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Brake disc modal behaviour

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Abstract. To study the behavior of the braking system in noise and vibration analysis, due to its complexity, numerical models based on finite element analysis are used. The dynamic structural characteristics of the braking system components are strongly coupled to the modal behavior. The modal analysis may provide predictions about the vibration modes of some system components. The modal frequencies and vibration modes are crucial in the design phase, because they provide information on the behavior of the components in dynamic conditions. The modal analysis can be successfully used to validate the design of complex and costly mechanical structures that with high risk for vehicle operation. The current paper concerns the modal behavior of the braking system. The aim of the paper is to perform finite elements analysis of modal behavior for a brake disk. Validation of modal analysis is performed by experimental investigation. Taking into account the similarity of frequencies obtained, the model is partially validated by experimental approach.

1. Introduction

The vibration and noise raised by automobile braking have been one of the most difficult concerns since their inception, especially due to the fact that root causes are very difficult to discover. The vibrations nature is not repeatable due to the different parameters: pressure, velocity, temperature. These parameters create difficult conditions to investigate the phenomena.

In braking process low frequency vibration energy propagate throughout the structure and resonate with others vehicle components, meanwhile high frequency vibration induces undesirable noise levels. Disc brake squeal is often caused by high friction coefficient from pad – disc interaction, induced by structural instability. A NVH analysis is required in order to diminish the resonance conditions in the brake system.

Brake noise can be divided in few categories based in general on the excitation sources and occurring frequencies. More often squeal is the most perceived noise, occurring in braking process, appearing on frequencies between 1 kHz and 16 kHz. This paper analyses the squeal in the frequency interval from 1 kHz to 3 kHz.

The noise and vibration behavior of braking system has been analyzed since 1930 using a lot of theories and models. Specialized literature reveals papers that carry out extensive surveys on theoretical analysis and experimental investigation. Most of researchers tried to reduce squeal by modifying the disc technical characteristics or by changing the factors associated with the brake squeal.

Parra C. et al. carried out both theoretical and experimental analysis for each part of braking system and also for the entire system using finite element models. First mode was discovered on frequency of 844.03 Hz from theoretical model and 852.26 Hz in experimental test for braking disc. In this case were analyzed first 11 modes [1].

Nilesh K. K. et al. used a hammer excitation measuring technique for modal analysis in order to verify the physical vibration properties of a brake disc. The brake disc was analyzed at free-free boundary conditions and results were validated by experimental test. According to these analyses, first mode was observed at 1526 Hz in experimental analysis and 1557.6 Hz for finite element analysis [2].

Abu Bakar et al. obtained an error of 0.8% between the finite element analysis and measurement. The measurements provided the first eigenvalue at 937 Hz and 944 Hz in the finite element analysis [3].

Kung S. validated results from modal analysis of the brake components using a structural dynamic measurements on dynamometer bench. He obtained a difference of 7% between experimental and numeric analyses for disc [4].

Papinniemi A. et al. carried out theoretical and experimental analysis for braking system and also for each part. In experimental analysis an impact hammer procedure was used to find the circumferential in-plane modes of the rotor. In this way, the first and second circumferential in-plane modes were determinate [5].

Nouby et al. validated, in 2009, numerical analysis of the brake components and assembly models through experimental tests. By analyzing only the braking disc, they obtained second mode at 1220 Hz, for experimental investigation and 1303 Hz for numeric analysis, and third mode at 2551 Hz respective 2636 Hz [6]. In 2012, Nouby validated a numerical analysis of braking system with experimental tests. In this case the first tree modal eigenvalues were obtained for disc, this time with 0% error for the first mode, 1.2% for the second one and 3% for the third [7].

João Gustavo Pereira et al. compared the results obtained after they conducted numerical and experimental analysis on brake squeal. For the numerical analysis they used a finite element model, and for the experimental analysis a commercial full-scale brake dynamometer - Link 3900 – was used for testing the break performance of a Brazilian passenger car front brake on a NVH test procedure [8].

Holger Marschner et al. present in their paper a comparison of results obtained through finite element methods and measuring processes by 3-D laser vibrometry [9].

The book '*Recent studies of car disc brake squeal*' presents a modal analysis for braking system on each component. Regarding brake disc, the analysis was taking into consideration the first seven eigenvalues, first one was found at frequency of 937 Hz, with a 0.5% error comparing to the finite element method [10].

2. Finite element analysis

The modal analysis can be successfully used to validate the constructive design and structural integrity of complex mechanical structures, to reduce costs and avoid risk in operation. Finite element analysis can provide important information about vibration characteristics of a system, like coupling between the dynamic behavior, the pre-stress caused by the applied load, and friction characteristics.

Brake disc is one important component of the braking system in producing noise and vibration, and these phenomena could be amplified within the system.

Numerical analysis provides the resonant frequencies, as well as modal shapes, and thus allows the accelerometers to be positioned in the areas of interest.

The accuracy of numeric analysis is often limited, especially due to part geometry, material properties, and coupling parameters between subsystems. For this reason it is important to validate the finite element model through experimental analysis.
In the modal analysis were analyzed vibrations under 3000 Hz in order to obtain the mode shapes for the brake disc.

In the 3D model design were taken into account the next hypotheses: the material of the brake disc is homogeneous and isotropic; the inertia of the brake disc is not taken into consideration during the analysis. The geometry of the ventilated disc was realized using CATIA V5 software (Figure 1).

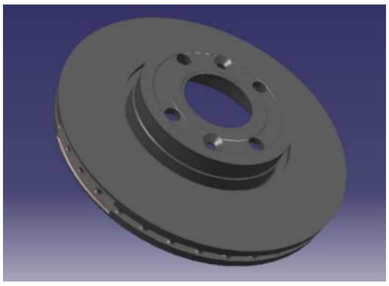

Figure 1. Brake disc modeled using CATIA software

After the design was realized in CATIA, the brake disc model was imported in the ANSYS finite element software. The first step is to define the material properties of brake disc (Figure 2). The material used for the brake disc is cast iron GL 11.

1	Property	Value	Unit
2	Density	7,1E-09	kg mm^-3
3	Isotropic Secant Coefficient of Thermal Expansion		
6	Isotropic Elasticity		
7	Derive from	Young's M...	
8	Young's Modulus	1E+08	Pa

Figure 2. The selected material properties for brake disc in ANSYS software

The next step for the modal analysis is meshing the analyzed part. The mesh has tetrahedral type finite elements with dimensions of 4 mm (Figure 3).

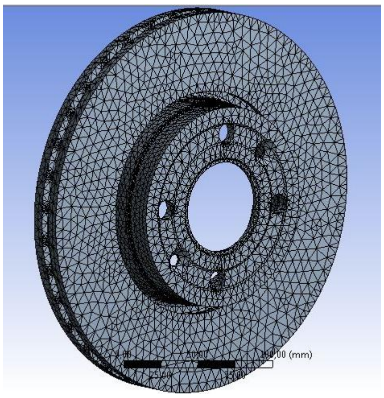

Figure 3. Brake disc mesh

The brake disc is analyzed in the free-free boundary condition, which allows the structure to vibrate without interference from other parts, being easier to correlate the mode shapes with each natural frequency.

Five eigen modes of the brake disc are presented in the interval of 0-3000 Hz (Figure 4).

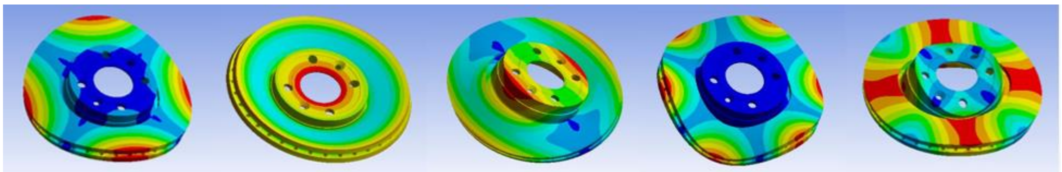

Figure 4. First five modes strain energy distribution and deformation on disc

All values of resonant frequencies obtained from modal analysis are presented in Table 1.

Table 1. Resonant frequency values for brake disc

Eigen Mode	1	2	3	4	5	6	7	8	9
Frequency [Hz]	1086.1	1092.8	1935.7	2283	2289.5	2423.4	2430.6	2589	2599.4

3. Experimental study

Experimental modal analysis is used, in general, for understanding dynamic characteristics of structures in real conditions such as natural frequencies, damping factors and mode shapes.
In order to validate the numerical model some experimental analyses on eigenvalues were performed on brake disc (Figure 5), and also on brake disc - wheel hub assembly (Figure 6).

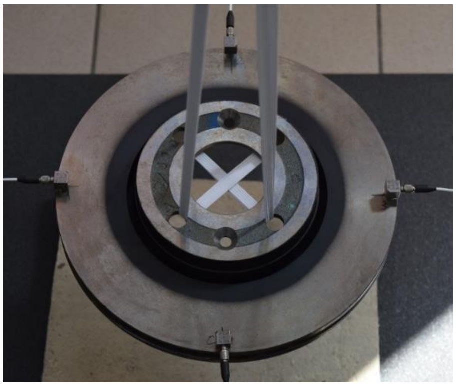

Figure 5. Brake disc in free – free condition

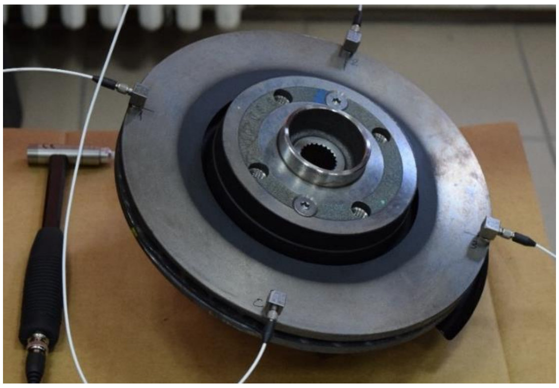

Figure 6. Brake disc - wheel hub assembly

Experiments were made on a front braking system of a SUV.
Testing with an impact hammer determines the resonant frequencies. The force sensor with a piezoelectric transducer in the hammer head provides the amplitude and frequency content of the energy stimulus that is applied to the test object. The impact hammer was used to provide the excitation directly into the in-plane direction. Hammer tips permit the energy content of the force impulse to be adapted to suit the requirements of the item under test. In order to provide sufficient energy for high frequencies, the hard tip is selected for this kind of material and for this range of frequencies.
The disc was elastically suspended to induce free-free condition that allows the structure to vibrate independently.
In the analysis four piezoelectric single-axis accelerometers were used, one impact hammer with two different hardness tips and a dynamic signal analyzer.
Accelerometers are used in conjunction with the hammer in order to provide a structural response during impact and after application, as well as the frequency response function.
To obtain sufficient vibration modes for the analysis, four accelerometers were placed in four diametrically opposed points.

For the given impulse, the response is captured by all four accelerometers, and then converted using a signal conditioner into electrical signals, which were acquired and displayed. Signals from all four accelerometers and impulse hammer were acquired and processed in LabView Software (Figure 7), thus the natural frequencies were obtained using frequency analysis. Impulse given by impact hammer has a quick damping, while accelerometers still record the disc vibration.
Vibration acceleration is measured at point four in axial direction, while the excitation was provided by the impact hammer in the normal direction near point four; for this reason only the bending modes were obtained. Frequency response function is almost the same for all four accelerometers, with four peaks at the same frequencies (Figure 8 – Figure 11).

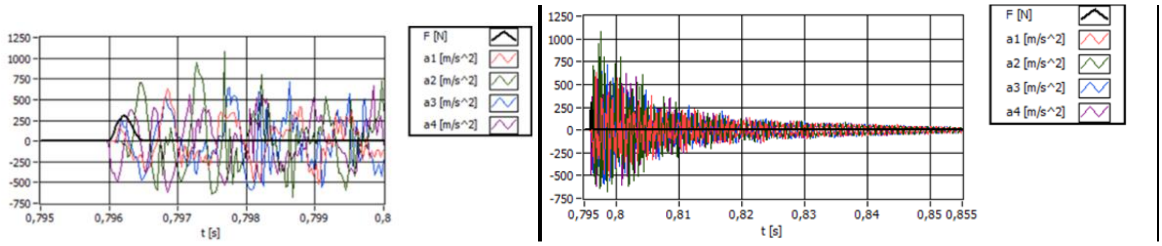

Figure 7. Force and acceleration vs. time for all four accelerometers (left - the first 4 ms; right - entire process)

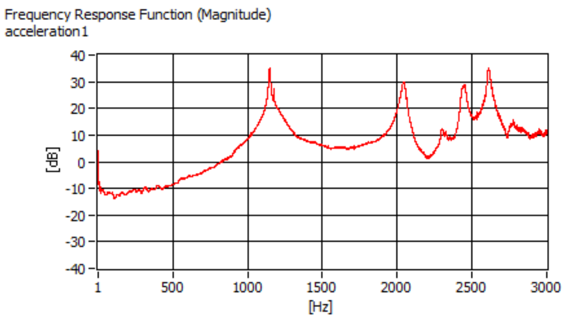

Figure 8. Frequency response function for impact at 90 deg. offset from accelerometer 1

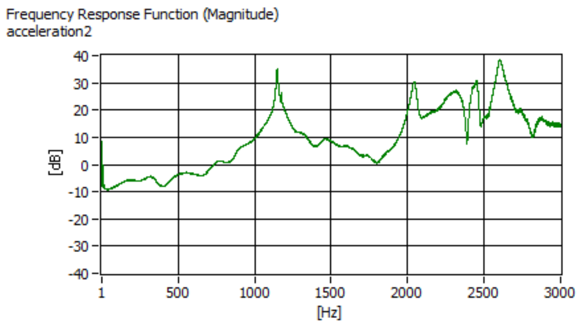

Figure 9. Frequency response function for impact diametrically opposed to accelerometer 2

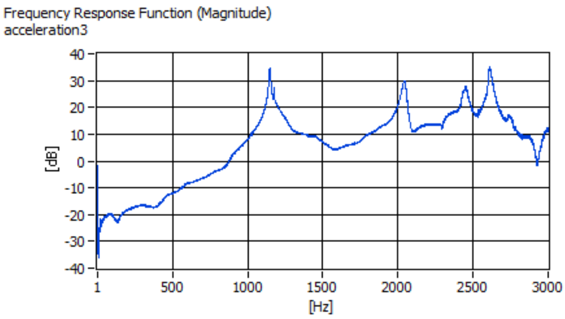

Figure 10. Frequency response function for impact at 90 deg. offset from accelerometer 3

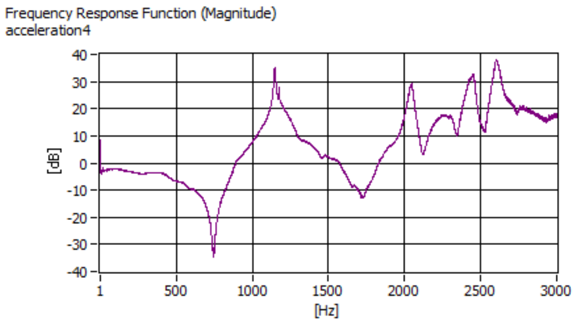

Figure 11. Frequency response function for impact near accelerometer 4

Radial vibration measurements were performed by keeping three accelerometers in the same position and displacement accelerometer 1 in radial plane. In this way only accelerometer 1 was able to provide information.
From the measurement of radial vibration acceleration of the brake disc, a radial eigenvalue is found (the peak marked with green in Figure 12).

In the measurement performed on brake disc - wheel hub assembly, two more resonant frequencies were obtained, most likely generated by wheel hub (green indicator in Figure 13).

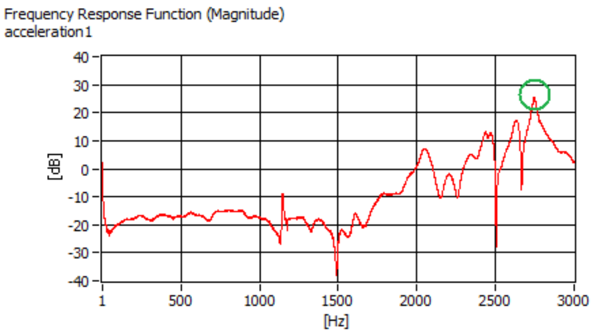

Figure 12. Frequency response function for impact near accelerometer 1, in radial direction

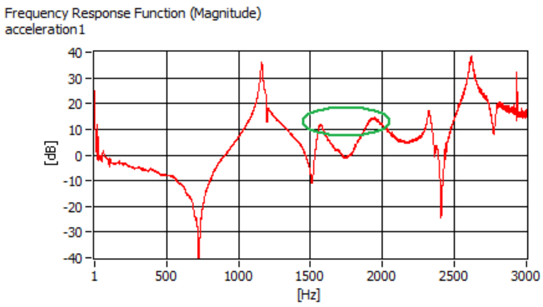

Figure 13. Frequency response function for brake disc - wheel hub assembly near accelerometer 1

Table 2. Resonant frequencies for brake disc and disc - wheel hub assembly

Frequency obtained theoretically [Hz]	Frequency obtained experimentally [Hz]	Direction	Observations
1093	1150	axial	disc
-	1565	axial	disc - wheel hub assembly
-	1930	axial	disc - wheel hub assembly
1936	2050	axial	disc
2290	2300	axial	disc
2431	2450	axial	disc
2600	2610	axial	disc
-	2740	radial	disc

For finite element model validation, natural frequencies obtained from simulation were compared with the ones determined by experimental analysis. Analyzing values from Table 2, there is similarity between resonant frequency values: 1093 Hz in finite element model analysis and 1150 Hz in experiment; also 2290 Hz in finite element model analysis and 2300 Hz in experiment test; also 2431 Hz in finite element model analysis and 2450 Hz in experiment test; also 2600 Hz in finite element model analysis and 2610 Hz in experiment test. Thus the model has been partially validated.

4. Conclusion

This paper presents numerical analysis of brake disc, using ANSYS software, for a realistic finite element model. Numerical analysis investigated vibrations under 3000 Hz and provided first five eigen mode and their modal shapes deformation. Finite element analysis leads to information about the mode shapes and correlate them with corresponding frequencies. Initially the experimental analysis was performed on brake disc, followed by an analysis on disc - wheel hub assembly, for both was used an impact hammer in order to determinate the resonant frequencies.

In order to validate the results obtained from numerical analysis, experimental analysis was needed. Taking into account the similarity of frequencies obtained, the model is partially validated by experimental approach.

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