A EXPERIMENTAL STUDY OF DISC IN-PLANE MODE INDUCED BRAKE SQUEAL,

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ABSTRACT

It is one of the most important issues to reduce brake squeal in brake development. Brake squeal is classified in two kinds of vibration mode of the disc. One is out-of-plane mode which vibrates in out-of plane direction of disc. The other is in-plane mode which vibrates in circumferential or radial direction of the disc. For out-of-plane mode, a number of existing countermeasures can be potentially applied after characterization of the squeal occurrence condition by direct experiment or simulation analysis. On the other hand, in-plane mode, is occurred under high frequency, often involves the tangential (also called circumferential or longitudinal) vibration modes of brake rotor. Therefore, the characterization of the squeal occurrence condition has not been completely understood and still remains as one of important issues in the automotive industry due to it is much complicated mechanisms. In this paper, we show the experimental analysis results of the vibration mode of parts in a brake system while squeal is occurring when the disc vibrates in in-plane mode. And we consider a generating mechanism of brake squeal and countermeasure logic of reduction for disc in-plane mode brake squeal.

INTRODUCTION

Reducing brake noise to an acceptable level for the end customers is one of the main tasks of brake development today. As regards brake squeal, many research institutes have been researched and presented reducing brake squeal method[1,2,3,4,5,6,7,8] such as experimental modal analysis of brake system measured by laser Doppler vibrometer, oscillation mechanistic analysis using simplified friction vibration model, or simulation using Finite Element Methods.

Brake squeal phenomena can be generally separated into 2 main mode types related to the direction of disc vibration involved: out-of-plane and in-plane mode. For out-of-plane mode, a number of countermeasures can be potentially applied after characterization of the squeal occurrence condition by direct experiment or simulation analysis as above. However, as there are many possible mechanisms and root causes for the in-plane mode[9,10,11,12,13,14,15,16], it is generally necessary to perform a detailed analysis of the vibration mechanism before implementing a countermeasure.

In this paper, at first, we show the experimental analysis results of the vibration mode of parts in a brake system while squeal is occurring when the disc vibrates in 1st in-plane mode and consider a generating mechanism of the disc in-plane vibration. Secondly, we compose the simplified theoretical model and ascertain a new parameter to consider a disc in-plane vibration mode. Finally, we clarify a new parameter need by experimental investigation which changes a pad shape configuration.
EXPERIMENTAL ANALYSIS OF IN-PLANE MODE BRAKE SQUEAL

1. Test bench

The experimental apparatus is shown in figure 1. The brake system which is composed of disc rotor and calliper mounted through a drive shaft and a suspension knuckle attached to the experimental apparatus. The disc is driven through a drive shaft which is connected to an electrical motor. This motor could provide a variable speed between 0-50r/min. Brake squeal could reproduce by controlled brake pressure and rotation speed. The disc vibration frequency is measured by a laser vibrometer which is pointed to the disc indirectly surface.

![Figure 1: Test bench](image)

2. Disc brake specification

The brake system considered in this study is a front 16 inch brake. The disc diameter is 296mm and is a ventilated disc type. The caliper is a single piston floating type. The disc natural frequency which measured by hammering test was shown in Figure 2. The disc 1\textsuperscript{st} in-plane mode frequency is 6700Hz, therefore reproduced 6700Hz squeal in this brake system could be adopted to investigate 1\textsuperscript{st} in-plane mode phenomena.

![Figure 2: The disc circumferential eigenfrequency data by hammering](image)
The dynamometer equipped with this front brake system assembly test result is shown in figure 3. The squeal noise occurrences 6700Hz were recorded from brake pressure range between 0.5 to 1.0MPa. From the result, this brake system was adopted to investigate in-plane mode phenomena.

3. Measurement method of in-plane mode

3.1. 3D laser vibrometer

A 3D laser vibrometer is shown in figure 4. 3D laser scanning test needs to be implemented to acquire a disc and a caliper 6700Hz vibration mode clearly. Measuring points of a disc and caliper were shown in figure 5. During measurement, each point was scanning by three laser vibrometers at same time when 6700Hz brake squeal occurred. A disc and caliper and outer pad which could be irradiated a laser were measured by this method. Inner pad which was difficult to irradiate with a laser was measured by tri-axial accelerometer instead of 3D laser scanning vibrometer. Measuring method of tri-axial accelerometer is described in next section.

3.2. Tri-axial accelerometer

The test equipment is shown in figure 6. Tri-axial accelerometer needs to be implemented to acquire pads 6700Hz vibration mode clearly. In order to measure high frequency such as 6700Hz pad vibration mode, pads would have complex deflection. The procedure for measurement pad vibration mode, 6 tri-accelerometers are installed on each inner and outer pad backplate (outer side: leading corner, centre, trailing corner, inner side: leading, centre, trailing), as shown in figure 7. The purpose of measuring outer pad is for correlation between inner one, and also a disc and caliper which were measured by 3D laser scanning vibrometer. Therefore same reason for measuring the caliper, 8 tri-axial accelerometers are installed on each corner, as shown in figure 7. Pads and caliper vibration modes were represented by wireframe model which are drawn by points and lines. Acceleration data measured by tri-accelerometer installed on pads and caliper were input into each point.
Figure 4: 3D scanning vibrometer

Figure 5: Measuring points (left: z-axis direction, right: x-axis direction)

Figure 6: Tri-accelerometer measuring apparatus

Figure 7: Measuring points
4. Measurement result

4.1. The disc

The disc vibration mode during brake squeal generated at 6700Hz is shown in figure 8. As figure 8, a displayed number (1) is set at a disc maximum deformation in positive y-axis direction, (2) is 0.000037sec later from (1), (3) is 0.000074sec later from (1), and (4) is 0.000112sec later from (1). As the result of measurement in the case where a pad centre is 0 degree, “node” which has maximum circumferential amplitude would be located at 0 degree and 180 degrees, “Anti-node” which has zero or minimum circumferential amplitude would be located at 90 degrees and 270 degrees. A phase difference of disc slide surface circumferential amplitude between 0 degree and 180 degree is inverted. And when “anti-node” located at 90 degrees has contraction mode, in oppositional 270 degrees has extension mode. Then one pair of contraction and extension wave could be confirmed, the disc vibration mode during squeal generation at 6700Hz is 1st in-plane.

4.2. The caliper

A caliper vibration mode during brake squeal generated at 6700Hz is shown in figure 9. A displayed number (1) is set at a disc maximum deformation in positive y-axis direction as shown in figure 8, (2) is 0.000037sec later from (1), (3) is 0.000074sec later from (1), and (4) is 0.000112sec later from (1) Measuring a disc and a caliper is implemented at same time. And the range of amplitude is same as disc. As the result of measurement, mounting abutment deflection has y-direction, and cylinder fingers have z-direction. However both values were 0.2 times less than disc amplitude, a caliper was less influenced to in-plane mode brake squeal.

4.3. Brake pads

Pads vibration mode during brake squeal generated at 6700Hz is shown in figure 10, and both inner and outer pads centre of z-axis direction transient acceleration data is shown in figure 11. As shown in figure 10, a displayed number (1) is set at a disc maximum deformation in positive y-axis direction as shown in figure 8, (2) is 0.000037sec later from (1), (3) is 0.000074sec later from (1), and (4) is 0.000112sec later from (1). From the result, both pads deformation was composed z-axis direction, and “anti-node” of the pad is located on pad centre. As shown in figure 11, the outer and inner pads were vibrating approximately inverted direction. Both pads vibrating frequency were 6700Hz.
Figure 8: 6700Hz vibration mode (Disc)

Figure 9: 6700Hz vibration mode (Caliper)

Fig.10 6700Hz vibration mode (the caliper, outer and inner pads)
CONSIDERATION OF GENERATION MECHANISM OF IN-PLANE MODE

Brake system vibration mode during brake squeal generated at 6700Hz was composed of mainly a disc 1st in-plane mode and pads 1st bending mode. Considering generation mechanism of 1st in-plane mode, the caliper vibration was neglected due to its deformation very small. Therefore, the disc and the outer and inner pads contact surface are focused and considered the force transmission relative between the disc and the outer and inner pads.

Based on (1) and (3) of figure 8 and 10, the disc of pad contact surface deformed circumferential direction, and pads had any deformation to z-axis direction. At this moment, inner and outer pad bending force is generated at disc and pad contact surface.

Based on (2) and (4) of figure 8 and 10, a disc deformed back to neutral position, and pads vibrated with z-axis direction bending deformation. At this moment, pressure distribution will be caused at a disc and pad contact surface. This pressure variation force was transformed to friction variation force then transmitted to the disc.

As shown in figure 12, continuing mechanical transmission of friction variation force and pressure variation force between the disc and the pad contact surface, self-excited oscillation was generated and this brake system 6700Hz brake squeal was oscillated 1st in-plane vibration mode.
Here, two noticeable factors could be found. The first is as shown in figure 11, the outer and inner pads vibrating phases were inverted during brake squeal generated at 6700Hz. In this situation, the outer and inner friction variation force acted to the disc circumferential direction at a time. The second is the outer and inner pads behaved 1st bending deformation. If the pad vibrated in rigid mode, its circumferential displacement could be assumed small. However, a pad during 1st bending deformation could not be assumed small so that this means the elastic restoring force might act to the disc surface as shown in figure 12. These two factors could be supposed to generate the disc in-plane mode easily. Thus, to clarify the relationship between the in-plane mode and the elastic restoring force is investigated by using theoretical model which is described in next section.

THEORETICAL MODEL

In here, the in-plane brake squeal theoretical model is discussed, which is based on the concept of Millner’s approach [1]. The brake squeal phenomenon is able to represent by the following linear vibration system equation:

$$\begin{bmatrix} M \\ C \\ K \end{bmatrix} \begin{bmatrix} \dot{U} \\ \ddot{U} \\ N \end{bmatrix} + \begin{bmatrix} M \end{bmatrix} \begin{bmatrix} \dot{U} \\ \ddot{U} \end{bmatrix} + \begin{bmatrix} C \end{bmatrix} \begin{bmatrix} \dot{U} \\ \ddot{U} \end{bmatrix} + \begin{bmatrix} \end{bmatrix} \begin{bmatrix} U \end{bmatrix} = \begin{bmatrix} f \end{bmatrix} \quad (1)$$

Where $[M]$, $[C]$, and $[K]$ are respectively the mass, damping and stiffness matrices of the brake system. $[U]$ is a displacement vector representing displacements of brake system parts and $f$ is an external exiting force vector representing the variation in the friction force between the disc and pads.

Here, damping matrix $[C]$ can be initially ignored for the brake squeal analysis, since values of $[C]$ are determined by each part’s material properties and their contact situation. Thus, the vibration equation can be rewritten as

$$\begin{bmatrix} M \end{bmatrix} \begin{bmatrix} \dot{U} \\ \ddot{U} \end{bmatrix} + \begin{bmatrix} K \end{bmatrix} \begin{bmatrix} \dot{U} \\ \ddot{U} \end{bmatrix} = \begin{bmatrix} f \end{bmatrix} \quad (2)$$

The friction force variation $f$ between the disc and pads is direct cause of vibration that initiates brake squeal occurrences, and can be described as

$$f = \Delta \mu N + \mu \Delta N \quad (3)$$

Where $\mu$ and $N$ are respectively the friction coefficient and contact force vector between the disc and pads. In here, friction coefficient is assumed constant and the force variation can be rewritten as

$$f \approx \mu \Delta N \quad (4)$$

Substituting equation (4) into (2) yields,

$$\begin{bmatrix} M \end{bmatrix} \begin{bmatrix} \dot{U} \\ \ddot{U} \end{bmatrix} + \begin{bmatrix} K \end{bmatrix} \begin{bmatrix} \dot{U} \\ \ddot{U} \end{bmatrix} = \mu [\Delta N] \quad (5)$$

Since brake squeal is a type of self-excited vibration phenomenon, the amplitude of vibration in the system rapidly increases once the initial self-excitation has occurred. To express the amplitude of displacement in function of time $t$ such that
\[ U(t) = u e^{\lambda t} \quad (6) \]

Where \( u \) and \( \lambda \) are the amplitude of displacement and a complex eigenvalue of the vibration system. By substituting equation (6) into (5), a brake system equation is obtained to be able to investigate the self-excited vibration phenomenon of brake squeal.

Based on experimental result, an in-plane simplified theoretical model is represented as figure 13 and Table 1. The model is composed of a disc, an inner pad, a caliper and a pad support bracket. In here, an outer pad is not included due to vibration deformation as same as the inner pad. The pad is supported by a support bracket and represented the stiffness in x-axis direction \((K_t)\). The contact surface between the pad and the caliper is represented the stiffness in z-axis direction \((K_{cp})\). To represent pad bending mode, z-axis origin is set on the pad centre and the rotational spring \((K_p \theta)\) is set z-axis line symmetry. The contact surface between the disc and the pad is a set of spring and represented the stiffness in z-axis direction \((K_{pr})\) and x-axis direction \((K'_{pr})\). This \(K'_{pr}\) expresses that a pad elastic restoring force act to the disc.

![Figure 13: Simplified theoretical model](image)

| Xr: x-axis displacement of the disc |
| Xp: x-axis displacement of the pad |
| Xc: x-axis displacement of the caliper |
| Zr: z-axis displacement of the disc |
| Zp: z-axis displacement of the pad |
| Zc: z-axis displacement of the caliper |
| \( \theta_r \): angle of rotation of the disc |
| \( \theta_p \): angle of rotation of the pad |
| \( \theta_c \): angle of rotation of the caliper |
| a: contact position of the caliper and the pad |

Table 1: Nomenclature
The equations of motion of the disc, the pad and the caliper displacement and rotations respectively are summarized as follow:

**The disc**

\[
Mr \cdot \ddot{Z}_r + Krz \cdot Z_r = -k_{pr}(Z_r - Z_p) \quad (7)
\]

\[
Ir \cdot \ddot{\theta}_r + Kr\theta \cdot \dot{\theta}_r + Kp\theta(\theta_r - 20p) = \mu \cdot K_{pr}(Z_r - Z_p) \cdot d + K'_{pr}(X_r - X_p) \cdot d \quad (8)
\]

\[
Mr \cdot \dddot{X}_r + Krx \cdot X_r + K'_{pr}(X_r - X_p) = \mu \cdot K_{pr}(Z_r - Z_p) \quad (9)
\]

**The pad**

\[
Mp \cdot \dddot{Z}_p + Kpz \cdot Z_p = Kp\rho_0 \cdot (Z_r - Z_p) - K_{cp}(Z_p - Zc) \quad (10)
\]

\[
Ip \cdot \dddot{\theta}_p + Kp\theta \cdot 20p + Kr\theta \cdot (20\theta - \theta_r) + Kc\theta \cdot (20 - 2\theta_c)
= -\mu \cdot K_{pr}(Z_r - Z_p) \cdot c - K_{cp}(Z_p - Zc) \cdot a - \mu'K_{cp}(Z_p - Zc) \cdot c - K'_{pr}(X_r - X_p) \cdot c \quad (11)
\]

\[
Mp \cdot \dddot{X}_p + Kpx \cdot X_p + K'_{pr}(X_p - X_c) + Kt \cdot X_p = -\mu \cdot K_{pr}(Z_r - Z_p) + \mu'K_{cp}(Z_p - Zc) \quad (12)
\]

**The caliper / piston**

\[
Mc \cdot \dddot{Z}_c + Kcz \cdot Z_c = K_{cp}(Z_p - Zc) \quad (13)
\]

\[
Ic \cdot \dddot{\theta}_c + Kc\theta \cdot 2\theta_c + Kp\theta(\theta_c - 20p) = K_{cp}(Z_p - Zc) \cdot a - \mu'K_{cp}(Z_p - Zc) \cdot b \quad (14)
\]

\[
Mc \cdot \dddot{X}_c + Kcx \cdot X_c = \mu'K_{cp}(Z_p - Zc) \quad (15)
\]
Substituting equations (6), (7), (8), (9), (10), (11), (12), (13), (14) and (15) into equation (2), then multiplied $[M]^{-1}$ to equation (2) from left side, equation (16) can be given as follow:

$$\lambda^2 \cdot [U] + [A] \cdot [U] = 0 \quad (16)$$

Where matrix $U$ is

$$[U] = \begin{bmatrix} Zr & \theta_r \\ Xr & Zp \\ \theta_p & c^{\lambda t} \\ Xp & Zc \\ \theta_c \end{bmatrix}$$

Matrix $A$ is

$$[A] =$$

$$\begin{bmatrix} Kzz + Kpr & 0 & -Kpr \\ -\mu Kpr \cdot d & Kp0 + Kp\theta & -Kp0 \cdot d \\ 0 & \mu Kpr \cdot d & -2Kp\theta \\ 0 & 0 & Kpr \cdot d \\ 0 & 0 & 0 \end{bmatrix}$$

Equation (16) can be rewritten as follow,

$$\lambda^2 + A \cdot [U] = 0 \quad (17)$$

In here, $U \neq 0$, then the corresponding eigenvalue problem will be given by,

$$\det(\lambda^2 + A) = 0 \quad (18)$$

Each eigenvalue $\lambda$ is a complex number that contains two parts, one is real and the other is imaginary. Here, based on stability criterion, if the real part is positive, it means that the model is assumed instable and brake squeal will be occurred.
THEORITICAL INVESTIGATION

To validate the relationship between pad elastic restoring force caused from pad bending deformation and in-plane brake squeal, the investigation is implemented by MATLAB software based on the equation (18). The instability resolution of this model is given by changing variable elastic restoring value which is indirectly calculated by changing circumferential stiffness value ($K'_{pr} = \alpha \cdot K_{pr}$). In here, two circumferential stiffness values, which are 0 and 1 respectively, are used in instability resolution. The value $\alpha = 0$ is expressed pad rigid vibration mode, and the value $\alpha = 1$ is expressed elastic restoring force which caused from pad 1st bending mode. And irresponsible values are used in other parameters in order to just validate $K'_{pr}$ affected in-plane vibration instability criterion or not. Each value concept is as follow;

The results are shown in figure 14. In case of $\alpha = 0$, the instability area could not confirmed. But, in case of $\alpha = 1$, the instability area could be confirmed. From these results, it is assumed necessarily to consider an elastic restoring force in the theoretical model when studying the disc in-plane vibration.

![Figure 14: Simulation result](image-url)
EXPERIMENTAL INVESTIGATION

This experimental investigation is used actual brake system and test bench as shown in figure 1. Brake pressure is set on 0.8MPa and a disc rotation speed is 5r/min. In here, two kinds of prototype pads are produced to investigate the relationship between pad elastic restoring force and a disc in-plane vibration. Actually, it is difficult to coordinate only circumferential stiffness (K’pr) by only changing pad configuration so that this experimental investigation is implemented by coordinating pad deformation (Xp) as follow:

(a) Prototype 1: machined two 9mm depth slots on the pad (link to theoretical model $a = 1$)
(b) Prototype 2: machined parallel chamfer on the pad (link to theoretical model $a = 0$)

The pad configuration is shown in figure 15. The contact surface of prototype 2 is same as piston diameter. This configuration could be pressed the pad more tightly than without chamfer so that the pad vibration mode might be behaved as rigid. Each pad 6700Hz vibration mode is shown in figure 16. Pad bending mode could be confirmed both pad. But prototype 2 pad could be confirmed that bending mode deformation is smaller than prototype 1 so that prototype 2 is roughly represented rigid mode.

Figure 15: Pad configuration

Figure 16: Pad 6700Hz vibration mode
And actual brake squeal evaluation result of prototype 1 and prototype 2 are shown in figure 17. The prototype 1 is occurred 6700Hz brake squeal frequently. But prototype 2 is not occurred.

![Figure 17: Brake squeal evaluation of study 2](image)

As shown in figure 16 and 17, the relationship between pad elastic restoring force and the disc in-plane vibration mode could be confirmed by experimental investigation. And this result is same as theoretical one. Therefore, controlling pad elastic restoring could be one of effective item to countermeasure a brake squeal which oscillates at in-plane vibration mode.

**CONCLUSION**

The conclusions are summarized up 4 points as follow:

(1) The 16 inch ventilated disc and floating type caliper front brake system brake squeal occurring at 6700Hz is measured by 3D scanning vibrometer and tri-accelerometers. The disc has 1<sup>st</sup> in-plane vibration mode which vibrates in circumferential direction and is confirmed one pair of contraction and extension wave. The pads has 1<sup>st</sup> bending mode which has anti-node in pad centre. The calliper vibrates small.
(2) By continuing transmission of friction force and pressure variation force between disc circumferential deformation and pad bending, self-excited oscillation is generated and the brake system 6700Hz brake squeal was occurred 1st in-plane vibration mode.

(3) Simplified theoretical model can be composed and pad elastic restoring force is assumed to include in the model when studying a disc in-plane vibration.

(4) Controlling pad elastic restoring by changing pad configuration is the one of effective item to reduce brake squeal which oscillates in in-plane vibration mode.

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