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Effect of adding mass to rotor on in-plane squeal in automotive disc brake

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Abstract. This study experimentally examined disc brake-generated in-plane squeal by looking at vibration modes. The in-plane squeal was determined to be closely related to both the out-of-plane squeal that has directionality caused by Coulomb friction and the in-plane squeal caused by dry friction. The characteristics of in-plane squeal were also analytically investigated using a concentrated mass model formed by connected massless beams, and the relationship between mass added to the rotor and squeal suppression was clarified.

1 Introduction

Noise and vibration caused by vehicular traffic is an important social and environmental problem. Because design improvements have decreased engine noise, squeal generated by car brakes has begun receiving increased attention. Consequently, research in this field is increasingly important, and several countermeasures for brake squeal have been proposed [1, 2]. However, the mechanisms that generate vibrations in automotive disc brakes have yet to be fully clarified.

In this study, we investigated squeal caused by vibrations primarily in the in-plane direction of the disc, a phenomenon that has recently become a serious problem. Although this in-plane squeal is classified as high-frequency squeal, its characteristics are different from those of typical high-frequency out-of-plane squeals [3, 4].

2 Experiments

Figure 1 shows our bench test rig for a floating disc brake unit, which consists of a disc, caliper, pads, knuckle (which connects the brake unit to the car body), hydraulic pump, and geared motor. The height of the disc hat in the rig is greater than the height in a normal disc. We assume that the hat part of the disc contributes significantly to the coupling of in-plane and out-of-plane vibrations of the disc.

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Fig. 1. Disc brake bench test rig.

2.1 Braking tests and natural frequencies of disc and pad

Braking tests were conducted on the bench rig, and the squeal frequencies were investigated based on the acceleration of the caliper measured with an accelerometer. The generation frequency of the in-plane squeal was 10,375 Hz, which is classified as a high-frequency squeal, referred to as base squeal. The acceleration amplitude of the in-plane squeal was 7.47 m/s^2.

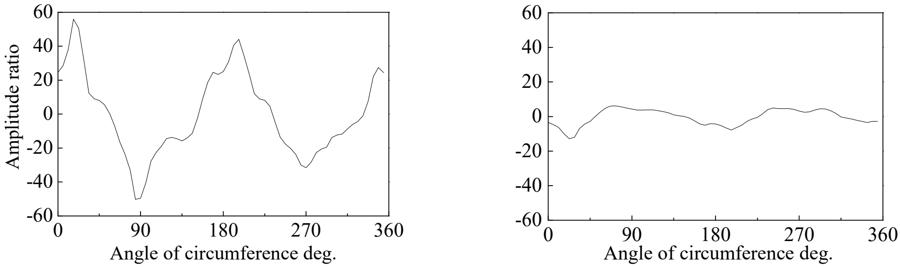
The natural frequencies of the disc in the out-of-plane and in-plane directions and that of the brake pad respectively were measured using accelerometers. Table 1 shows the natural frequencies of the disc in the out-of-plane and in-plane directions and that of the pad, respectively. The characteristic mode of the out-of-plane vibration of the disc is described as the (m, s) mode corresponding to the natural frequency of the disc, where m and s are the nodal diameter and nodal circle, respectively. As shown in Table 1, the second-mode natural frequency of the disc in the in-plane direction (10,400 Hz), and sixth mode of the pad (11,231 Hz) are close to the squeal frequency. This characterizes the high-frequency squeal, which is generated by the coupled vibration of the disc and the brake pads.

Table 1. Natural frequencies of disc and pad (Hz).

Disc				Pad	
In-plane direction		Out-of-plane direction			
1st mode	6,938	(8, 0)	10,550	5th mode	9,344
2nd mode	10,400	(9, 0)	12,300	6th mode	11,231

2.2 Vibration mode of disc during squeal

The vibration mode of the rotating disc in the out-of-plane and in-plane directions during squeal was investigated using an accelerometer applied with a slip ring. In this experiment, the out-of-plane vibration of the outer caliper was also investigated using accelerometers. Figure 2 shows the in-plane and out-of-plane vibration amplitude ratios of the disc in squeal to the vibration amplitude of the caliper, where the abscissa shows the circumferential angle of the disc. Figure 2 shows that the in-plane vibration amplitude of the disc was much larger than the out-of-plane amplitude, and the vibration amplitude of the caliper was much smaller than that of disc. This is the characteristics of high frequency squeal. Moreover, the disc vibrated elastically at the second mode in the in-plane direction during squeal, as shown in Fig. 2(a). This result coincides with the natural frequency of the in-plane vibration of the disc. Figure 2(b) shows that the two-diametrical-node disc vibration coupled by the second mode in the in-plane direction was confirmed. The height of the hat of the disc contributes significantly to the coupling of the in-plane and out-of-plane vibrations of the disc. The position of antinodes and nodes during squeal in Fig. 2 were fixed in space while the disc rotated.



(a) In-plane direction (b) Out-of-plane direction

Fig. 2. Vibration mode of disc in squeal.

3 Analytical calculation of elastic ring modeled as concentrated mass

Based on the experimental results described in Section 2, the in-plane squeal was ascertained to be closely related to both the directional out-of-plane squeal caused by Coulomb friction and the in-plane squeal caused by dry friction. In this section, we describe how a complex eigenvalue problem was analyzed to investigate the effect of the in-plane and out-of-plane vibration of the disc and pads and the rigidity of the disc hat on in-plane squeal.

3.1 Analytical model

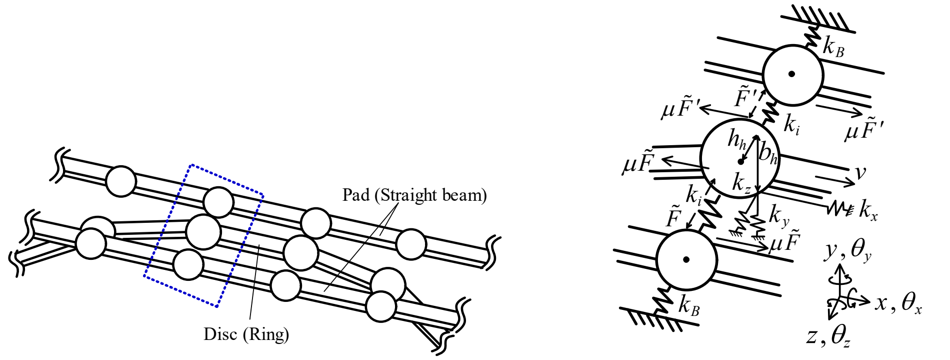
The analytical model used in the experiments (Fig. 3) was based on the model in Ref. [5]. This model did not faithfully reproduce the dimensions of an actual disc brake unit because the goal of the analysis is to qualitatively, rather than quantitatively, obtain characteristics of in-plane squeal and investigate the stability of the resulting self-excited vibration system.

The disc and the inner and outer pads were modeled as a system of massless elastic beams and concentrated masses with half of the mass of the beam and half of its moment of inertia concentrated at either end of the beam. The pad was formed as a straight beam divided into 14 elements. The disc was modeled as a ring divided into 80 elements, which were straight beams rather than curved beams, as shown in Fig. 3(a). The displacements in the tangential, circumferential, and out-of-plane directions of the ring are denoted as x, y, and z, respectively. Additionally, θ_x, θ_y, and θ_z are the rotation angles about the x-, y-, and z-axes, respectively. The corresponding displacements and rotation angles of the pads are denoted as x_p, y_p, z_p, θ_{xp}, θ_{yp}, and θ_{zp}, respectively.

As shown in Fig. 3(b), the ith spring k_i is in contact with the pad and ring in the z-direction. The inner and outer pads are supported on the caliper piston and holder, respectively, by spring k_B in the z-direction. As shown in Fig. 3(b), the rigidity of the hat section of the disc was taken into account by setting each node of the ring and the springs k_x, k_y, and k_z to be located from the center of gravity at distance b_h in the y-direction and distance h_h in the z-direction.

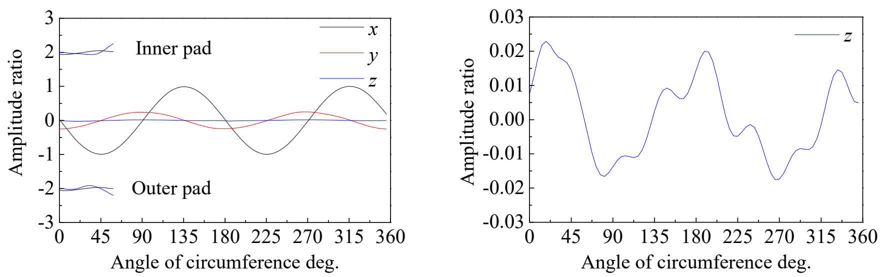
In this paper, primes [′] denote physical quantities or dimensions on the outer side of the rotor.The equations of motion for the ring and pads are all formally the same as those given in Ref. [5] and the numerical parameters used in the calculation were also same as those used in Ref. [5]. Detailed descriptions are not given here.

The natural frequencies of the ring and pads were first calculated without friction, a supporting spring ($k_B = k_1 = 0$), or a contact spring ($k_i = 0$). The natural frequency ratio between the second-mode frequency of disc in the in-plane vibration and sixth mode of the



(a) Structure of ring and pads (c) Relationship between ring and pad elements

Fig. 3. Analytical model.



(a) Ring and inner and outer pads (b) Enlarged view of ring mode in Fig. 4(a)

Fig. 4. Complex mode of unstable vibration.

pad were similar to that in an actual brake system.

The stability of the system was then analyzed by applying the characteristic value problem under the conditions of Coulomb and dry friction. The system contained only one unstable vibration. The positive real part of the eigenvalue was 88, and its frequency was 12,632 Hz. Figure 4(a) shows the vibration modes of the ring and the inner and outer pads, where the mode is based on the maximum amplitude and the horizontal axis indicates the circumferential angle.

In the unstable vibrations, the rings vibrated in the second in-plane mode with a large amplitude. Figure 4(b) shows an enlarged view of the ring mode in the *z*-direction from Fig. 4(a). Figure 4(b) shows that the ring vibrates in the out-of-plane two-diametrical node excited by the second mode in the in-plane direction. These results agree well with the results shown in Fig. 2. The pads vibrated in the bending mode, and the amplitude of the in-plane vibration of the ring was approximately 20 times larger than that of the out-of-plane vibration.

3.2 Effects of adding mass to rotor on in-plane squeal

The effects of added rotor mass on the in-plane squeal were investigated. Specifically, we analytically determined whether adding mass to the disc contributes to the reduction of the natural frequency of the disc and the suppression of the in-plane squeal. Figure 5 shows the position on the ring where 2 sets of 5 additional masses were added, for a total of 10 masses. As shown in Fig. 5, within each set, the 5 masses were attached to consecutive elements of the ring, and the circumferential angle between the 2 sets of 5 masses was determined by α. We analytically investigated how angle α affects in-plane squeal. We also calculated how adding masses affect in-plane squeal during one ring rotation by changing rotation angle β

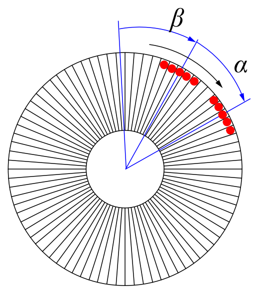


Fig. 5. Positioning of additional mass.

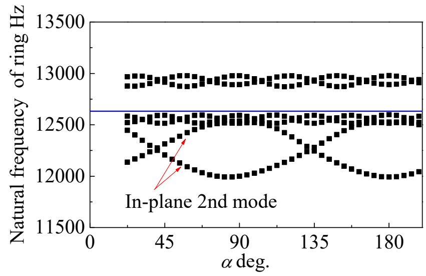


Fig. 6. Relationship between natural frequencies of ring and angle α.

after angle α is determined. The mass of one standard element of the ring was 39 g. The total mass of the 10 masses added to the ring was varied from 110 g to 210 g in increments of 50 g.

First, the relationship between the natural frequencies of only the ring with the 10 additional masses and angle α were investigated using the eigenvalue problem, as shown in Fig. 6. The total mass of the 10 masses added to the ring was 210 g. The abscissa is circumferential angle α between the 2 sets of 5 masses, and the ordinate is the natural frequencies of the ring. The blue line in Fig. 6 shows the base value of natural frequency for the in-plane second mode when no mass was added. Figure 6 shows that the natural frequency of the in-plane second mode when mass was added was lower than when no mass was added. The natural frequencies separated into sine and cosine natural modes of each other when the mass was attached. The sine and cosine natural modes of the in-plane second mode intersect each other when α is around 45° and 135°. The cosine natural modes of the in-plane second mode decreased the most when α was 90°.

Next, complex eigenvalue analysis was performed. Figure 7 shows the relationship between the real part of eigenvalues and angle α. The abscissa is α, and the ordinate is the maximum value of the real part of the eigenvalue for each value of α during one rotation of the ring as angle β was varied from 0° to 360°. The blue line in Fig. 7 shows the base value of the real part of eigenvalue 88 when no mass was added. From Fig.7, the real part of the eigenvalue decreased as more mass was added when α was around 45° and 135°. It finally became negative when the total mass was 210 g, which means the squeal did not occur during a complete disc rotation, even though the vibration became more unstable than for the base value when α was around 90°.

Figures 8(a) and 8(b) show the relationship between the real part of the eigenvalues and angle β when α was 45° and 90°, respectively, and the total added mass was 160 g. The red dots in Fig. 8 indicate regions where the real part of the eigenvalue is positive. The blue line in Fig. 8 shows the base value of the real part of eigenvalue 88 when no mass was added. Figure 8(a) shows that the real part of the eigenvalue was lower than the base value during the whole rotation when α was 45°. In Fig. 8(b), it was higher than the base value when angle β was around 0°, 90°, 180°, 270°, and 360° and α was 90°. Here, the complex squeal mode shown in Figs. 2 and 4 are fixed in space while the disc rotated; therefore, the effect of adding mass changed rotation angle β based on the relationship between the positions of the sets of five masses and the positions of antinodes and nodes in the complex mode of unstable vibration. In Fig. 8(a), one of the sets of five masses is always at an antinode position in the in-plane second mode. Therefore, it is effective through the whole rotation angle β when α is 45°. In contrast, both sets of five masses are at node positions in the in-plane second mode when β is 0°, 90°, 180°, 270°, and 360° in Fig. 8(b). Then, vibration became more unstable, even though suppression was otherwise very effective

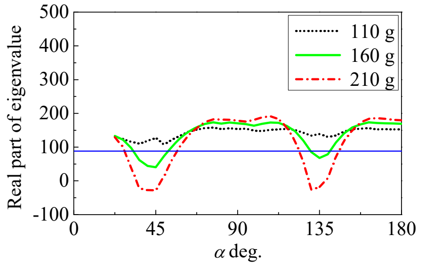


Fig. 7. Relationship between real part of eigenvalues and angle α.

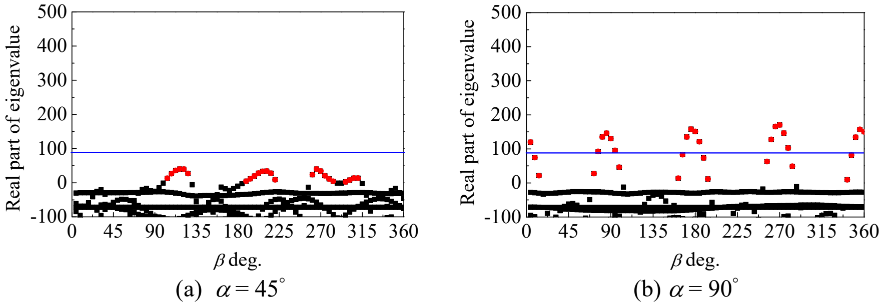


Fig. 8. Relationship between real part of eigenvalues and angle β (Additional mass is 160 g).

because both sets are at antinodes when β is 45°, 135°, 225°, and 315°.

4 Conclusions

In this study, the height of the hat of the disc contributed significantly to the coupling of the in-plane and out-of-plane vibrations of the disc. The out-of-plane pad vibration and the two-diametrical-node disc vibration coupled by the second mode in the in-plane direction were excited by Coulomb friction and could become more unstable when the in-plane vibration of the second mode was excited by dry friction. Thus, the resulting squeal was closely related to both the directional out-of-plane vibration caused by Coulomb friction and the in-plane vibration caused by dry friction.

The analytical results showed that vibration is suppressed by adding mass at the position of the antinode of the in-plane second-mode vibration of the disc. This reduced the natural frequencies of the disc, which disrupted the balance between the natural frequencies of the pads and disc and reduced the vibration coupling between the disc and brake pads.

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