# 802. Modifying the brake drum geometry to avoid self-excited vibrations and noise

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**Abstract.** A squealing noise of 50 dB was measured on a vehicle homologation test at around 900 Hz on the existing brake drum design, mounted on the rear axle of the mid-sized passenger automobile. Therefore, analysis of eigenfrequencies of the original drum design was performed using the impact hammer test and numerical analysis. It was established that a critical mode shape 0/2 exists at around 900 Hz, exactly where the squeal noise was recorded at the brake road noise evaluation vehicle test. The analysis was carried out with the intention to eliminate the possibility of the squealing noise by increasing the critical mode above 900 Hz. The relation between different brake drum modifications parameters and the eigenfrequencies was determined and the best solution was obtained. The first eigenfrequency of the proposed drum design was increased by 58 Hz and the difference between the in-plane and out-of plane mode shape was sufficient. We can conclude that the modified drum design will not have squeal issues at 900 Hz as there are no eigenfrequencies of the brake drum in that range and therefore the problem of the loud brake is solved.

**Keywords:** brake squeal, drum brake, eigenfrequency, impact hammer test, finite element method.

## Introduction

Reduction of the friction-induced noise, especially under nonstationary friction conditions, is one of the most complex problems in the transport industry. Expenditures on experimental and theoretical research studies on noise and vibration have recently constituted more than 50 % of the total budgets of the leading companies dealing with frictional materials and brake systems [1, 2]. The braking process in an automobile involves a contact of metallic solids sliding against each other, which sometimes generates undesirable noise, vibration, and harshness (NVH). Brake squeal is an annoying, usually single-tone and high-pitched noise. Brake noise below 1 kHz often depends strongly on the modal characteristics of suspension systems [3]. Brake squeal is a phenomenon of dynamic instability that occurs at one or more of the eigenfrequencies of the brake system. The excitation that is the cause of dynamic instability comes from the friction couple. The brake drum is acting like a resonator, as sound waves radiates from its surfaces. There are several of investigations, analyses and validation tools available such as vehicle test, dynamometer test, modal test – impact hammer test and finite element analyses. Once a problem is detected, common solutions to reduce the noise are decrease of the excitation (brake shoe chamfer design), increase of damping (insulator, grease) or shifting of component eigenfrequencies.

## Problem background

At the simplest level, the vibration characteristics of brake drums and discs are interpreted as bending vibrations of an idealized disc or plate. Although somewhat distinct from the real brake drums, this description is a natural starting point for understanding the physical and mathematical aspects of drum noise and vibration [2]. It was determined that the brake behaves similar to a perfectly axially symmetric plate (despite its asymmetry) [4].

Figure 1 represents the theoretical nomenclature of mode shapes for leveled round steel plate. Because the brake drum has a symmetric round shape, it strongly resembles those mode shapes. With the help of Cimos experience in this field, it was established that mode shapes 0/2 (circles/diameters), 0/3 and 0/4 are especially critical for generating brake noises [5].

Conceptually, a brake drum can be described as an ideal disc that vibrates in the bending direction, but some radial vibration, circumferential vibration, or combination of the two can also occur. Such motion is known as in-plane vibration [6]. For a brake component to be stable, the major drum out-of-plane an in-plane modes should be separated as much as possible. The in-plane mode should be at least one-third of frequency away from two major drum bending or out-of-plane modes [7]. The understanding of the vibration modes of the rotor will not only help to predict how a brake system may vibrate, but it will also be of great importance for developing countermeasures to eliminate the problem. In addition to the bending modes, the existence of in-plane modes is a further complication. Some analyses have determined that the in-plane modes, as well as the bending modes, can be the cause for some types of brake squeals [8].

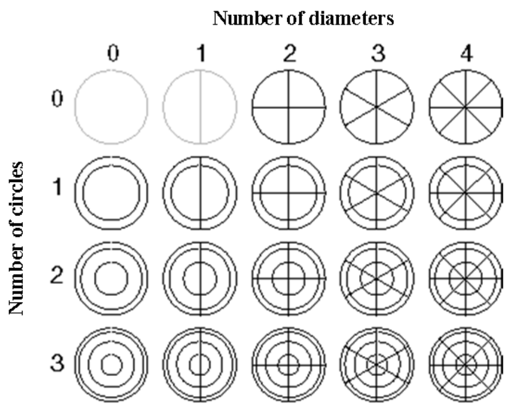



**Fig. 1.** Nomenclature of mode shapes

## Experimental procedure

In the Mojacar vehicle road brake noise evaluation test [9], the excessive noise was recorded at the rear brake mounted on a mid-sized passenger automobile. Because Cimos d. d. is the manufacturer and developer of brake drums and brake discs, the challenge was to modify the geometry of the current unsuitable rear brake drum. As shown in Figure 2, the excessive noise greater than 70 dB was mainly recorded at 900 Hz at every combination of temperature, speed and pressure, which indicated the resonance between brake components at this frequency.

The two main goals of this analysis are to investigate why the squeal noise occurs and to propose a brake drum design with an increased eigenfrequency to avoid the 900 Hz squeal problem.

At the beginning, the eigenfrequencies were investigated with the help of the impact hammer test. 30 pieces of brake drums placed on rubber supports were analyzed using B&K equipment. Excitation position is in the middle of the mud ring in radial direction. Accelerometer or the measurement point is placed at the same position on the opposite side as seen in Figure 3. Excitation force was set to be 100 N in radial direction.

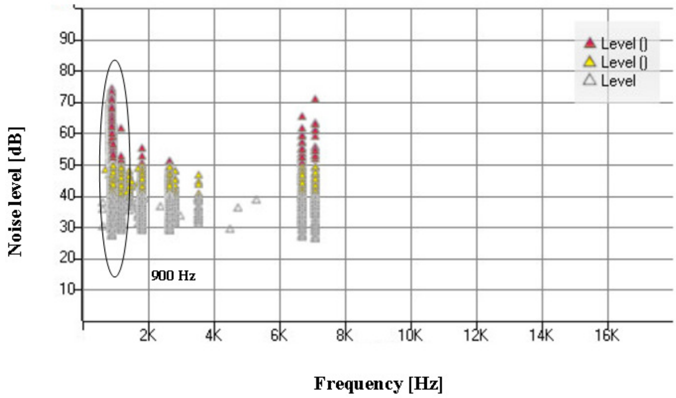



**Fig. 2.** Noise results of the considered rear brake drum

**Fig. 3.** Impact hammer test measurement setup

A very small deviation in frequency was noted between all measured specimens, indicating a very stable production process. Typical graphs, as shown in Figure 4, were obtained. The peaks in accelerometer voltage represent the eigenfrequencies. The first highest peak represents the first eigenfrequency that exists at 901 Hz. The second signal measures the impact force from the impact hammer, because the amount of force needs to be similar in all tests in order to excite the brake drum equally.

Mode 1 exists at 901 Hz (Figure 5), which was the exact frequency of noise generation in the vehicle test. It was determined that a critical mode shape 0/2 exists at this frequency, which is known to be particularly critical. Based on experimental observations, the out-of-plane (flexural) component of vibration is an important factor in the disc brake dynamics, including the audible noise [5]. We can claim that this eigenfrequency caused the excessive squeal.

## Numerical analysis

The numerical methods were used to analyze the current state and to propose modifications. Firstly, the material used for production of the brake drums (laminar grey cast iron DIN GG25 / EN-GJL-250) was carefully examined in Cimos laboratory and the acquired data was used as an input data in numerical analysis. Some basic material properties at room temperature are shown in Table 1.

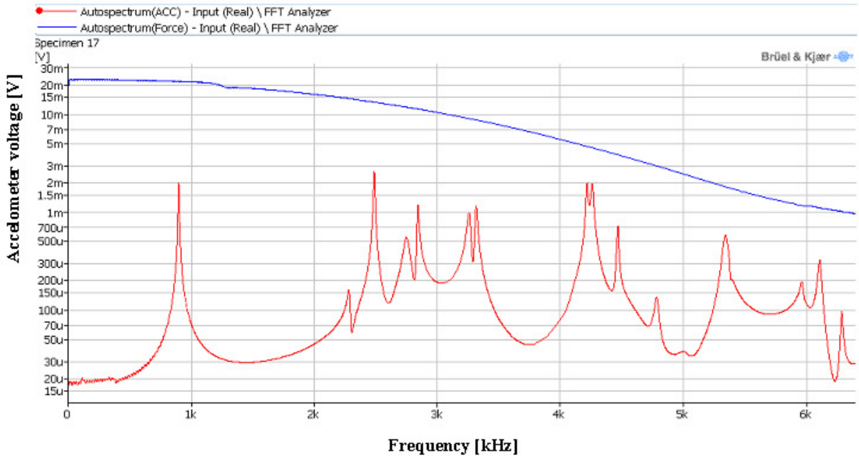



**Fig. 4.** Typical graph of brake drum frequency response

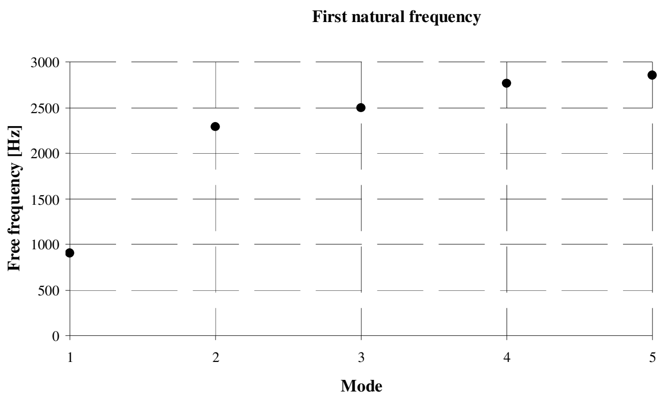



**Fig. 5.** Average eigenfrequency modes

**Table 1.** The data used for calculation

| | |
|---|---|
| Density – $\rho$ [kg/m$^3$] | 7200 |
| Elastic modulus – $E$ [MPa] | 100213 |
| Poisson's ratio – $v$ | 0,24 |

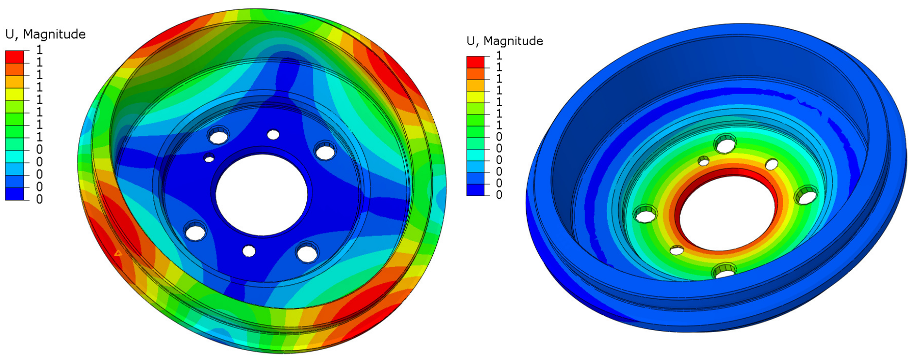
The brake drum was then modeled using Catia software [11] and carefully meshed in Abaqus FEM software with 79886 C3D4 elements [9]. A linear perturbation Free-Free frequency analysis was made using Lanczos method with eigenvectors normalized by displacement. The results are presented in Figure 6.

The first eigenfrequency mode exists at 893 Hz with a mode shape 0/2, which is known to be critical. This is the mode that caused the squealing problems at the vehicle test. A very strong correlation was established between drake drum bending modes and theoretical modes shown in

Figure 1. What is also important is that the span between the two first eigenfrequencies is 1260 Hz, which according to design proposals should be enough to ensure that mode shapes will not overlap [7]. It is now clear that the first eigenfrequency should be moved away from the critical area. A good correlation with experimental results was also found, but there is a small deviation because the brake drum was placed on three rubber supports at the impact hammer test experiment which may not represent the perfect free-free boundary conditions. As it can be seen from Eq. 1, for the simplest one-dimensional example, the stiffness directly increases the eigenfrequency:

$$f_n = \frac{1}{2\pi}\sqrt{\frac{k}{m}}, \tag{1}$$

where $f_n$ is eigenfrequency, $k$ is stiffness and $m$ is mass.




(a) (b)

**Fig. 6.** First 2 modes:
(a) 893 Hz, mode shape 0/2 (b) 2153Hz, mode shape 1/0

Naturally, the problem in a brake drum is much more complex, but the expression gives the direction how to address the problem.

## Results

With the bending mode shapes known, the geometry of the brake drum was analyzed to define the areas that can be modified and the stiffness to be increased in order to raise the fundamental eigenfrequency. When analyzing bending mode shapes, the weaknesses of the design were discovered. However, some areas of the design cannot be changed because of the limitations of neighboring systems, such as the wheel, hub, shoes and back-plate. Different parameters were investigated as shown in Figure 7 and tested using numerical methods. Every parameter was progressively increased and for each step the eigenfrequency was calculated. Only the first two eigenfrequencies were inspected in this research. The dependency of the eigenfrequency on the mass increase was determined and is represented in Figures 8 and 9. Thicknesses were gradually increased to 3 mm with the step of 0,5 mm and radius was increased to 6 mm from original 1,5 mm with the same step.

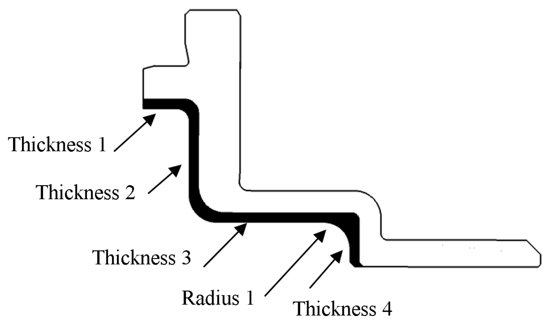



**Fig. 7.** Possible brake drum modifications

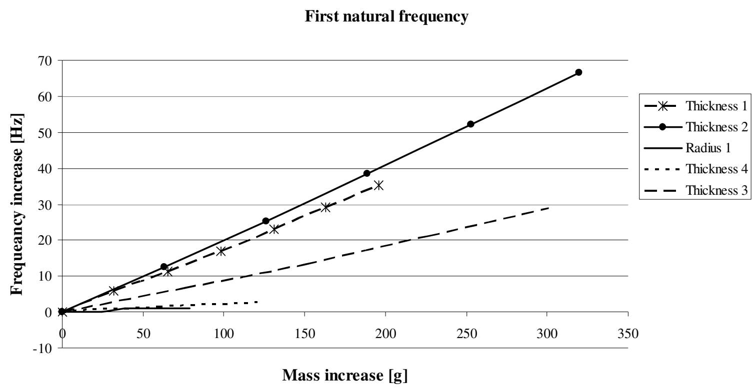



**Fig. 8.** First eigenfrequency dependency on mass increase

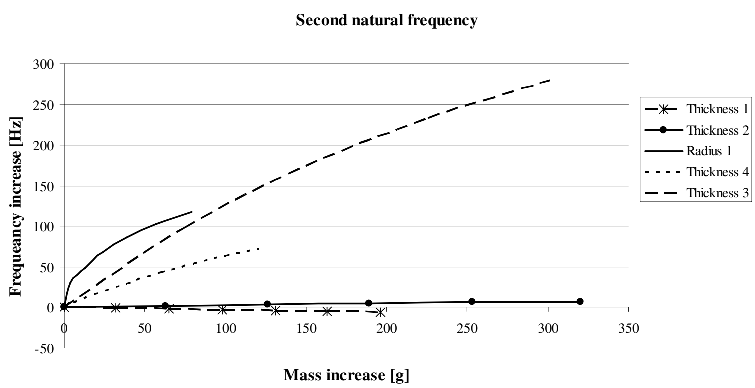



**Fig. 9.** Second eigenfrequency dependency on mass increase

Some of drum modifications have a greater influence on the first eigenfrequency and some on the second one. The tendency indicates that the more the modification is positioned away from the centre in radial direction, the more it influences the first eigenfrequency and vice versa. In order to increase both frequencies for the same value, the Thickness 2 (*T*) and Radius 1 (*R*) were selected for further analysis as they have the biggest influence on rising in-plane and out-of-plane frequencies. Final modification should influence both in-plane and out-plane mode shapes to ensure the shifting of both and to prevent them to exist too close together and potentially resonate when excited. The decision was made to leave the difference between the first two out-of-plane and in-plane frequencies as unchanged as possible (equation 7). Both selected modifications dependency functions were acquired from the dependency graphs using the MS Excel "Trendline" tool [10]. Equations 2 – 5 represent approximate dependency equations, where $R_1$ is Radius 1 of the first eigenfrequency, $R_2$ is Radius 2 of the second eigenfrequency, $T_1$ is Thickness 2 of the first eigenfrequency and $T_2$ is Thickness 2 of the second eigenfrequency. The maximal mass increase of final design should not be larger as 300 g, because this is the condition of the customer. It is important to keep the mass as small as possible because this influences costs, unsprung mass and vehicle fuel consumption. Equation 6 represents this boundary condition, where $x_t$ is the mass increase due to the increase in Thickness 2 and $x_r$ is the mass increase due to the increase in Radius 1.

$$R_1 = 0{,}0149 x_r \tag{2}$$

$$R_2 = 0{,}0004 x_r^{\ 3} - 0{,}0629 x_r^{\ 2} + 3{,}9292 x_r + 7{,}1059 \tag{3}$$

$$T_1 = 0{,}2081 x_t - 0{,}5416 \tag{4}$$

$$T_2 = 0{,}0228 x_t + 0{,}5547 \tag{5}$$

$$x_t + x_r = 300 \tag{6}$$

$$(T_2 + R_2) - (T_1 + R_1) = 0 \tag{7}$$

The system of 6 variables and 6 equations was solved and the solution was acquired: $x_r = 14{,}488$; $x_t = 285{,}511$.

Using the 3D modeler [10] the following thickness and radius increase that correspond to mass increase were defined: $T$ = 2,25 mm; $R$ = 4,80 mm.

Figure 10 shows that the first eigenfrequency was shifted away from original 893 Hz, which is known as the critical frequency area, where excitation in the considered brake system exists. As the stiffness was increased, the first eigenfrequency increased as well. The first eigenfrequency was increased by 58 Hz, which should be enough for the brake drum not to resonate because of the excitation around 900 Hz or a value that is a multiple of it. In order to ensure the non-overlapping of fundamental in-plane and out-of-plane frequencies not only the first eigenfrequency was changed, but also the second one from original 2153 Hz to 2213 Hz, for 60 Hz. The slightly larger increase in the second eigenfrequency than in the first one can be attributed to inaccurate modeling of the proposed masses, which was limited to resolution of 0,05 mm because of manufacturing limitations.

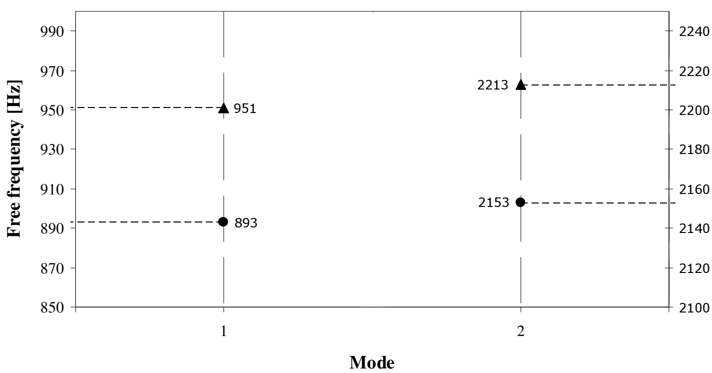



**Fig. 10.** The first and second eigenfrequency shift

## Conclusions

Analysis of the eigenfrequencies of the original drum design was performed as shown in Figure 6. It was established that a critical mode shape 0/2 exists at around 900 Hz, where the squeal noise was recorded at Mojacar road vehicle brake noise evaluation test. The conclusion was that mode 0/2 is responsible for the squeal issue on the brake drum. The analysis was made to increase the frequency of this mode above 900 Hz in order to eliminate squeal possibility but preserving the difference between 0/2 in-plane and 1/0 out-of-plane mode shape. The modification of the design was made with a minimal mass increase of 8 %. The brake drum design modification proposal was analytically obtained from dependency from several possible parameters that can be modified on brake drum geometry on the first two eigenfrequencies. Figure 10 clearly demonstrates that the proposed drum design has no eigenfrequencies near 900 Hz. The mode 0/2 has increased by 58 Hz and the difference between 0/2 in-plane and 1/0 out-of-plane mode shape remained essentially unchanged. We can confidently state that the modified drum design will not have squeal issues at 900 Hz as there are no eigenfrequencies of the brake drum there. But we cannot be certain that with the changed eigenfrequency spectrum the brake drum will not resonate at other frequencies. Further testing should be performed to determine that there are no excitations on friction couple that catches new eigenfrequencies.

## Acknowledgments

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