Wear Simulation and Its Effect on Contact Pressure Distribution and Squeal of a Disc Brake

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SYNOPSIS

In the past, wear at the disc and pads interface of disc brakes has rarely been accounted for in a 3-dimensional finite element (FE) model for studying brake squeal. Thus, its effect on disc brake squeal is only possible to investigate through experimental methods. In the present paper, wear over time at the pads interface is simulated using a modified wear rate formula. Confirmation of the proposed wear formula is made against experimental results.

The surface topographies of two brand new pairs of brake pads are measured. The contact tests using pressure-indicating films are carried out in order to capture static pressure distributions. The same brake pads are tested under several braking applications. For each braking application, the static contact pressure distribution is measured. The results are used to compare with the simulated results predicted by the FE model. To do so, the detailed 3-dimensional finite element model of a real disc brake is developed and validated through appropriate analyses.

The paper also investigates squeal generation in the above braking applications using complex eigenvalue analysis that is available in a commercial software package. The predicted results are then compared to the squeal events observed in the experiments.

1 INTRODUCTION

Wear can take place when two or more bodies in frictional contact slide against each other. The significant effect of wear, particularly in friction material of a disc brake system, is reduction of its life span. The more the wear, the sooner the friction material needs to be replaced. There are various wear laws established from experimental results. For instance,

Archard (1) proposed that wear rate (displacement per unit time) $\Delta \dot{h}$ for general applications could be given in the following expression:

$$\Delta \dot{h} = \frac{\Delta h}{t} = \frac{k}{H} P v \qquad (1)$$

where Δh is the wear displacement, t is the sliding time, k is the wear coefficient, H is the hardness of tested material, P is the applied pressure and v is the sliding speed.

For a specific application such as brakes systems, Rhee (2) developed an empirical wear equation at fixed temperature where he expressed the mass loss of the polymer friction material in terms of the contact normal force F, the speed v and the time t as follows

$$\Delta W = k F^a v^b t^c \qquad (2)$$

where k is the wear rate coefficient obtained from experiments and a, b and c are the set of parameters that are specific to the friction material and the environmental conditions. He stated that a, b and c could be unity in a special case but suggested in (3) that for a commercial friction material $a = 0.42$ and $c = 1.0$.

For composite friction materials, Pavelescu and Musat (4) proposed that the wear intensity I_u without temperature effect could be in the following form:

$$\frac{\Delta h}{L} = I_u = k F^a v^b \qquad (3)$$

where L is the sliding distance and k, a and b are the constant. Those constants vary for different materials.

Another wear law that incorporated the effect of temperature was studied in (5). The authors of (5) reported that mass loss of the aircraft brake material was proportional to the surface temperature T relative to the melting point T^*:

$$\Delta W = k \frac{T}{T^* - T} \qquad (4)$$

and

$$T = \beta F^a t^b \qquad (5)$$

where β, a and b are the parameters that are related to the frictional coefficient and to other properties of the material involved, k is a wear coefficient. In their investigation they found that $k = 39.8$, $T^* = 1083\text{K}$, $\beta = 9.1$, $a = 0.5$ and $b = 0.3$ gave the best fit of the experimental data.

In a recent investigation, Hohmann et al (6) developed their own wear law based on experimental data. They suggested that the wear displacement of friction material was a function of pressure, temperature, sliding time and apparent contact area A as

$$\Delta h = \sum (c_{ij} P^i T^j) \frac{t}{A} \qquad\qquad \text{with } i=1,\ldots 4 \text{ and } j= 0,\ldots.3 \qquad\qquad (6)$$

Based on the afore-mentioned formula they computed wear displacement using a 2-dimensional FE model and reported that the predicted results were close to experimental ones.

Since there are a number of wear laws available in the literature, it is possible to select one of them and simulate it numerically. Computer simulations of the wear process for brake friction materials and other applications can be found in (6-9) and (10-12) respectively. The authors of (10-12) favoured Archard's wear law (1) while Bajer et al (8) used a very simple wear formula which is a linear function of a constant k and the local contact pressure p for their disc brake FE model. AbuBakar et al (9) on the other hand used modified Rhee's wear formula (2) and assumed constants a, b and c as unity. Barecki and Scieszka (7) similarly used almost the same empirical wear equation of Rhee (2) for their winding gear, post-type brake. Of those computer simulations, the authors of (8,9) did not make comparison with experimental results.

The effect of wear on squeal generation has been studied experimentally by many researchers (13-16). Unfortunately there is very little investigation by means of numerical methods. The authors of (8,9) recently attempted to relate wear with squeal generation using the finite element method. Bajer et al (8) reported that considering the wear effect, predicted unstable frequencies were close to experimental ones. AbuBakar et al (9) used wear simulations to investigate the fugitive nature of disc brake squeal. In this paper, wear simulations are performed for two pairs of brake pads. The wear formula proposed by Rhee (2) is adopted and then modified in order to suit with current investigation. Instead of verifying wear displacement as has been done by the authors of (6-7,10-12), this paper attempts to verify the wear progress predicted in the simulations using measured static contact pressure distributions from contact tests. Pressurex $^®$ pressure-indicating film and Topaq$^®$ Pressure Analysis system are used to obtain pressure distributions and magnitude. From the comparison, correct values of constants a, b and c can be obtained. Then complex eigenvalue analysis is used to predict unstable frequencies under various wear conditions. The predicted results are then compared with the squeal events observed in the experiments.

2 CONTACT TESTS

In order to verify predicted results from FE analysis, Pressurex$^®$ Super Low (SL) pressure indicating film, which can accommodate local contact pressure in the range of between 0.5 ~ 2.8 MPa is used. The tested film as shown in Figure 1 can only provide stress marks but cannot reveal its magnitude. It is necessary in this work that both qualitative and quantitative measurements are taken. To do so, a post-process interpretive system called Topaq$^®$ Pressure Analysis that can interpret the stress marks is utilised. The system is accurate to within $\pm 4\%$, which is very accurate in the field of tactile pressure measurement (17).
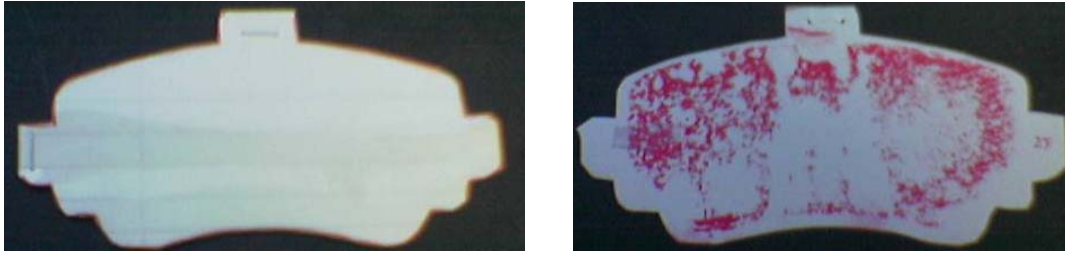
Figure 1. Pressure indicating film before (left) and after (right) testing

In this study, two brand new pairs of brake pads from the same manufacturer are tested. Hence, it is assumed both brake pads have the same specifications except for surface roughness. The arithmetic mean deviation R_a is about 62.8μm (finger pad of pad 1), 64.4 μm (piston pad of pad 1), 38.3μm (finger pad of pad 2) and 45.0μm (piston pad of pad 2). There are three stages of wear tests under a brake-line pressure of 1 MPa at a rotational speed of 6 rad/s. In the first stage, these brake pads are run for 10 minutes. Another 10 minutes are used in the second stage. In the third and last stage, the brake pads are run for 60 minutes. At the end of each stage, contact tests are conducted where the disc brake assembly is subjected to a brake-line pressure of 2.5 MPa with a stationary disc (static condition). Details of wear investigation are given in Table 1.

The seemingly short durations of wear tests are due to a numerical consideration. In the wear formula of equation (2) used in this paper, the duration of wear, t, must be specified. The longer the duration of wear, the more the wear and the greater change of the contact pressure. If t is too big, there will be numerical difficulties in an ABAQUS run. It has been found through trial-and-error that $t = 200$ s gives reasonably good results and good efficiency. Consequently a simulation of 10-minute wear means four ABAQUS runs. In line with this numerical consideration, wear tests have not lasted for numerous hours as normally done in a proper wear test or a squeal test. In theory, however, numerical simulations of wear may cover an arbitrary length of time.

Figures 2a and 2b show static contact pressure distributions for brake pad 1 and brake pad 2 respectively. It is shown that contact pressure distributions are slightly different between the two pairs of pads. This might be due to surface topographies of those new brake pads that are initially different. It can be seen that distributions of contact pressure vary and the contact areas of pad 1 and pad 2 increase as wear progresses with time. It also can be seen that the maximum contact pressure (in red colour) is located mostly at the outer region of the brake pads. Squeal generation is also examined for each stage of braking application. During the wear tests a squeal frequency at about 4 kHz is found in the last stage of braking application, i.e. at 80 minutes for both pads. However, such a squeal frequency is not observed in the early two stages as described in Table 2. Incidentally, this disc brake tended to squeal at 2.4 kHz, 4 kHz and 7.8 kHz in the past.
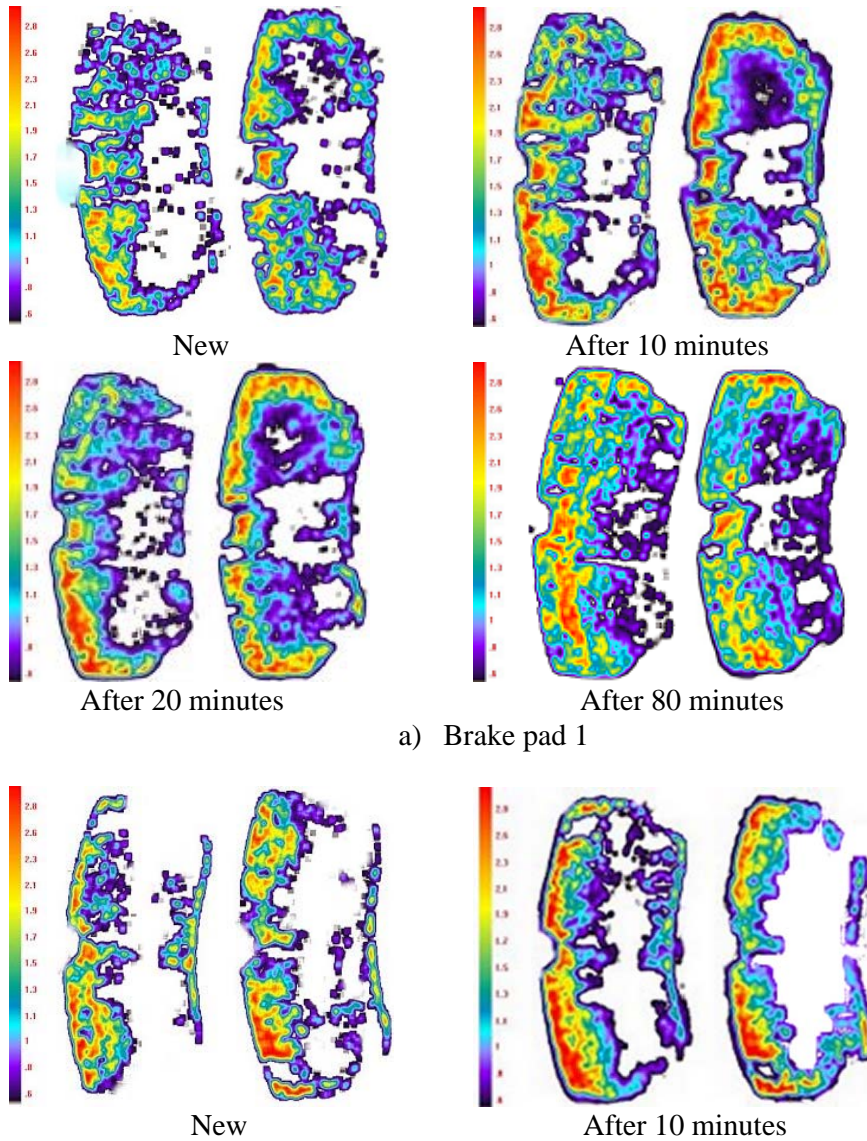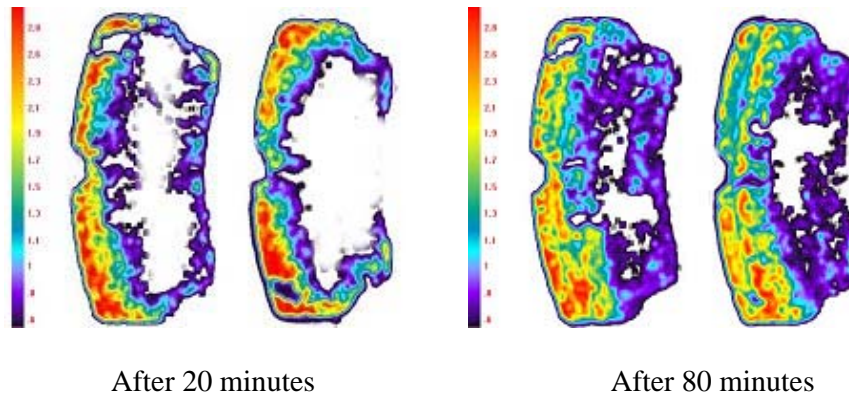
Table 1. Configurations of tested pad

Identification	Instants in Wear Tests (minutes)			
Brake pad 1	0 (New)	10	20	80

| Brake pad 2 | 0 (New) | 10 | 20 | 80 |

Table 2. Squeal observation during braking applications

Length of braking application (minutes)	Brake pad 1	Brake pad 2
10	No	No
20	No	No
80	Yes	Yes

New

After 10 minutes

After 20 minutes

After 80 minutes

a) Brake pad 1

New

After 10 minutes

After 20 minutes   After 80 minutes

b) Brake pad 2

Figure 2. Contact pressure distribution at the piston pad (left) and finger pad (right). (top of each diagram is the leading edge)

3 FINITE ELEMENT MODEL

A detailed 3-dimensional finite element (FE) model of a Mercedes solid disc brake assembly is developed as shown in Fig. 3a. The finite element (FE) model consists of a disc, two pads, a calliper, a carrier, a piston and two guide pins. A rubber seal (attached to the piston) and two rubber washers (attached to the guide pins) in this brake assembly are not included in the FE model. Damping shims are also absent since they have been removed in the squeal experiments. The FE model uses up to 8350 solid elements and approximately 37,100 degrees of freedom (DOFs). Figure 3b shows a schematic diagram of contact interaction that has been used in the disc brake assembly model. Surface based elements are used at the disc/pad interfaces while spring elements are used for other contact interactions between the disc brake components. A rigid boundary condition is imposed at the boltholes of the disc and of the carrier bracket, where all six degrees of freedom are rigidly constrained. The FE model has been verified through three validation stages. Details of the validation processes are given in (9).
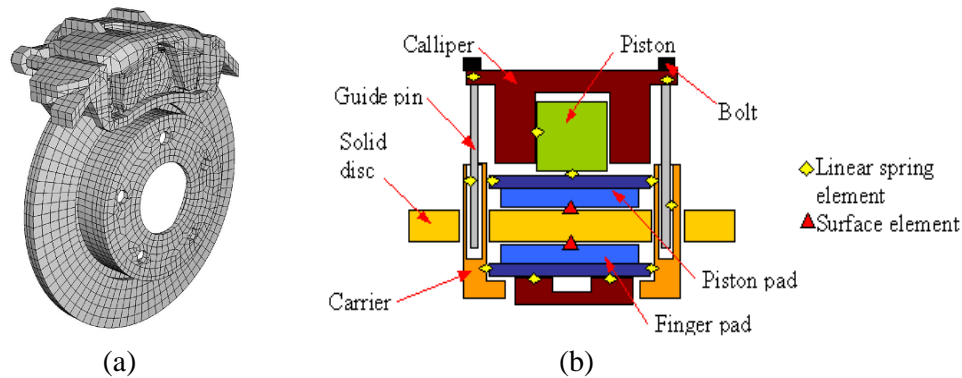

(a)   (b)

Figure 3. FE model of the disc brake and contact interactions

In this work, actual surfaces of the piston and finger pads are measured and considered. A Mitutoyo linear gauge LG-1030E and a digital scale indicator are used to measure and provide readings of the surface respectively. Using those measurement data the normal

coordinates of the nodes of piston and finger pads in the brake pad interface model are adjusted. Figure 4 shows those surface topographies of the piston and finger pads.
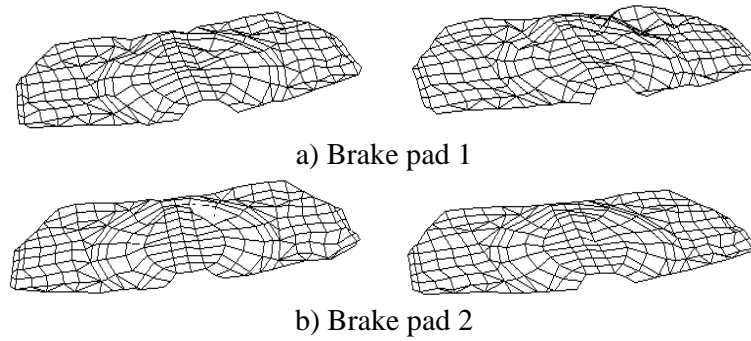

a) Brake pad 1

b) Brake pad 2

Figure 4. A real surface topography of the piston pad (left) and finger pad (right)


4. WEAR SIMULATION AND STABILITY ANALYSIS

Rhee's original wear formula (2) cannot be used in the present investigation. Since mass loss due to wear is directly related to the displacement that occur on the rubbing surface in the normal direction, Rhee's wear formula is modified as:

$$\Delta h = kP^a (\Omega r)^b t^c$$

where Ω is the rotational speed (rad/s), r is the pad mean radius (m), and a, b and c are all constants which remain to be determined. In the simulation the wear rate coefficient is set to $k = 1.78 \times 10^{-13} \, \text{m}^3/\text{Nm}$ (18) and $r = 0.111$ m. Contact analysis of the baseline model is performed similarly to the operating conditions of the experiments described in section 2. From the contact analysis, contact pressure can be obtained and hence wear displacements can be calculated. Having obtained the wear displacements, nodal coordinates at the friction interface model in the axial direction are adjusted. This process continues until it reaches braking application period of 80 minutes. Having simulated for various values of constants a, b and c, it is found that the wear displacement is given by:

$$\Delta h = kP^{0.9} \Omega r t$$

These constant values seem to give a close prediction of contact pressure distributions to the experimental ones as shown in Figure 5.

New

After 10 minutes

After 20 minutes

After 80 minutes
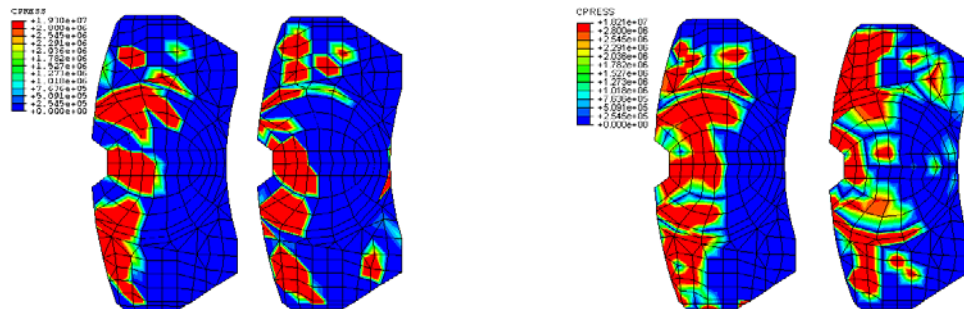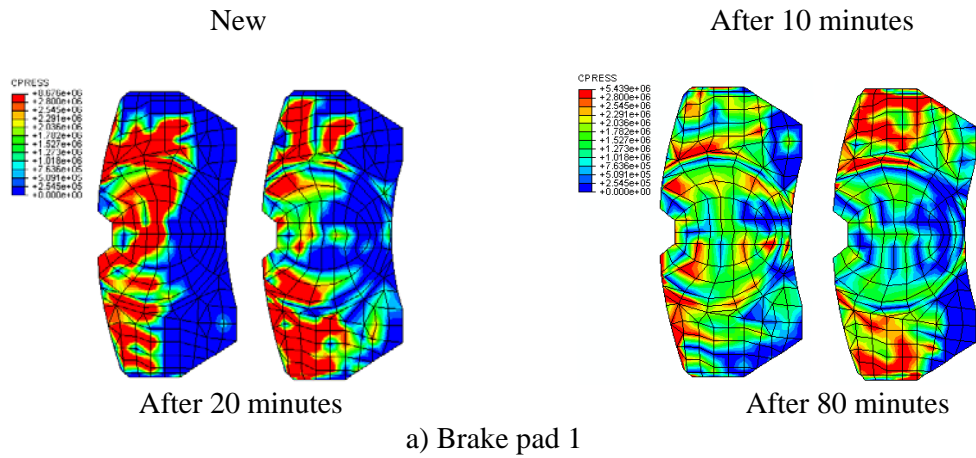
a) Brake pad 1

Figure 5. Predicted Contact pressure distribution at the piston pad (left) and finger pad (right) (top of each diagram is the leading edge)

New

After 10 minutes

After 20 minutes

After 80 minutes
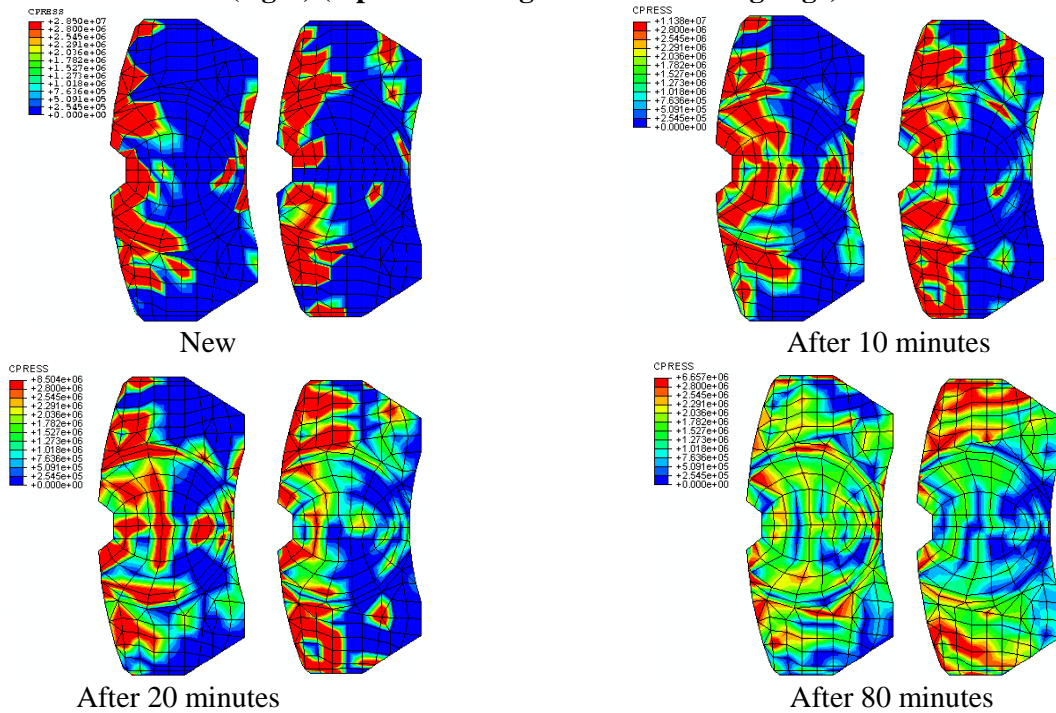
b) Brake pad 2

Figure 5 (cont'd). Predicted Contact pressure distribution at the piston pad (left) and finger pad (right)

Having completed wear simulations for all stages of braking application, stability analysis is performed using complex eigenvalue analysis. A similar operating condition to that of the experiments is applied. Kinetic friction coefficient is set to $\mu_k = 0.393$ which is obtained from the experiments. From complex eigenvalue analysis, it is found that there is an unstable frequency around 4.0 kHz predicted during 80 minutes of braking application for both brake pads. Significantly, for the first two stages of braking application such an unstable frequency is not predicted. Hence these results are in reasonably good agreement with the observation made in the experiments.

Table 3. Predicted unstable frequencies

Braking application (minutes)	Brake pad 1		Brake pad 2	
	Frequency (kHz)	Real Part	Frequency (kHz)	Real Part
10	-	-	-	-
20	-	-	-	-
80	4.2	+39.8	4.0	+24.6

CONCLUSIONS

This paper presents numerical simulation of wear using the finite element method. Modified Rhee's wear formula is adopted and used to calculate wear displacements for a particular braking period. Prior to the simulation contact tests are carried out in the actual disc brake assembly. Static contact pressure distributions are measured using Pressurex® pressure indicating film and then analysed by Topaq® pressure system to get better visualisation and results of the pressure distribution. The results show that the contact area increases as wear progresses. Results from the contact tests are then compared to the contact pressure distribution predicted from finite element analysis. It is found that the values of $a = 0.9$, $b = 1.0$ and $c = 1.0$ in the modified Rhee's wear formula give a close prediction to the experimental ones. Reasonably good agreement is also found in the squeal prediction. This suggests that Rhee's wear formula can be used in the finite element analysis in order to examine wear of the friction material and in turn to predict squeal generation.

ACKNOWLEDGEMENTS

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