

A Study of Effects of Brake Contact Interfaces on Brake Squeal

Yan Yue and Lixin Zhang

Robert Bosch LLC

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ABSTRACT

Brake squeal is caused by the friction-induced vibration at the rotor/pad interfaces (primary contact interfaces) in a disc brake system. While there have been numerous research work evaluating the influence of primary contact interfaces on brake squeal, few studies can be found on the effect of the secondary contact interfaces, i.e., outer pad/caliper fingers, inner pad/pistons and pad/abutment, which can also significantly affect brake squeal based on our various dynamometer and vehicle tests. It is therefore the objective of this paper to investigate both the primary and the secondary contact interfaces and their influence on brake squeal. Simplified analytical models are created to gain insight into the stability of the brake system under low and high brake pressure; non-linear FEA analysis is employed for parametric study and countermeasure development; dynamometer and vehicle tests are used for verification. Case studies on low frequency squeal under low pressure and high frequency squeal at high pressure are presented.

INTRODUCTION

Brake squeal has been the primary subject in automotive industry not only due to its considerable warranty claims, but also due to its immense complexity. Over many decades, extensive experimental and theoretical investigations have been conducted to understand the fundamental mechanisms of brake squeal. Based on these researches, the possible mechanisms of brake squeal generation are summarized in the review papers [1-7].

The commonly accepted theory about brake squeal is that it is the result of dynamic instability caused by friction-induced vibration at the rotor/pad interfaces. Considerable efforts have been directed to the subject of understanding and modeling of the rotor/pad interface. One of such efforts has been focused on investigating the influences of interface conditions on both friction coefficient and contact stiffness [8-10]. The interface conditions covered by these studies include the rotor/pad relative speed, contact area/pattern (including wear and thermal effects), as well as interface temperature and pressure distribution, etc. The results of these parametric studies have then been incorporated in the friction model and contact model. The combination of more realistic friction and contact models has led to more comprehensive CAE model of brake squeal simulation [11-12].

Many efforts have also been devoted to develop squeal reduction methods by improving the rotor/pad contact interface. The most common strategy of interface modification is pad chamfering and slotting to optimize the rotor/pad contact area, pattern and pressure distribution.

While so much attention has been paid to pad/rotor friction interface, only few literature have slightly touched the topics on the secondary contact interfaces. Kumamoto et al. investigated the effects of pad/caliper contact condition on 2.0 kHz squeal which occurred in low pressure operation [13]. They concluded that it was inevitable to rigidly and stably connect the pad with caliper to improve the squeal performance since the contact condition of the pad and the caliper contributes to the squeal performance. They also confirmed that the

stainless steel contact shim structure is very effective for stable and rigid connection of pad with caliper. According to the studies on the rotor/pad interface conducted by Samie & Sheridan [14] and Tirovic & Day [15], the relative sliding between pad and rotor leads to an interface pressure offset to the leading edge. Rumold and Swift [16] studied the effects of the finger positions and piston locations of a twin piston medium truck disc brake. They concluded that a shift of fingers or pistons to trailing side was beneficial since a more uniform interface pressure distribution was achieved. Fieldhouse [17-18] conducted the measurement and analysis of the rotor/pad interface dynamic center of pressure and its influence on brake noise. They suggested that a disc brake requires a low friction coefficient between pad abutment and caliper mounting bracket to promote stability.

It is the objective of this paper to further comprehensively study the contact conditions of both the primary and secondary contact interfaces and their influences on the brake squeal. Case studies on low frequency squeal under light pressure and high frequency squeal under high pressure are presented.

EFFECT OF SECONDARY CONTACT INTERFACES ON LOW FREQUENCY SQUEAL UNDER LOW PRESSURE

Low frequency brake squeal (<3 kHz) under very light brake pressure is very common and structural modification usually does not help much. For pads, both rigid body motion and the first bending mode could be involved, but rigid body motion is the dominate mode. Therefore, insulator damping at low frequency shall be very small and is not effective in noise reduction.

Kumamoto [13] investigate 2 kHz and 2.5 kHz brake squeals at very low brake pressure. The interesting observation is that when the pads don't contact the anchor bracket at higher brake pressure (0.7 MPa), brake becomes noisier. We have very similar experiences for several brake platforms which lead us to believe that pad static instability could encourage low frequency noise generation at low brake pressure.

A pad is defined as static stable if there is always a continuous contact (support) between pad and anchor bracket at low pressure, that is, there is no sudden loss or change of contact patterns. Based on this definition, the following requirement should be met for a static stable pad,

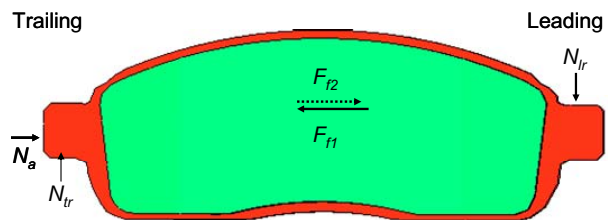


Figure 1. In-plane forces of inner Pad

$$\begin{aligned} N_a &> 0 \\ N_{tr} &> 0 \\ N_{lr} &> 0 \end{aligned} \quad (1)$$

Where, F_{f1} is the friction force between rotor and lining; F_{f2} is the friction force between insulator and piston for inner pad or caliper finger for outer pad; $N_a = F_{f1} - F_{f2}$ is the trailing abutment reaction force in tangential direction; N_{tr} is the trailing abutment reaction force in radial direction; N_{lr} is the leading abutment reaction force in radial direction. It should be noted that for backward drive or contact pressure center of lining is lower than abutment, the contact location of pad ear will be different.

To ensure the pad is always in contact with abutment in the tangential direction, F_{f2} needs to be less than F_{f1} . An effective way to achieve it is to reduce the friction coefficient of the secondary contact interfaces. Reducing the tangential clearance between the pad ear and abutment can also reduce load transferred through caliper pins which can shorten transition states and thus improve the stability of pads.

Pad rotational stability in the radial direction is very important to low frequency squeal under low brake pressure. A continuous contact in radial direction at both leading and trailing ends provides constraint to pad position. Pad geometry needs to be designed to avoid situations in which the moment of friction forces F_{f1} and F_{f2} with respect to trailing abutment is close to zero. Therefore, a low abutment design generally is good for pad static stability. Also, reducing abutment radial clearance should help stabilize the pad quicker.

CASE STUDY 1: A BRAKE WITH 2 kHz AND 2.6 kHz SQUEAL AT LOW PRESSURE

A brake with 2 kHz and 2.6 kHz at very low brake pressure (< 8 bar) is studied. Figure 2 shows actual road recording of the vehicle. The Brake Insulator was damaged even after one chassis test, which indicated the presence of a large and uneven contact force between the pistons and IB pad during the test. Due to insulator damage, the friction coefficient between piston/inner pad could be higher than that of rotor/lining and thus the pad ear doesn't contact the abutment at low brake pressure. The brake system becomes statically unstable according to equation (1) and low frequency brake squeal occurs.

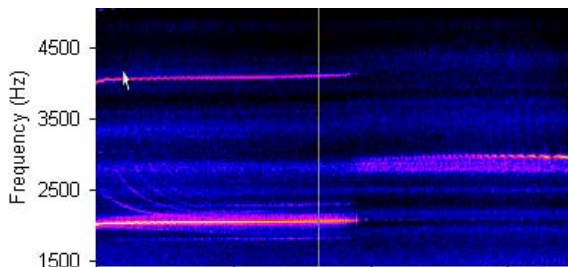


Figure 2. Road recording of the vehicle

A FEA model with good correlation with noise test results is used to investigate the effect of secondary contact interfaces on brake squeal. In nonlinear contact analysis, the friction coefficient between pad and rotor is 0.5; the friction coefficient between piston/inner pad and caliper finger/outer pad varies from 0.25 to 0.0. Figure 3 shows the tangential friction forces of inner and outer pads and tangential forces transferred through abutments. It can be seen that by changing friction coefficient between inner pad/piston from 0.25 to zero, inner tangential abutment force increases a lot while outer tangential abutment force decreases, but not too much. Reducing friction coefficient between finger/outer pad will increase tangential force transferred through outer abutment. It will also significantly reduce the force transferred through the inner abutment.

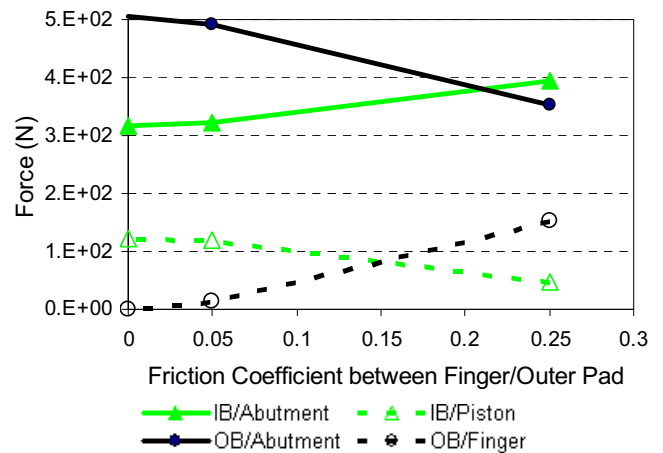
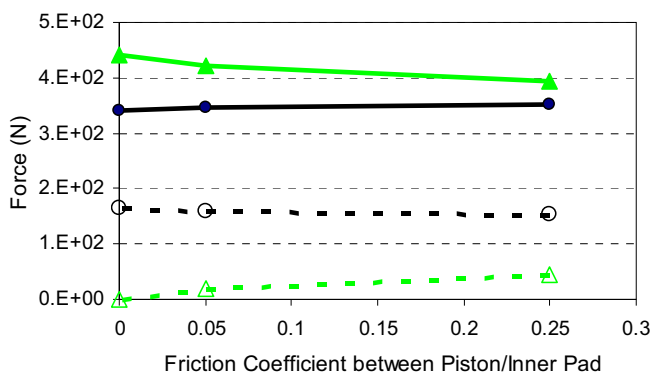


Figure 3. Pads and anchor tangential loading

Complex eigenvalue analysis is performed to evaluate the effect of friction coefficient of the secondary contact interfaces. It can be seen that there is no 2 kHz and 2.6 kHz if friction coefficient between piston and inner pad is zero. Reducing friction coefficient between caliper finger/outer pad only probably will not help 2 kHz and 2.6 kHz because it will increase friction force between piston/inner pad through higher anchor deformation. Based on this simulation, a floating shim is proposed as a countermeasure and it reduced 2 kHz and 2.6 kHz squeal propensity significantly in dyno test.

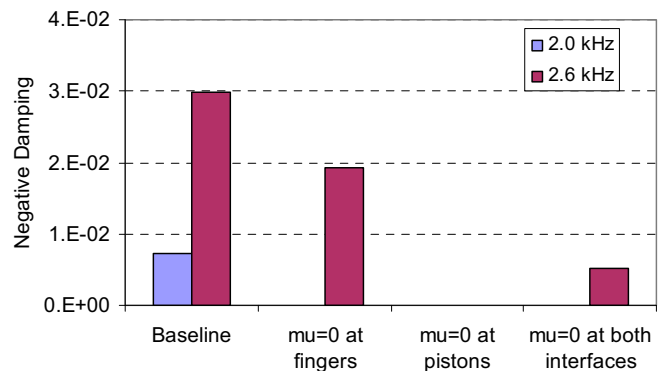


Figure 4. Effect of friction coefficients at secondary contact interfaces

Effect of abutment clearance on low frequency squeal at low pressure is shown in Figure 5. It is clear that tight abutment tangential clearance could improve 2 kHz squeal performance. However, abutment clearance has a limited influence on 2.6 kHz squeal. In Dyno validation test, thin shims are inserted into trailing slipper to reduce abutment tangential clearance and it shows a promising solution with almost no noise.