WEAR ANALYSIS ON CONTACT PRESSURE OF DISC BRAKE/SQUEAL GENERATION

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Abstract: **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 was only possible to investigate through experimental methods. In this 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.**

Keywords: **disc brake squeal, experimental methods, formula, braking application.**

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 reason for its life span. The more the wear, the sooner the friction material needs to be replaced.

Disc brakes are safety-critical components whose performance depends strongly on the contact conditions at the pad to rotor interface. When the driver steps on the brake pedal, hydraulic fluid is pushed against the piston of caliper, which in turn forces the brake pads into contact with rotor. The frictional forces at the sliding interfaces between the pads and the rotor retard the rotational movement of the rotor and the axle on which it is mounted. The kinetic energy of the vehicle is transformed into heat that is mainly absorbed by the rotor and the brake pad.

The pad to rotor interface can be classified as a conformal dry sliding contact. During braking a point on the brake pad is in constant contact with the rotor, whereas a point on the rotor experiences intermittent contact. A natural consequence of sliding contact is that both pads and rotor are worn, affecting the useful life of brake as well as its behavior.

This paper discusses how wear of the pad to rotor interface can be predicted using general-purpose finite element analysis software. A 3-D finite element model of the brake pad and the rotor is developed to calculate the pressure distribution in the pad to rotor contact. A wear simulation procedure based on a generalized form of Archards wears law is adopted to simulate the macroscopic wear of the brake pad.

2. WEAR LAWS

There are various wear laws established. For instance, Archard proposed that wear rate (displacement per unit time) Δh for general applications could be given in the following expression:

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

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

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

$$\Delta w = kF^a v^b t^c$$

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 that for a commercial friction material a=0.42 and c=1.0

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

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

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

In this paper we are considering Archards wear law to calculate the wear rate. The diagram of a completely worn out pad is given below:

3. BRAKE PAD LINING LIFE SPAN

Brake pad lining life span can be found out in two common ways: 1) Mileage 2) Year

Driving Pattern:

1) A highway or motorway brakes pad last up to 60,000 miles.

2) Busy traffic or city between 25,000 to 30,000 miles.

Load Transferred:

1) Front brakes pad last between 30,000 and 60,000 miles.

2) Rear brakes pad last between 40,000 and 70,000 miles.

***List of Minimum Thickness Allowed Of Brake Pad:***

1) Brake pads should typically be replaced when approximately 1/8" to of lining remains on the steel backing plate.

2) Brake pad should be replaced at least 2mm lining left on the pad

3) If the lining appears to be 25% from its original thickness, replacement is needed.

4) Pad lining should be replaced if the friction material has worn down to a thickness of 1.6mm above the shoe platform.

5) Pad material must be replaced when the pad lining not be less than 1.5mm thick at any point.

Abrasive Wear:
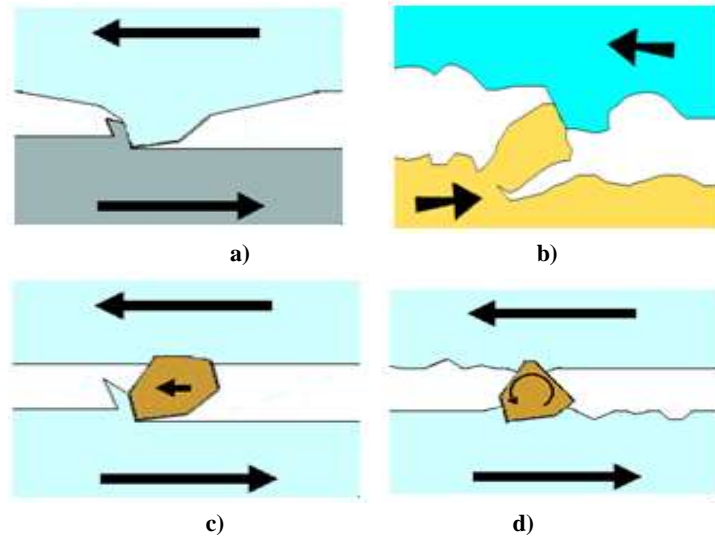
(a) Two surface rubbing each other.

(b) Harder surface cut material away from second surface

(c) Debris (removed materials) changes to three body abrasion.

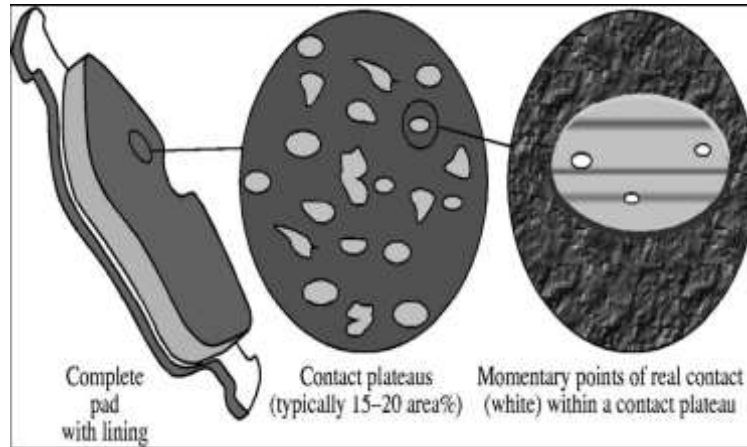 (d) Wear debris then acts as an abrasive between two surfaces.

4.   OVERALL SCHEME OF THE PROJECT
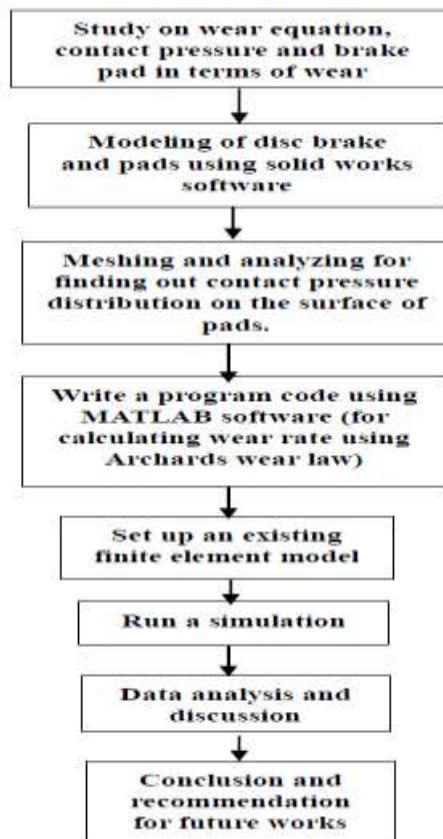
5. CONTACT PRESSURE

Contact pressure should be kept as uniform as possible in order to minimize tapered wear.

Topography of a brake pad shows the contact plateaus rising typically a few microns above the rest of the surface.


General view of the brake pad area and its division into contact plateaus and areas of real contact within the contact plateaus

These plateaus are based on the more wear resistant constituents on the pad such as fibers and abrasive particles. The rough surface topography will cause contact pressure between brake pad and disc did not distribute uniformly at each point.
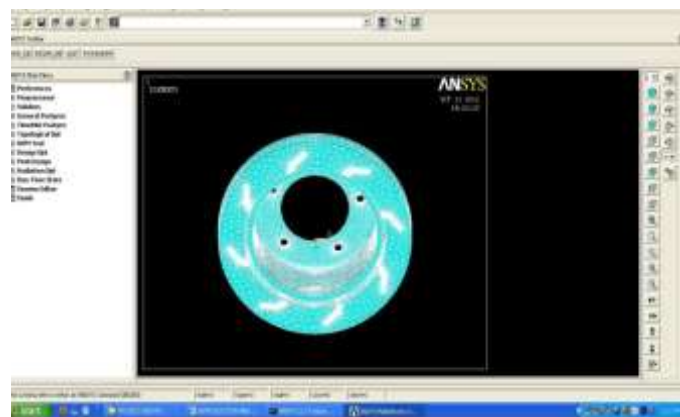
6. FINITE ELEMENT MODEL

A detailed 3-dimensional finite element model of the solid disc brake assembly is developed as shown below. The finite element model consists of a disc and two pads. The FE model uses up to 8350 solid elements and approximately 37,100 degrees of freedom. Surface based elements are used at disc/pad interface. Schematic diagram of contact interaction that has been used in the disc brake assembly model is also shown below. A rigid boundary condition is imposed at the bolt holes of the disc and of the carrier bracket where all six degrees of freedom are rigidly constrained. In this work actual surfaces of the finger pad and piston pad are measured and considered. Using those measurement data the normal coordinates of the nodes of piston and finger pads in the brake pad interface model are adjusted.
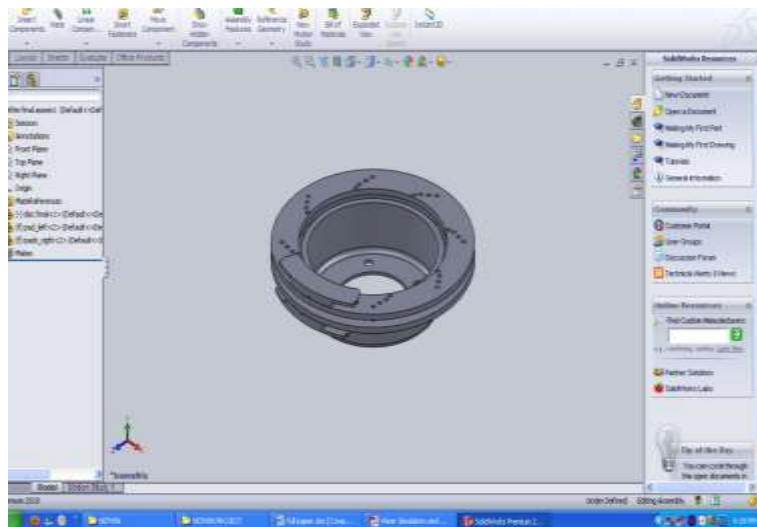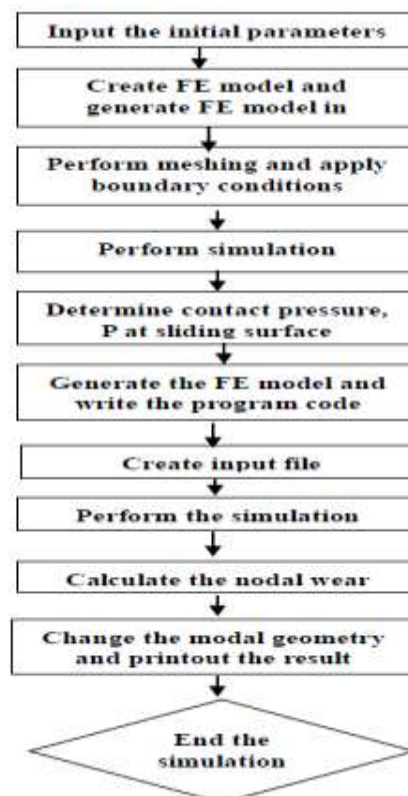

Disc brake assembly
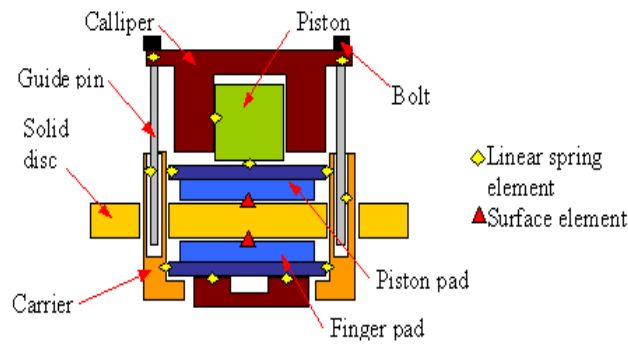
Contact interactions

Finite element model

7. SIMULATION APPROACH

Wear Simulation And Stability Analysis:

Archards wear formula is used in the present investigation. The modified equation is given below:

$$\Delta h = \dot{k}Pvt$$

Where, ω is the rotational speed (rad/sec), r is the pad mean radius (m). In the simulation the wear rate coefficient $k=1.78*10^{-13}m^3/Nm$ and r= 0.111m.

The wear tests are carried out under a brake line pressure of 1 MPa at a rotational speed of 6 rad/sec.at first these brke pads are run for 2 minutes, then for 4 minutes and then 6,8,10 minutes respectively. 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).

The seemingly short durations of wear test are due to numerical consideration. The longer the duration of wear the more the wear and the greater change of contact pressure. If t is too big there will be numerical difficulties in an ABAQUS run. Consequently a simulation of 10 minute wear means four ABAQUS run.
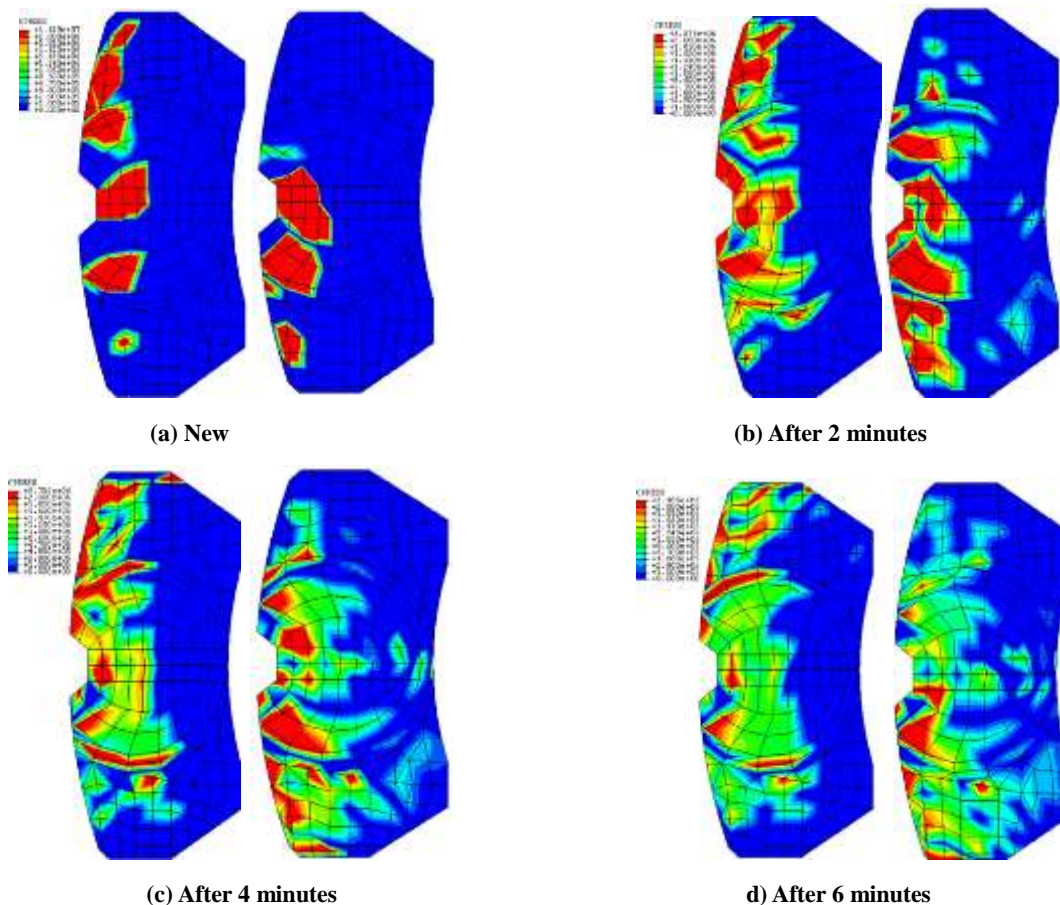
Baseline parameters are Brake line pressure is 1.0 MPa, sliding time is 30 seconds, rotational speed is 6 rad/seconds, and simulation duration is for 600 seconds.
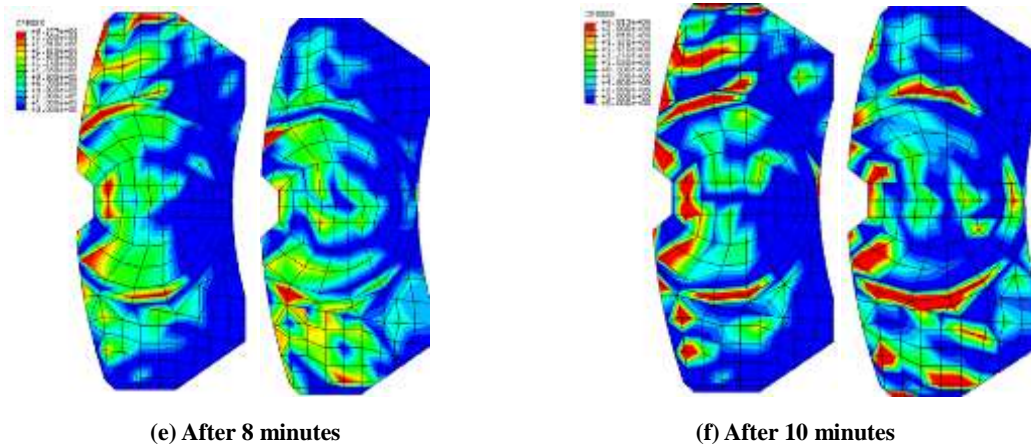
8. CONCLUSION

Contact plateaus are one of the factors that cause contact pressure cannot distribute uniformly. Contact area increases with the period of wear. Wear of lining material takes place when a thin layer of surface is removed. Brake pad profile initially scatters throughout interface but distributed uniformly after 10 minutes. Leading edge experiences more wear than trailing edge. Leading edge and the outer region of the pad are prone to wear.

9. RESULTS {(CONTACT PRESSURE DISTRIBUTION) FINGER PAD AND PISTON PAD}

Predicted Contact Pressure Distribution:

(a) New

(b) After 2 minutes

(c) After 4 minutes

d) After 6 minutes

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(e) After 8 minutes	**(f) After 10 minutes**

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