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

**A R ABU BAKAR, L LI, S JAMES and H OUYANG**
Department of Engineering, the University of Liverpool, UK
**J E SIEGEL**
Sensor Products LLC, USA

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 has been investigated largely through experimental methods. In the present paper, wear over time at the pad 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 braking applications of three durations. For each braking application, the static contact pressure distribution is measured. The results are used to compare with the simulated results predicted by the 3-dimensional finite element model of a real disc brake, which has 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 and they are in reasonably good agreement.

## 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:

$$\Delta\dot{h} = \frac{\Delta h}{t} = \frac{k}{H} Pv \tag{1}$$

where $\Delta h$ 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 normal force $F$ at the contact interface, the speed $v$ and the time $t$ as follows:

$$\Delta W = kF^a v^b t^c \tag{2}$$

where $k$ is the wear rate coefficient obtained from experiments and $a$, $b$ and $c$ are a set of parameters that are specific to the friction materials 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:

$$\frac{\Delta h}{L} = I_u = kF^a v^b \tag{3}$$

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 $T^*$:

$$\Delta W = k \frac{T}{T^* - T} \tag{4}$$

and

$$T = \beta F^a t^b \tag{5}$$

where $\beta$, $a$ and $b$ are parameters that are related to the frictional coefficient and other properties of the materials involved, $k$ is a wear coefficient. In their investigation they found that $k$ = 39. 8, $T^*$ = 1083K, $\beta$= 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 a friction material was a function of pressure, temperature, sliding time and apparent contact area $A$ as

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

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 compare their numerical results with experimental ones.

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 the 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 the 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 information is found. 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 ±4%, which is very accurate in the field of tactile pressure measurement [17].

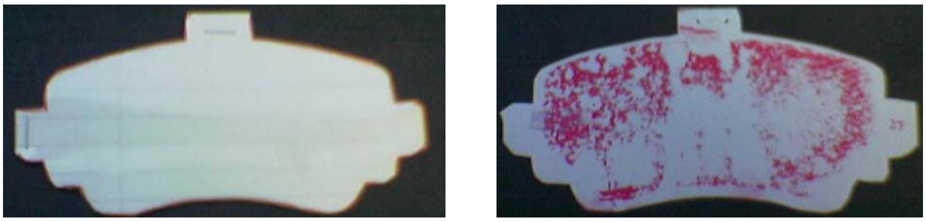


**Figure 1. Pressure indicating film before (left) and after (right) testing**

In this study, two brand new pairs of brake pads from the same manufacturer are tested. It is assumed that both pads have the same specifications except for their 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 a wear test 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 stationary disc is subjected to a brake-line pressure of 2.5 MPa. Details of the wear tests 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 $t$ = 200s gives reasonably good results and also good efficiency. Consequently a simulation of 10-minute wear means three 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 progress may cover an arbitrary length of time.

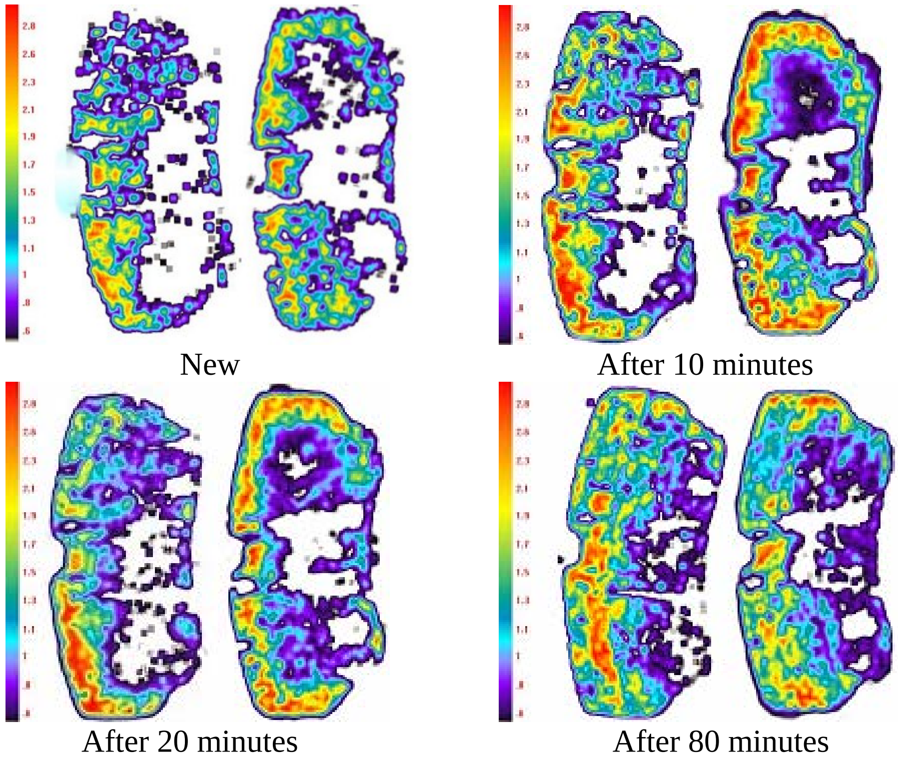
Figures 2a and 2b show static contact pressure distributions for brake pad 1 and brake pad 2 respectively. It is shown that contact pressure distributions are slightly different between the two pairs of pads. This might be due to surface topographies of these new brake pads that are initially different. It can also be seen that contact pressure distributions vary and the contact areas of pad 1 and pad 2 increase as wear progresses with time. It can also 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 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 previous independent squeal tests that are unrelated to the current investigation.

**Table 1. Configurations of tested pad**

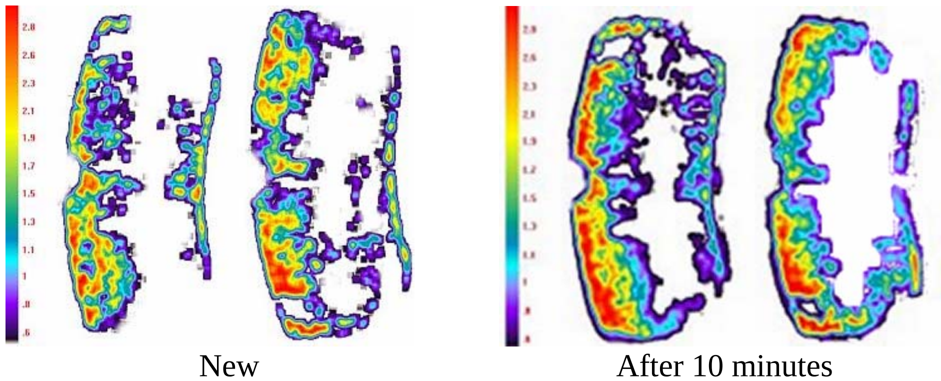
| Identification | Instants in Wear Tests (minutes) | | | |
|---|---|---|---|---|
| Brake pad 1 | 0 (New) | 10 | 20 | 80 |
| Brake pad 2 | 0 (New) | 10 | 20 | 80 |

**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 |

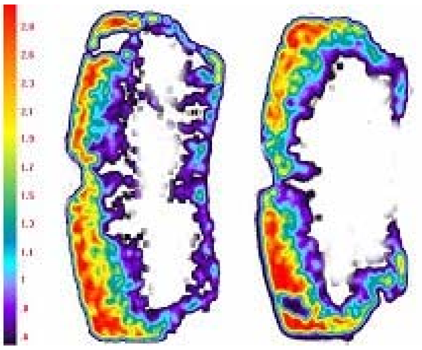



a) Brake pad 1

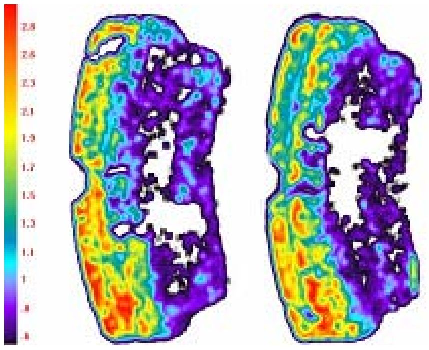



b) Brake pad 2

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



After 20 minutes



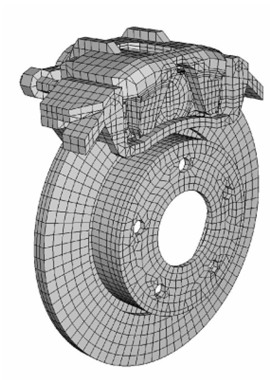
After 80 minutes

b) Brake pad 2

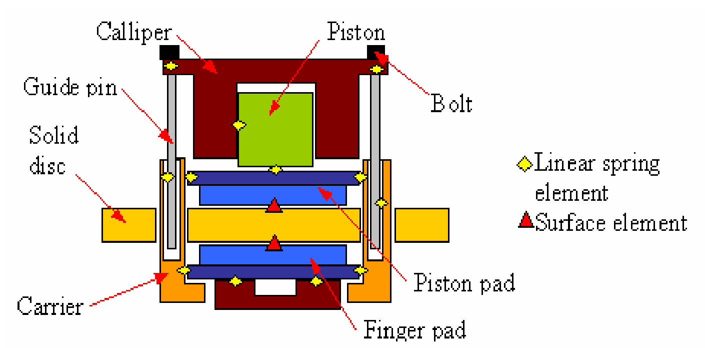
**Figure 2(cont'd). Contact pressure distribution at the piston pad (left) and finger pad (right). (top of each diagram is the leading edge)**

## 3. FINITE ELEMENT MODEL

A detailed 3-dimensional finite element model of a Mercedes solid disc brake assembly is developed as shown in Fig. 3a. The finite element 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].



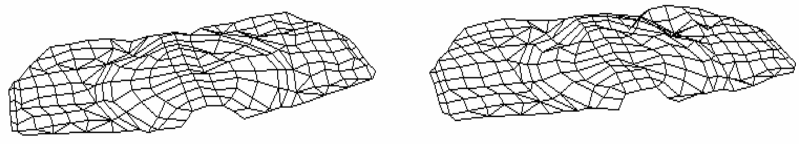
(a)



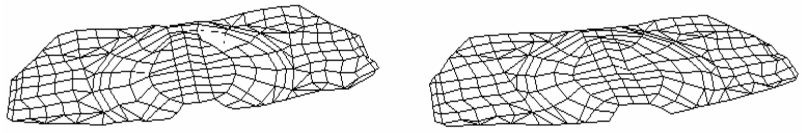

(b)

**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 FE 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 occurs 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 $\Omega$ 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 $k = 1.78\text{x}10^{-13}\,\text{m}^3/\text{Nm}$ [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 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:

$$\Delta h = kP^{0.9}\Omega rt$$

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

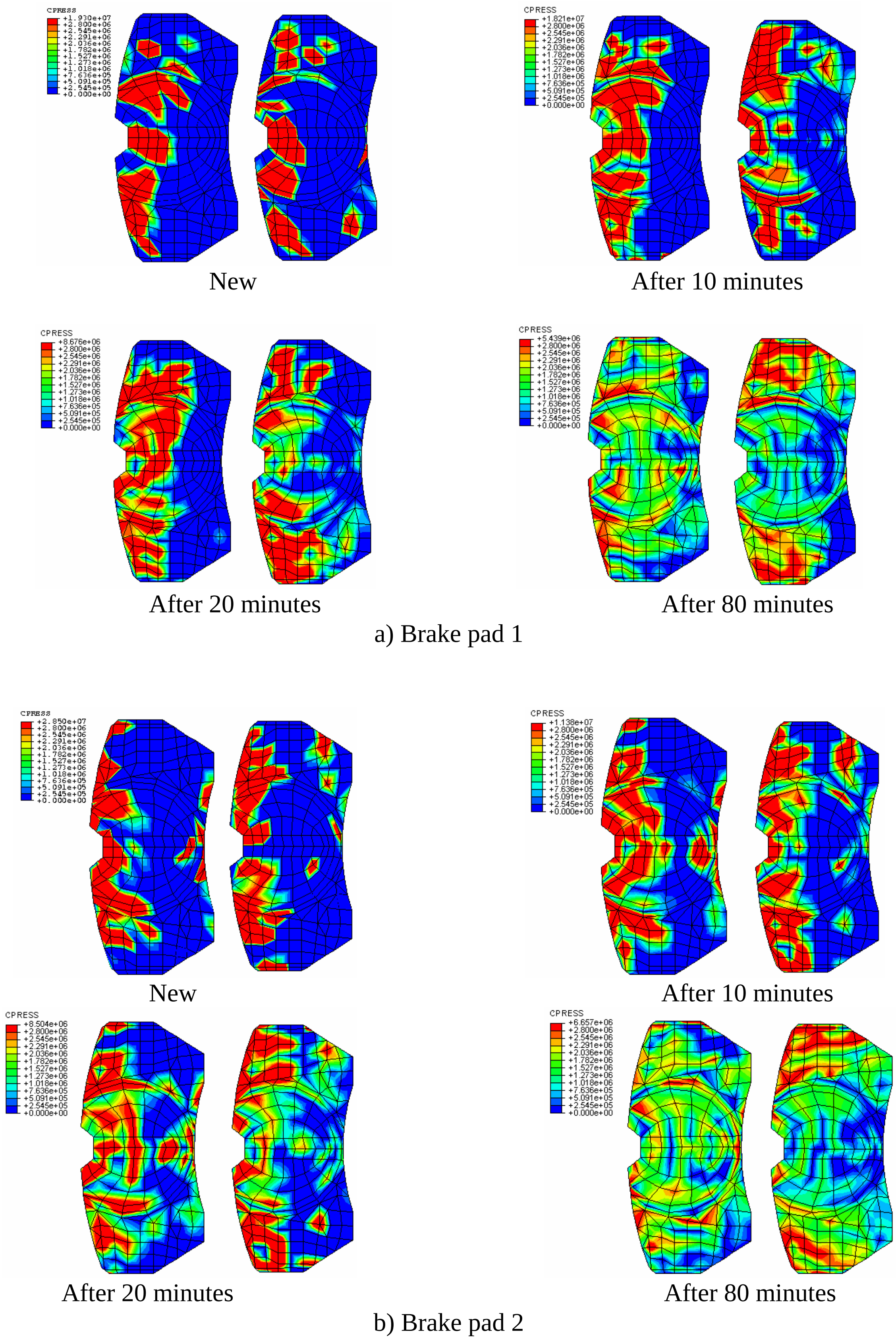



**Figure 5. Predicted Contact pressure distribution at the piston pad (left) and finger pad (right)**

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 towards the 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 |

## 5. CONCLUSIONS

This paper presents numerical simulation of wear using the finite element method. Rhee's wear formula is modified 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 a 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 fairly 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

The first author wishes to thank Sensor Products LLC for donating pressure-indicating film and images and Universiti Teknologi Malaysia for the financial support.


## REFERENCES


**[1] Archard, J. F.** Contact and rubbing of flat surfaces, *Journal of Applied Physics*, 24 (8), pp. 18-28, 1953.

**[2] Rhee, S. K.** Wear equation for polymers sliding against metal surfaces, *Wear*, 16, pp. 431-445, 1970.

**[3] Rhee, S. K.** Wear mechanisms for asbestos-reinforced automotive friction materials, *Wear*, 29 (3), pp. 391-393, 1974.

**[4] Pavelescu, D. and Musat, M.** Some relations for determining the wear of composite brake materials, *Wear*, 27, pp. 91-97, 1974.

**[5] Ho, T. L. and Peterson, M. B.** Wear formulation for aircraft brake material sliding against steel, *Wear,* 43, pp. 199-210, 1977.

**[6] Hohmann, C., Schiffner, K. and Brecht, J.** Pad wear simulation model, *SAE Technical Paper*, 1999-01-3392, 1999.

**[7] Barecki, Z. and Scieszka, S. F.** Computer simulation of the lining wear process in friction brakes, *Wear*, 127, pp. 283-305, 1988.

**[8] Bajer, A., Belsky, V. and Kung, S. W.** The influence of friction-induced damping and nonlinear effects on brake squeal analysis, *SAE Technical Paper*, 2004-01-2794,2004.

**[9] AbuBakar, A. R., Ouyang, H., Li, L. and Siegel, J. E.** Brake pad surface topography part II: squeal generation and prevention, *SAE Technical Paper*, 2005-01-3935.

**[10] Podra, P. and Andersson, S.** Simulating sliding wear with finite element method, *Tribology International*, 32, pp. 71-81, 1999.

**[11] Molinari, J. F., Ortiz, M., Radovitzky, R. and Repetto, E. A.** Finite element modeling of dry sliding wear in metals, *Engineering Computations*, 18 (3/4), pp. 592-609, 2001.

**[12] Kim, N. H., Won, D., Burris, D., Holtkamp, B., Gessel, G. R., Swanson, P. and Sawyer, W. G.** Finite element analysis and experiments of metal/metal wear in oscillatory contacts, *Wear*, 258 (11/12), pp. 1787-1793, 2005.

**[13] Eriksson, M., Bergman, F. and Jacobson, S.,** Surface characterization of brake pads after running under silent and squealing conditions, *Wear*, 232 (2), pp. 163-167, 1999.

**[14] Jibki, T., Shima, M., Akita, H. and Tamura, M.,** A basic study of friction noise caused by fretting, *Wear*, 251, pp. 1492-1503, 2001.

**[15] Chen, G., Zhou, Z., Kapsa, P. and Vincent, L.** Effect of surface topography on formation of squeal under reciprocating sliding, *Wear*, 253, 411-423, 2003.

**[16] Sherif, H. A.** Investigation on effect of surface topography of disc/pad assembly on squeal generation, *Wear*, 257 (7/8), pp. 687-695, 2004.

**[17]** http://www.sensorprod.com/topaqvfuji.html

**[18] Jang, H., Ko, K., Kim, S. J., Basch, R. H. and Fash, J. W.** The effect of metal fibers on the friction performance of automotive brake friction materials, *Wear*, 256, pp. 406-414, 2004.