

Reduction of Brake Squeal Analyzed in Terms of Coupling Between In-Plane and Out-of-Plane Modes

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ABSTRACT

This paper examines the onset of brake noise and the mechanisms involved during the initial stages of noise occurrence. The emphasis is directed towards disc mode coupling and improvements are suggested to inhibit the coupling of the in-plane and out-of-plane modes with similar frequencies. To better understand the mechanisms, modal analysis of the system is carried out in a static condition but with varying pressure loads. In addition the operational deflection shape (ODS) was recorded under normal braking conditions using tri axial accelerometers attached on a disc. It was identified by a 3D scanning technique during squeal braking. Through a parametric study and using complex eigenvalue analysis the disc mode shapes were reproduced and correlated to the system results. The proposals to reduce mode coupling were successfully evaluated on a brake dynamometer and the results were confirmed with vehicle testing.

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INTRODUCTION

Brake squeal noise is one of the important NVH issues experienced during braking and has been investigated for many years by automotive engineers. The mechanism of squeal noise should be studied to satisfy the requirement of customers to drive a car comfortably and quietly. In general the brake squeal noise has complicated phenomenon due to self-exciting vibration in terms of resonance or modal coupling and stick-slip due to friction materials and sprag-slip in terms of pad dynamic pressure behavior by John Fieldhouse [1]. Many researchers [2,3,4,5] have applied mathematical methods, for example CEA (Complex Eigenvalue Analysis), and experimental approaches to find the mechanisms of disc brake squeal.

The stick-slip behavior is depending on the characteristic of friction material, due to the variation of friction coefficient which is depending on the variation of contact area, or the difference between static and dynamic friction coefficients as relative speed changes. The sprag-slip theory has been studied by John Fieldhouse, the important factor is the centroid of pad pressure. Leading offset should be smaller to reduce the squeal noise according to sprag-slip theory. However, it seems that only one single theory cannot explain every brake squeal phenomenon, suggested combining negative damping theory with structural coupling mechanism by which the squeal may be better explained.

A brake system has a self-excited and nonlinear dynamic system that is caused by friction induced dynamic instability which have negative damping, stick-slip, sprag-slip, structural/modal coupling, parametric excitation [1, 6] or internal resonance, etc. Many researchers[2,7] proved, using experimental approaches, that high squeal noise has a close relation of IPC (In-Plane Circumferential) frequency and had been studied in the mode coupling between IPC and OPD (Out-of Plane Diametric) mode. Some researchers [8] have been investigated higher squeal noise due to modal coupling caused by IPC mode and OPD of disc by using 3D laser scanning. However, it is very difficult to fully understand the squeal mechanism of mode coupling only by one or two approach methods because there are probably principle limitations on each analysis tool to clarify the transient squeal phenomena.

This study introduces four kinds of approach tools to investigate the modal coupling while supplementing their shortcomings and focused on self-excited vibration in terms of coupling between in-plane and out-of plane, especially IPR (In-Plane Radial), OPD and IPR self-coupling. In order to understand clearly low frequency squeal mechanism due to mode coupling, the combination approach is applied to have good correlation and be complementary. Finally a disc is suggested and validated to reduce 3.0 kHz squeal propensity by the dynamometer evaluation.

1. Time-Frequency Analysis

This paper considers 3.0 kHz squeal noise which is generated on a type of floating caliper at 15bar operating pressure in reverse direction. [Figure 1](#) shows the 1/4 vehicle dynamometer test rig used to generate the squeal noise and [Figure 2](#) shows the relevant 3.0 kHz frequency as recorded.



Figure 1. Vehicle setting on 1/4 car dynamometer

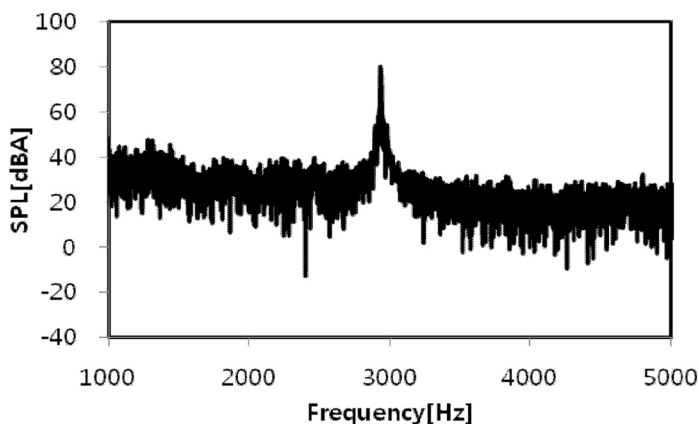


Figure 2. 3.0 kHz Squeal Noise

The evolution of squeal noise should be considered to understand mechanisms of the disc brake squeal because it is complicated and occurs in a short duration. FFT cannot analyze complex data in the transient period of the squeal. However Hilbert-Huang Transformation method (HHT) has been developed in recent years to analyze non-stationary signals by Huang et al [8]. The EMD (Empirical Mode Decomposition) method [2, 8] sifts the dominant squeal components from the original data so that a spectrogram analysis can be used to provide such detailed information as the time history. A series of disc brake squeal data is processed by the EMD method. An IMF (Intrinsic Mode Function), as defined in Huang et al, is a function that satisfies two conditions: (1) in the segment of data being

analyzed, the number of extreme values and zero crossings must either equal or differ at most by one; and (2) at any point, the mean value of the envelope defined by the local maxima and the envelope defined by the local minima is zero. A spectrogram method is then applied to analyze the evolution of frequency components at the first IMF in the squeal process, and probably to reveal and verify the coupling mechanism of in plane and out-of-plane vibration modes from the brake rotor during squeal. The EMD method is a local and adaptive method in frequency-time analysis. It decomposes the original signal $f(t)$ into a number of IMFs:

$$f(t) = \sum_{j=1}^n C_j(t) + r_n(t) \quad (1)$$

Where an individual $C_j(t)$ represents an IMF and $r_n(t)$ is the residue after the n IMFs have been extracted. In general, the first IMF, C_1 , contains the finest scale or the shortest period component of the signal and $r_n(t)$ can either be the mean trend or a constant. [Figure 6](#) shows that a tri axial accelerometer is attached near the outer edge on inner cheek of disc. As the accelerometers are available up to disc temperatures of approximately 70°C, the measurement was carried out under 10bar @ 5KPH (40rpm) reverse braking condition for 30 seconds. Two directional vibration signals were acquired with a tri-axial accelerometer; radial direction and axial direction. These data were analyzed by using HHT method. There are three stages until drivers can hear the brake squeal noise as follows (see [Figure 3, 4](#));

1st Stage: initial state of squeal evolution without squeal occurrence.

2nd Stage: transient state of squeal evolution without squeal perception

3rd Stage: audible squeal occurrence

In 1st stage, two kinds of frequencies have good decoupling and are tangled so that it is unlikely for there to be modal coupling. There is no increase of vibration amplitude. In 2nd stage, two kinds of frequencies are close and approaching to one frequency but still are separated. Even if drivers cannot hear the noise, the vibration amplitude is going up. In 3rd Stage, two kinds of frequencies seem to be one frequency. Definitely the noise is audible. Besides, there are 90 degree phase between axial vibration and radial vibration in time range from 0.099 sec to 0.1 (in the 3rd stage, fully development of squeal evolution). It means that there are two kinds of modes to be expected to have in-plane (radial vibration) and out-of plane (axial vibration).

Even if HHT provides a useful tool for definite identification of modal coupling between two modes, it is not enough to understand the modal coupling between in-plane and out-of plane because the mode shapes cannot be defined to separate easily. Further research is required to find out the mode shape in relation to in-plane and out-of plane in details by using assembly modal test

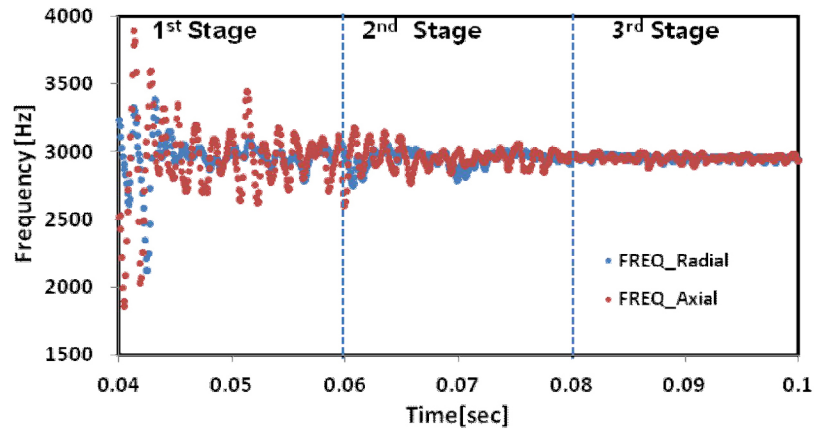


Figure 3. Instantaneous Frequencies of 1st IMFs

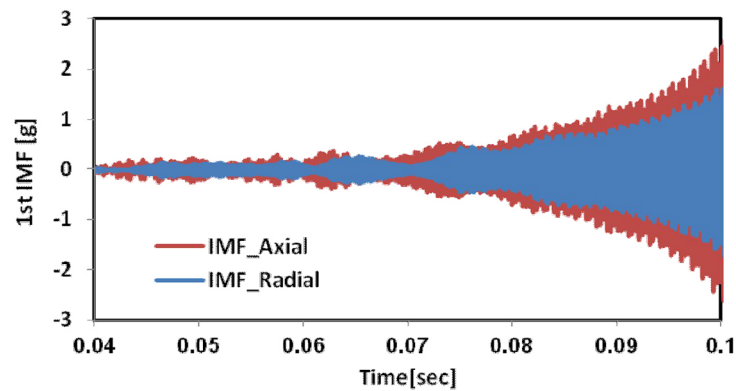


Figure 4. 1st IMFs of disc radial and axial vibration

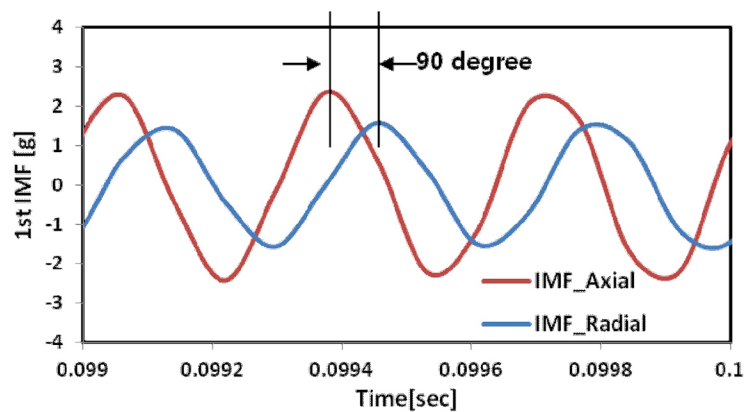


Figure 5. Zoomed vibration signals at 3rd stage

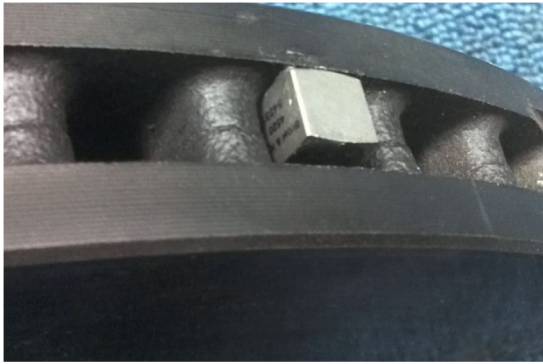
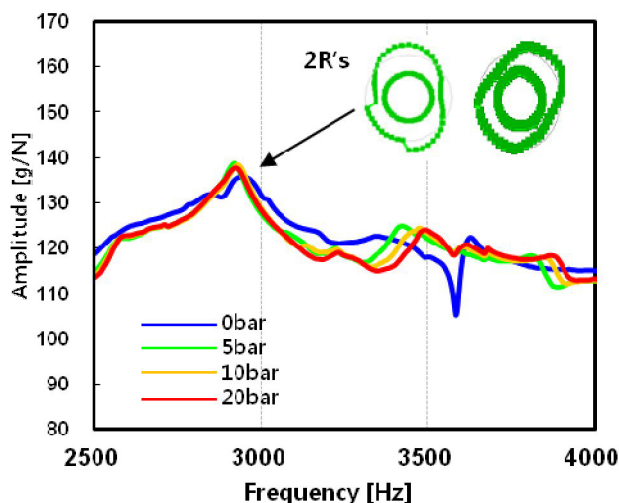


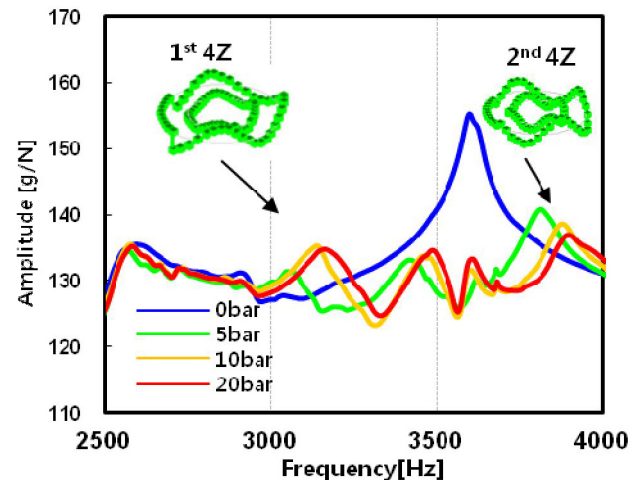
Figure 6. Installation of tri axial accelerometer on inner cheek of disc

2. Static Assembly Modal Test

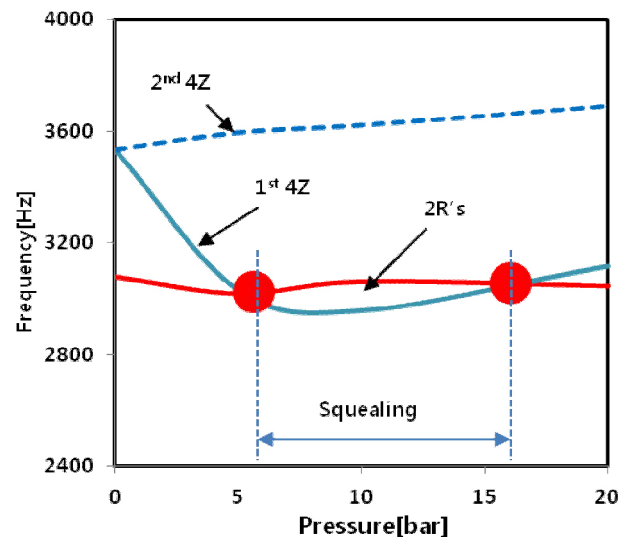
There is poor correlation between components modal test and assembly modal test due to assembly conditions and pressure conditions. Assembly modal test is carried out to get modal parameters considering boundary conditions and load conditions to simulate real vehicle condition. In case of both 2R (2nd IPR) modes, two modes can be separated by braking pressure based on 2 DOF vibration system of dynamic damper. 1st 2R mode is governed by caliper weight so that the frequency is lower than frequency of 0bar. On the contrary, 2nd 2R will be affected by pad and housing stiffness so that the frequency is higher than that of 0bar. In case of disc, the frequency of OPD mode is independent of assembly conditions, but 2R's (1st IPR and 2nd IPR) frequency is not. If the brake pressure is applied, frequencies of 2R are not changed but OPD frequency is going up (See Figure 7). As the frequency difference of 2R is very small, it may be assumed that the frequency behaviors of 1st 2R and 2nd 2R were expressed as one line.



(a). 2R Mode Shapes and FRF Results



(b). 4Z Mode Shapes and FRF Results



(c). Frequency variation according to pressure

Figure 7. Frequency behavior according to static pressure



Figure 8. Assembly modal test equipment

In case of 4Z modes (4th OPD modes) in common with 2R modes, two modes can be separated by braking pressure. According to pressure, 2R's and 1st 4Z are approaching and finally these frequencies meet at 5bar. Considering the pressure range of squealing is from 5bar to 15bar during

dynamometer test, 2R's and 1st 4Z are quite close under such pressure. It can be expected that 3.0 kHz noise can occur due to modal coupling between 2R and 4Z. The above assembly modal test is not under dynamic conditions. There is still little confidence what happens when the disc rotates. It is needed to investigate the mechanism of squeal occurrence by using ODS analysis during dynamometer braking.

3. System Vibration Analysis with tri-axial accelerometers during rotating disc without squealing

In dynamometer evaluation, the 3.0 kHz squeal occurs under reverse braking condition of 10bar, 5KPH (40rpm) 70°C. To get 3D ODS, 8 tri-axial accelerometers are installed at the inner face of disc (see Figure 9). As the speed of wheel during squealing is very low, data acquisition system (LMS Mobile 8ch) can be connected to accelerometers by each direct cable without slip-ring. Another accelerometer was attached on the caliper for the reference vibration signal.

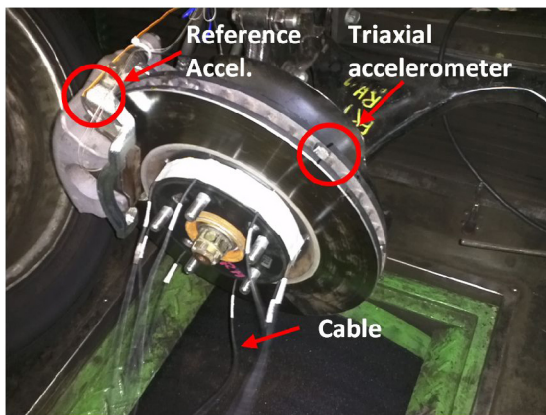
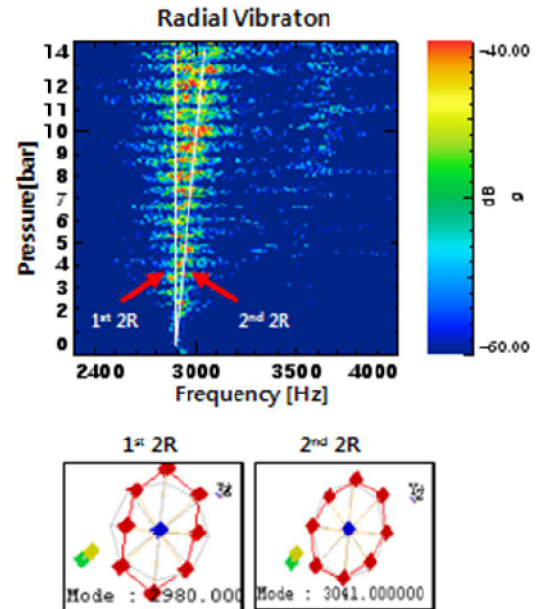


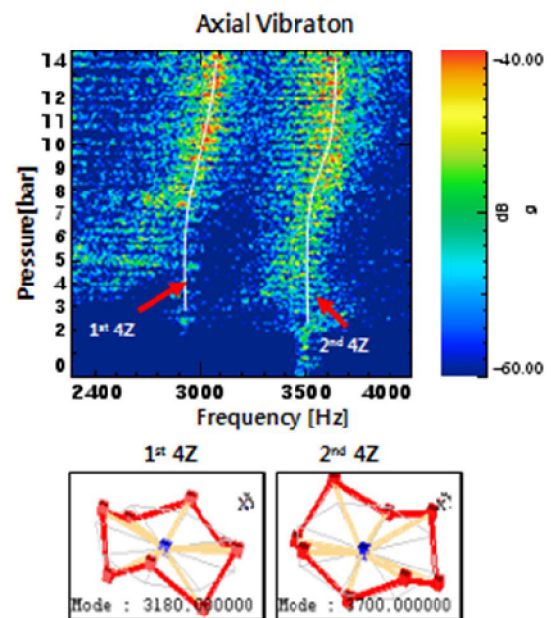
Figure 9. Installation of 8 tri axial accelerometers on inner cheek of disc

Figure 10 shows disc drag vibration behavior according to pressure 0~15bar before squealing. The behavior is similar to the results of assembly modal test. The frequency of 1st 2R is not changed but frequencies of 2nd 2R and 4Z are slightly increasing. The phase of 2R's has 22.5degree and the frequency difference, 60Hz, is very small so that it is possible to close together and the system will easily become unstable. Also frequencies of 1st 4Z and 2nd 2R are very close. In general 2R mode has axial vibration amplitude because of hat geometry not pure in-plane radial mode. In general squeal propensity mechanism due to modal coupling, vibration coupling and force coupling are composed of three factors, which are frequency, phase and amplitude (or damping). When the frequency difference is small and the damping coefficient is low, if such phases are different, the squeal propensity would be reduced. The phase is also important factor to be studied. Actually the phase can be changed by variations for the pressure or friction coefficient. Further

study will be continued by CEA (Complex Eigen value Analysis).



(a). Colormap results of 2R's ODS



(b). Colormap results of 4Z's ODS

Figure 10. Disc vibration and ODS under braking pressure (0~15bar)

To clarify the squeal mechanism, it is needed that the results of ODS under drag braking condition is reflected on the results of HHT and FFT (see Figure 11). There is well decoupling between 2R's and 4Z's in 1st stage. In 2nd stage the frequencies of 2nd 2R and 1st 4Z are coming up. Finally 3rd stage shows the same frequency to be squealing.