the contact during the braking has to be studied. It is believed that the Tribo-layer formation and the change of real contact area distribution are the main mechanism in achieving this optimized braking condition.

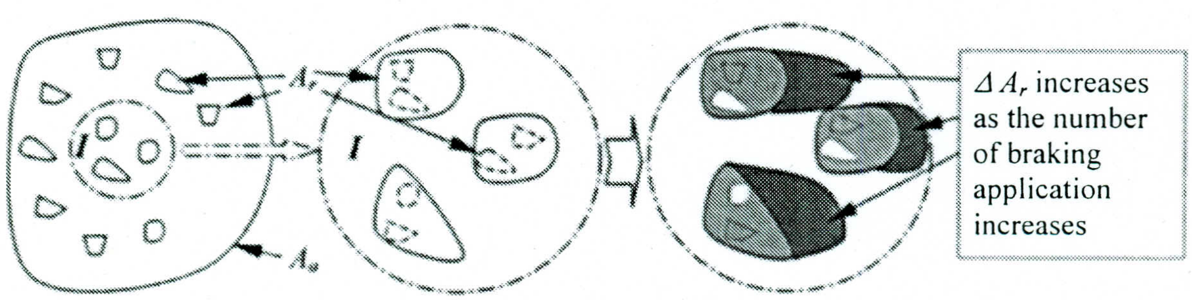


Fig. 6 Shown schematically the wear due to friction causes the increase of the effective contact area A_{ea}

4.2 Effects of Tribo-layer formation and friction braking distance on the interface temperature measured

4.2.1 The change of real contact area distribution and the local interface temperature

The real contact area changes its distribution during braking process for several reasons. When two fresh pad and disc surfaces are in contact under braking load, the pad surface normally only contacts partly, as the two surfaces do not match each other. Wear due to friction will increase the effective contact area A_{ea}. This change will decrease the local maximum temperature as the average normal and friction stress is decreased due to the increase of the effective contact area A_{ea}. Further more, as illustrated in Fig.6, as the number of braking application increases the local real contact area distribution will change at the macro level. Under fresh conditions the pad surface is in an open condition, i.e. the ratio of real contact area A_r and the apparent contact area A_a is small (<< 1). Due to friction and especially the formation of secondary contact plateaux, wear debris fills in the "open" area as described by Eriksson [11], resulting in an increase of the ratio, A_r/A_a. As a result, the local contact stress will decrease, and the local temperature will decrease. The change of the ratio, A_r/A_a, however, cannot explain change of the friction coefficient with the number of the braking applications, as by the friction law, the contact area has no effect on the friction coefficient when $A_r/A_a << 1$.

4.2.2 Tribo-layer formation and its effect on the friction coefficient

It is known that the pad/disc contact surfaces will change their characteristics due to friction, wear and other mechanical or chemical interaction at the interface as the friction braking distance increases. The concept of a Tribo-layer or Tribo-film has been used to distinguish the difference between the surface layer and the main body of the pad and disc [11, 12]. As a result, the friction coefficient, which is a function of the friction pair's properties, will change, and here, it changes in a favourable way, i.e. it increases as the number of braking applications increases. One explanation of this phenomena is that as metal wear debris or metal oxide fills in the "open" area on the pad surface by mechanical or chemical means, the affinity between the two contact surfaces will increase. This makes the two contact surfaces more easily to form an adhesive bond (or welding junction). In order to withstand the friction motion a higher share force are required between disc and pad under the same normal load.

4.3 Discussion on the exposed WIRE thermocouple method used

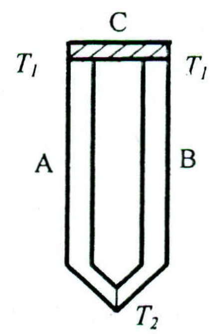
By the law of intermediate metals in thermocouple principle, the algebraic sum of the thermo-electromotive forces (emf) in a circuit composed of any number of dissimilar materials is zero if all the circuit is at a uniform temperature. A third homogeneous material always can be added in a circuit as shown in Fig 7a, with no effect on the net emf of the circuit so long as its extremities are at the same temperature [6, 7]. It follows that any junction whose temperature is uniform and makes good electrical contact does not affect the emf of the thermoelectric circuit regardless of the method employed in forming the junction. The 'open' hot junction configuration used in this investigation is based on this thermocouple principle.

4.3.1 The configuration of the open hot junction

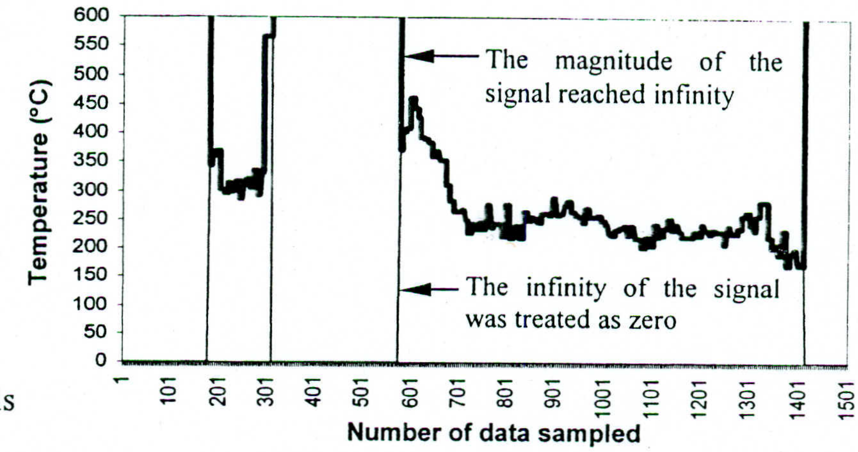
An idealized view of the exposed thermocouple system with "open" and "closed" hot junctions used in this investigation has been given in Fig. 1. The contact between disc and pad during braking is assumed to occur over the hot junction (i.e. at least full contact locally). For the configuration of "open" hot junctions, the disc material bridges the electrical circuit at the hot junction as the material C in Fig 7a. Therefore, during braking, the thermal signal can be recorded as if it were the "closed" configuration. If the disc is removed from contact with pad, however, the electrical circuit is broken. Consequently, the magnitude of the signal reaches infinity as shown in Figure 7b, and the infinity has been treated as zero in the data processing process, as shown in Fig. 7b. Fig. 8 shows the signals obtained by the WIRE (a) with an open hot junction and (b) with a closed hot junction under similar test conditions.

4.3.2 Characteristics of the signals from "open" and "closed" configurations

The results in Fig. 8 shows that the exposed thermocouple technique with the configurations of either closed or open hot junction gives useful data relating to the interaction between the disc and pad during the braking process. The closed hot junction can record the pad surface temperature in braking as well as between braking applications. The open hot junction can detect whether the disc and pad are in contact locally or not. It also can detect whether a Tribo-layer is formed on the disc, based on the assumption that the Tribo-layer formed is a non-conductor.

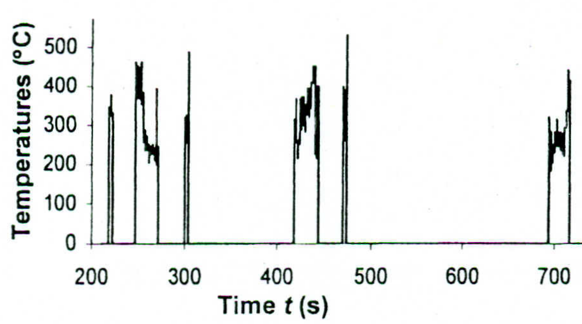


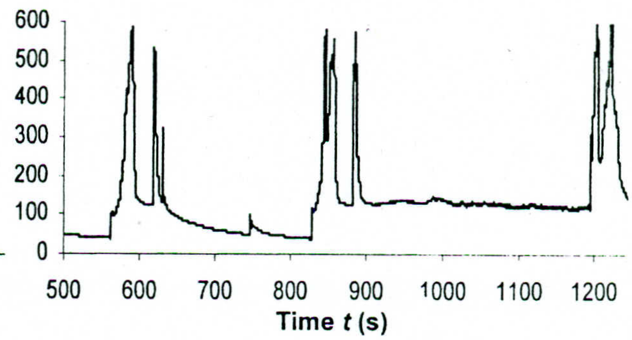
(a) Law of intermediate metals



(b) A signal by the WIRE with an 'open' hot junction

Fig. 7 The exposed thermocouple method with an 'open' hot junction





(a) By the WIRE with an open hot junction (b) By the WIRE with a closed hot junction

Fig. 8 Signal comparison

5 CONCLUSIONS

1. Interface temperatures and friction coefficients have been investigated using thermocouple methods and a statistical designed experiment (DoE) approach. Factors including sliding speed, braking force, and friction material type affect the interface temperatures, and, starting from "new" contacting surfaces, the number of braking applications has the strongest effect alone and through interaction with other factors.
2. The real contact area distribution changes as the number of braking applications increases, and this is one of the main reasons for changes in interface temperature with time of rubbing.

3. The existence of a Tribo-layer, which is generated at the disc/pad contact interface in any braking process, is supported by the temperature measurements.
4. The results show that the exposed thermocouple technique with the configurations of either closed or open hot junction gives very useful data on the interaction between the disc and pad during the braking process. The closed hot junction can record the pad surface temperature in braking as well as between braking applications. The open hot junction can detect whether the disc and pad are in contact locally or not, and can also indicate whether a Tribo-layer is formed on the disc, based on an assumption that the Tribo-layer is a non-conductor. The experiments confirm that the exposed wire thermocouple technique can be used effectively and reliably to measure friction interface local temperatures and their variation during braking.

REFERENCES

1. Santini, J. J.; Kennedy, F. E., Experimental investigation of surface temperatures and wear in disk brakes, Lubr Eng, v 31, n 8, 1975, pp. 402-404, 413-417
2. Day, A. J.; Newcomb, T. P., Dissipation of frictional energy from the interface of an annular disc brake, Proceedings of the Institution of Mechanical Engineers, Part D: Transport Engineering v 198, n 11, 1984, pp. 201-209
3. Tian, X. F., and Kennedy, F. E., Contact surface temperature models for finite bodies in dry and boundary lubricated sliding, Transactions of the ASME, Journal of Tribology, 1993, Vol. 115, pp. 411-418
4. Evtushenko, A.A.; Ivanik, E.G.; Konechny, S., Determination of the effective heating depth of the disc brake pad, Trenie i Iznosv 19, no. 3, 1998, pp. 318-322
5. Wollenweber, K.H., and Leiter, R., Function-monitoring brake system: temperature monitoring brake system, proceedings of the Institution of Mechanical Engineers, International Conference on Braking of Road Vehicles, IMechE 1993, C444/049/93, pp. 23-48
6. Dinc, O. S., C., Ettles, M., Calabrese, S. J. and Scarton, H. A., The measurement of Surface Temperature in Dry or Lubricated Sliding', Transactions of the ASME, Journal of Tribology, 1993, Vol. 115, pp. 78-82
7. Shu, H. H. H., Gaylord, E. W. and Hughes, W. F., The relationship between the rubbing interface temperature distribution and dynamic thermocouple temperature, ASME Journal of Basic Eng., Vol. 86, 1964, pp. 417-422
8. Anon, Material development using infra-red thermography, Metallurgia, v 62, 12, (Dec 1995), pp. 409-410
9. Grove, D. M., Davis, T. P., Engineering quality and experimental design, 1992, Longman Group UK Ltd.
10. Borjesson, M., Eriksson, P., The role of friction films in automotive brakes subjected to low contact forces, Proceedings of the Braking of Road Vehicles, IMechE, 1993, vol. C444/026, pp. 259-268
11. Eriksson, M., and Jacobson, S., Tribological surfaces of organic brake pads, Tribology International 33, 2000, pp. 817-827
12. Yu, L, Bahadur, S., An investigation of the transfer film characteristics and the tribological behaviours of polyphenylene sulfide composites in sliding against tool steel, Wear 214, 1998, pp. 245-251