

Study of Effects of Residual Stress on Natural Frequency of Motorcycle Brake Discs

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ABSTRACT

In brake squeal analyses using FE models, minimizing the discrepancies in vibration characteristics between the measurement and the simulation is a key issue for improving its reproducibility.

The discrepancies are generally adjusted by the shape parameters and/or material properties applied to the model. However, the discrepancy cannot be easily adjusted, especially, for the vibration characteristic of the disc model of a motorcycle.

One of the factors that give a large impact on this discrepancy is a thermal history of the disc. That thermal history includes the one experienced in manufacturing process.

In this paper, we examine the effects of residual stress on the natural frequency of motorcycle discs. The residual stress on the disc surface was measured by X-ray stress measurement method. It was followed by an eigenvalue analysis. In this analysis, we developed a unique method in which the residual stress was substituted by thermal stress. Using this method, the discrepancy between measurement and calculation of the natural frequency was reduced from $\pm 5.2\%$ to $\pm 1.3\%$. In further study on the relationship between stress distribution and natural frequency, we clarified that the gradient of residual stress in radial direction and the stress distribution in depth direction have a large dependency on natural frequency.

The developed method made it possible to minimize the frequency discrepancies between measurement and calculation that were unable to be compensated by the adjustments using parameters of part shapes and properties of materials. The method significantly contributed to the improvement of the accuracy of brake squeal analyses.

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INTRODUCTION

Brake squeal is a vibration phenomenon caused by the mode coupling of the brake system due to the friction force generated between the brake pads and the brake disc.

The frequency range of squeal extends across the entire audible zone (from several hundred Hz up to 10+ kHz) involving a number of vibration modes. The occurrence of the brake squeal depends on various factors such as the vehicle speed, pressure placed on the brake pad, temperature, humidity, etc. For these reasons, cause analysis or countermeasure planning is difficult, so brake squeal is frequently analyzed using a dynamic model.

The equation-of-motion model or the finite-element model is often applied to brake squeal analysis. Although the equation-of-motion model, in which a relatively small number of parts are regarded in a structure model consisting of the spring, mass

and damper system, allows analysis of the time-varying friction force, it does not focus on the contribution to the system of the vibration characteristics determined by the shape of a part. On the other hand, as an analysis by the finite- element model, which contains information on the shape of each part, allows reproduction of the complex vibration mode of the system in which the parts are assembled, it is suitable for brake part development. For instance, Nakata et al. applied a numerical analytical method [1] to a motorcycle brake squeal analysis, in which the system's instability is calculated with the real eigenvalues and eigenvectors acquired in advance using a finite element model. They proposed an efficient design modification method to reduce brake squeal [2]. For effective application of analysis by the finite element model to the part development, it is necessary to bring the vibration characteristics of all components as close as possible to the actual vibration characteristics by adjusting parameters such as the material properties, and the spring constant at each

part-to-part connection. Reducing the gap between the measurement and the calculation of the natural frequency in each unit part is therefore key. The brake disc is one of the main components of disc brake system and its vibration characteristic is particularly important.

As the brake discs for automobiles are commonly made of cast iron and have complex shapes such as with ventilated discs, research has been conducted on how the shape, material, and production method affect the vibration characteristics [3]. On the other hand, the discs for motorcycles are generally made of stainless steel sheets and predominantly have simple shapes created by punching, so the differences in shape parameters and/or material variations do not usually become a problem when modeling. Nevertheless, depending on the thermal history of the disc, including the manufacturing process, the vibration characteristics of the model cannot be matched to those of the actual product.

It can be considered that such a gap occurs due to the large residual stress generated by the induction hardening applied on the sliding surface to improve abrasion resistance. As an example of research on the relationship between residual stress and vibration characteristics, there is the study on circular saws, showing how the residual stress generated by rolling affects the natural frequency [4]. There has been research on residual stress in motorcycle brake discs, including the measurement of residual stress in the proximity of the tiny holes on the sliding surface [5], but there are no reports on measurement of the residual stress over the entire surface.

In this paper, attempts were made to clarify the relationship between the residual stress generated in the motorcycle brake disc and the natural frequency in order to improve reproducibility of vibration characteristics of the finite element model in the brake squeal analysis. To be specific, we investigated the distribution of residual stress on the surface of a brake disc using X-ray stress measurement, and conducted an eigenvalue analysis under conditions where the measured stress distribution was reproduced. Using the eigenvalue analysis under various stress distribution conditions, we examined the effect of the residual stress generated in a brake disc on its natural frequency. We also used an annealed disc to clarify how the residual stress on disc surface affects the natural frequency. Finally we applied the simulation method to development and verified its effectiveness.

NATURAL FREQUENCY GAP

Analysis Model and Disc Vibration Mode

The natural frequency of a motorcycle brake disc (hereafter, "disc") is calculated by the eigenvalue analysis of a solid mesh model using Young's modulus of typical steel after adjusting the shape and mass to actual measurements. It should be

noted that the vibration mode of a disc is generally divided into the out-of-plane vibration mode and the in-plane vibration mode. In this paper, among the out-of-plane vibration modes, the vibration mode with the number of nodal circles "m" and the number of nodal diameters "n" is referred to as Mode (m, n). Mode (m, 0) is called the nodal circle mode and Mode (0, n) is called the nodal diameter mode (Figure 1). The in-plane mode is not taken into account.

Definition of Natural Frequency Gap

The natural frequency calculated by the above-mentioned method sometimes has error of several % relative to the natural frequency measured by an impact test. Figure 2 shows the shape and excitation response point of the disc used in this study. A cylindrical coordinate system having the disc rotation axis as the "z" axis is used in the measurement and calculation. Figure 3 shows examples of the FRF (Frequency Response Function) obtained from the experiments and the calculations. The following features of the natural frequency are seen from Figure 3.

Feature 1. The orientation of deviation in the natural frequency of nodal circle modes from the measurement to the calculation is opposite that of the nodal diameter modes.

Feature 2. The gap (%) between the measurement and the calculation decreases along with an increase of the order of mode.

Especially when Feature 1 is seen, as the adjustment of mass or Young's modulus equally affects the natural frequency of all modes, the gap cannot be reduced. In this paper, we choose to call this gap the "natural frequency gap".

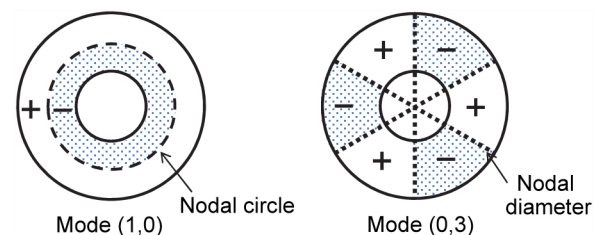


Figure 1. Nodal circles and nodal diameters

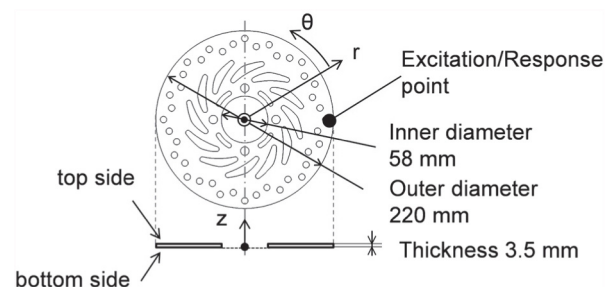


Figure 2. Configuration of test disc and FRF measurement point

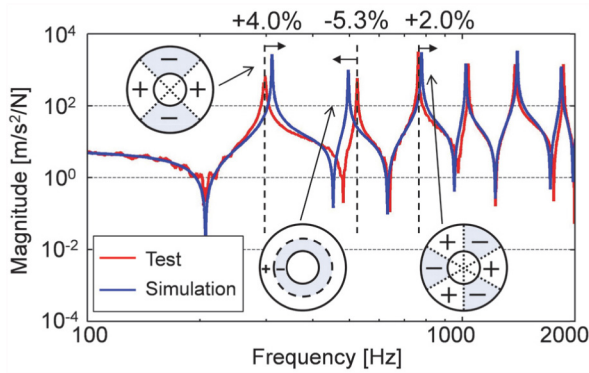


Figure 3. Natural frequency gap between measured and simulated vibration frequencies

MEASUREMENT OF RESIDUAL STRESS

It has already been reported that the change in the natural frequency of circular saws due to the residual stress generated by rolling has the same tendency as Feature 1 in the preceding section. Presuming that there is considerable residual stress on the disc surface, we measured the residual stress.

Test Disc

To confirm the effect of residual stress on natural frequency, we measured the residual stress of two kinds of disc having the same shape but different quenching zones. Figure 4 shows the discs used for the measurement. Here, the disc in which only the pad sliding surfaces are induction-hardened is referred to as Disc A, and the disc in which all surfaces are entirely hardened by heating in a furnace is referred to as Disc B. Note that in both samples, martensitic stainless steel is used as the disc material.

Figure 5 shows the FRF obtained from the measurements and calculations for Disc A and Disc B. While the maximum natural frequency gap was approximately 5% in Disc A, in which only the sliding surfaces were hardened, it was less than 1% in Disc B, in which all the surfaces were hardened.

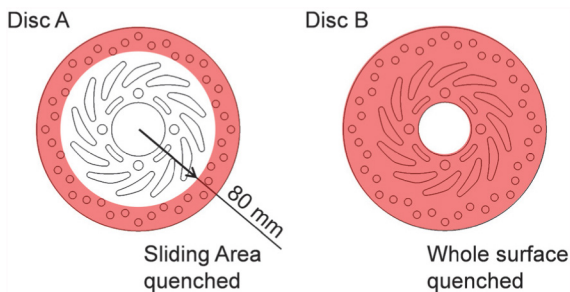
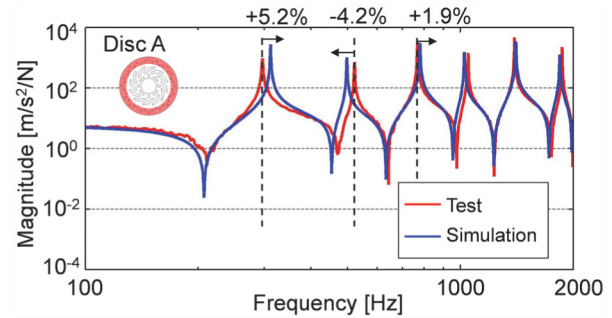
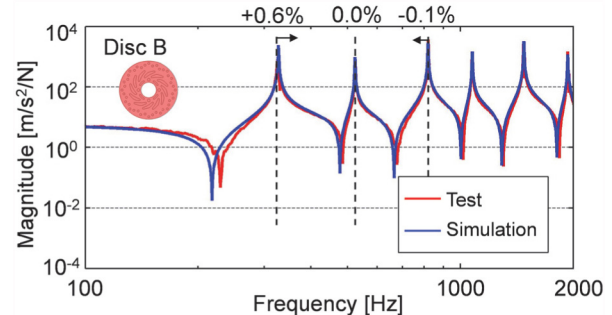


Figure 4. Comparison of quenched area between test discs



a. Measured and simulated FRF of Disc A



b. Measured and simulated FRF of Disc B

Figure 5. FRF of Disc A and Disc B

Measurement Method

The residual stress in the circumferential direction of the disc surface was measured using the micro-area X-ray stress measurement system from Rigaku Corporation ($\sin^2\psi$ method).

Figure 6 shows the measurement points. In the radial direction, the residual stress was measured at seven points, i.e., $r=31.5, 49, 74, 79, 89, 94,$ and 106.5mm , on both the top and bottom sides of the disc. For Disc A, the measurements were performed in five patterns at various center angles i.e., $\theta=-90^\circ, -45^\circ, -30^\circ, 0^\circ, +90^\circ$ on the top surface.

Measurement Results

Figure 7 shows the measured residual stress in the circumferential direction (σ_θ) of Disc A. In this disc, a compressive residual stress of several hundred MPa was seen over the whole surface both on the top and bottom sides. A larger compressive residual stress was seen at the outer circumference (induction hardened part) than at the inner circumference (unquenched part). There was a gradient of residual stress in the radial direction from the inner circumference to the outer circumference. Figure 8 shows the measured residual stress of Disc B. Although compressive residual stress was generally seen on the top and back surfaces of Disc B, just as in Disc A, the stress was virtually equal at the inner circumference and at the outer circumference (both are quenched) without a gradient of residual stress in the radial direction, as seen in Figure 7. Figure 9 shows the residual stress of Disc A. As no apparent difference exists in the

tendency of measurements at each direction, it can be considered that the residual stress is almost evenly distributed in the circumferential direction.

From the above measurements, such trends were seen that the natural frequency gap was large in Disc A in which there was a gradient of residual stress in the radial direction, while the natural frequency gap was small in Disc B in which the residual stress in the radial direction was virtually constant. It is therefore presumable that the gradient of residual stress in the radial direction from the inner circumference to the outer circumference affects the natural frequency of the disc.

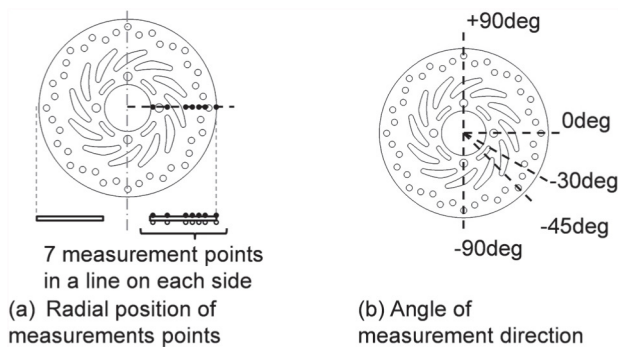


Figure 6. X-ray stress measurement points

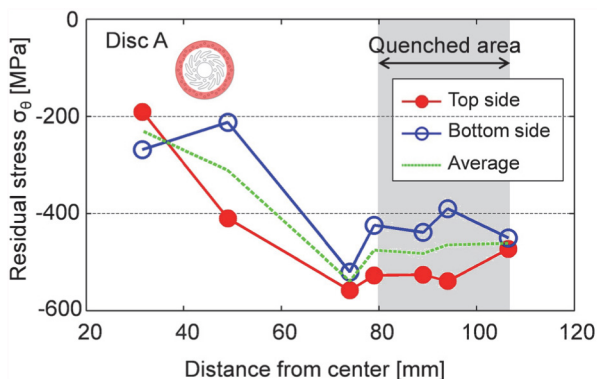


Figure 7. Residual stress distribution on the surface of Disc A

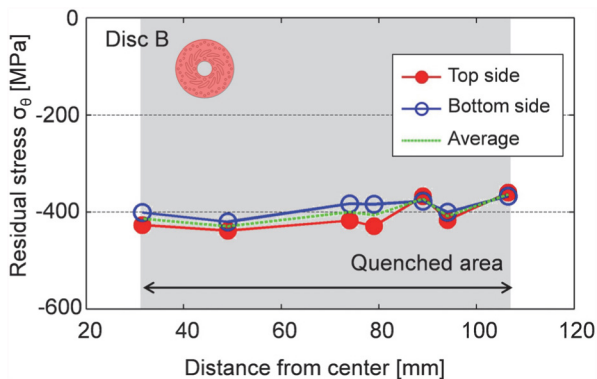


Figure 8. Residual stress distribution on the surface of Disc B

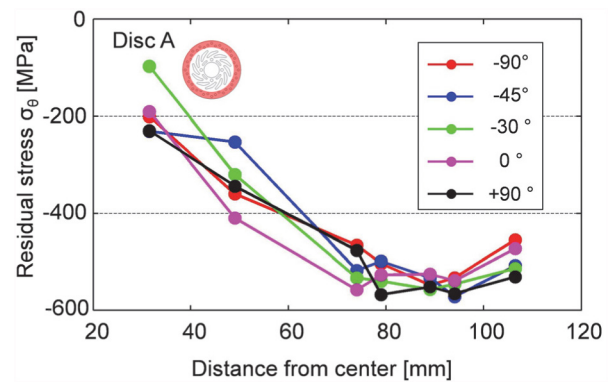


Figure 9. Multi-directional residual stress distribution on Disc A

SIMULATION CONSIDERING RESIDUAL STRESS

To clarify how the residual stress distribution mentioned in the preceding paragraph affects the natural frequency, we calculated the eigenvalues of the disc model, taking the residual stress into account. The analysis was conducted in two steps. The first step was the substitution of the residual stress distribution with the thermal stress. The second step was the eigenvalue analysis considering the effect of the stress distribution on the differential stiffness.

Step 1. Reproduction of Residual Stress Distribution by Thermal Static Analysis

With respect to the symmetric residual stress distribution in an equilibrium state inside of the disc, we considered it appropriate to substitute the residual stress distribution with the thermal stress distribution.

The measurement shows the presence of compressive residual stress on the entire disc surface. To balance that, tensile residual stress presumably exists inside the disc.

We have therefore tried to formulate the temperature distribution of perpendicular direction (i.e. thickness direction). Note that as the top and bottom surfaces are equally treated in the production process, we considered that the residual stress distribution in the perpendicular direction was symmetric from the center of thickness. Also, the thermal distribution in the circumferential direction was regarded as uniform from the measurement in the preceding paragraph.

We have thus set Temperature Equations (1) and (2), given below, to reproduce the distribution of the residual stress component in the circumferential direction.

$$T_{surf}(r) = \begin{cases} T_0 + \frac{T_1}{L-R_{in}}(r-R_{in}) & (R_{in} \leq r < L) \\ T_0 + T_1 & (L \leq r < R_{out}) \end{cases} \quad (1)$$