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THE INFLUENCE OF MATERIAL CHARACTERISTICS AND ALTERED GEOMETRY DUE TO PROCESS ON THE VIBRATIONAL BEHAVIOUR OF THE BRAKE DISK

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Abstract: *Lately, in the field of car design, the level of noise perceived by the client has been continuously decreased as a result of progress. An important element of this approach is the study of vibration induced by braking behavior of the coupling pad / disk brake. Based on observations made over time regarding brake noise of various combinations of pad / disk, this paper proposes a method to anticipate the possibility of producing noise when braking due to brake disc , as well as control the manufacture, to avoid possibility of its occurrence . This article does not address the mechanism which produces noise brake in the coupling pad / disk, which is fully yet unexplained by researchers. The disk is considered in two cases (made of different materials or geometric errors caused by drift in the manufacturing process), at a standard definition of the pad. A disk identified as " noisy " brake (type β) and a disk identified " silent " (type α) are compared and the influence on the potential presence of noise caused by changing the width of vent fins disk when changing casting parameters is also analyzed . For this, it is proposed to calculate the natural vibration modes and the dynamic stiffness brake disk in different definitions, based on the finite element method.*

Keywords: brake disk; vibrational behaviour; material characteristics.

INTRODUCTION

The objective of the modal analysis is to identify and evaluate the dynamic features of a structure and relies on both analytic and experimental procedures.
The analysis of the dynamic characteristics of the structure can be made by determining the natural vibration modes due to an initial impulse. This reveals the extent to which the part is deformed during vibration. The natural vibration modes can be analyzed either by considering damping or not.
The advantage of using modal analysis for the study of a system is that it allows the evaluation of the dynamic characteristic of the structure and also the determination of natural frequencies. Once they are determined, the natural frequencies of a structure can be used for experimental analysis and improving dynamic behavior.
The purpose of the present study is to identify the vibrational behavior of the vented brake disk and interpretation of results in order to determine the potential for noise due to the bream disk. There is no reference to the mechanism of noise generation in the coupling pad/ disk. Different opinions on this subject aren't widely accepted - see [3], [7], [9], [10], [11], [13].

CLARIFICATIONS ON THE MODAL ANALYSIS CALCULATION

A structure is sensitive to vibrations depending on its modal characteristics. The sensitivity to vibrations of any structure depends on the following parameters:
- its natural frequencies signifying max/min amplitude oscillations of the structure

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- natural vibration modes, deformations due to oscillations of max/min amplitude, associated with each natural frequency
- modal damping meaning the amplification of the corresponding vibration.

These parameters depend on the physical features of the structure, on the limit conditions and also on the characteristics of the material.

Exciting the system up to a frequency similar to the natural one induces increasing vibration amplitude and the system will be exposed to the risk of resonance. This might lead to exceeding the elastic limit of the system, to its deterioration or to the production of a noise beyond the accepted limit (120dB). The human ear is sensitive to acoustic pressure from 0dB up to 120dB. Noise level up permissible limit means sound pressure level > 20 Pa (120 dB). For this reasons, on the brake system design will be find technical solutions for diminishing the functional noise as much as possible so that it should be bearable.

Structure damping fewer than 15% will be ignored in calculations of modal analysis because natural frequencies are insignificantly affected.

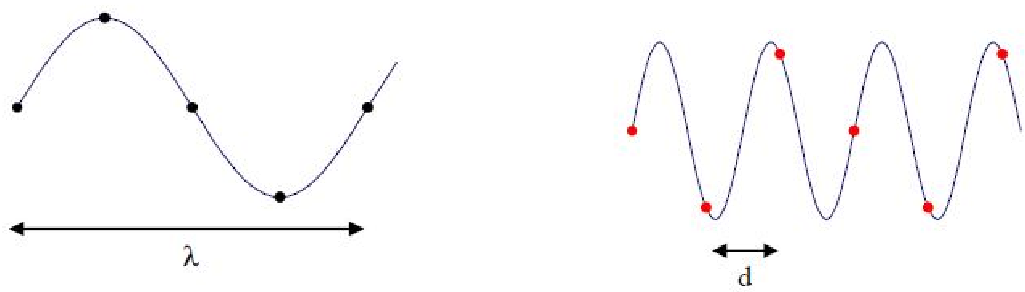
The calculation of modal analysis allows better choice of frequency range studied to achieve the frequency response calculation.

The deformation energies that appear in the most sensitive points (such as the center of the disk or the coupling points on the wheel hub) and generate important vibrating movements will also be calculated

THE USE OF THE FINITE ELEMENT METHOD FOR CALCULATION IN MODAL ANALYSIS

The finite element method is a technique which allows the evaluation of technical solutions by calculation. It helps to analyze the parameters characteristic to continuous mediums such as displacements, deformations or strains.

The basic concept in the process of numeric modeling is that of approximation by meshing**.** It is essential that the meshing be accurate enough in order to represent the modal deformations correctly. Also, it is necessary that there should be at least four elements on the wavelength, as can be seen in Figure 1.



Long wavelength, which is the correct representation of deformation: four elements on a wavelength

Short wavelength, which is the wrong representation of deformation

Figure 1. Sizes of the finite elements varying with the wavelength.

This implies that a more accurate meshing be made while increasing the frequency of study, because the wavelength decreases with the increase of frequency. For the frequency ranges studied in the case of the brake disk, the density of meshing used for dynamic study is the same as the one used in the case of static study. It is considered that six elements on the wavelength are adequate for an accurate dynamic study.

According to the conditions stipulated in the contracts of automobile producers, and to the type of dynamic calculation, there are two kinds of modal analysis:

1. 'free-free' ,when the constraints are not taken into consideration;
2. with imposed conditions, when the constraints are taken into consideration.

In the first case, the studied structure is considered suspended, subject to no effort or constraints. By using this type of analysis, the intrinsic characteristics of the structure can be obtained. It is the type of analysis to be performed in this study.

The second type, with special limit conditions, is dealt with in most cases in actual functioning conditions.
For this study will be analyzed. This case will be the subject of another study.
A meshed car part will have a certain number of vibrations equal to the sum of freedom degrees from which is subtracted the number of freedom degrees annulled by constraints. In "free-free" case, degrees of freedom by constraints are not canceled.
In order to solve the movement equation for the natural vibration modes, a simplified form is necessary, in order to use the damping and loading factors. The equation of the dynamic equilibrium for a multiple-degrees-of-freedom of system in motion done by Mechanical science is:

$$[\mathbf{M}]\{\ddot{\mathbf{u}}\}+[\mathbf{C}]\{\dot{\mathbf{u}}\}+[\mathbf{K}]\{\mathbf{u}\}=0 \tag{1}$$

Where [**M**] is the mass matrix, **[C]** is damping matrix, [**K**] is the stiffness matrix, and **u** is the vector of movement.
A harmonic solution would have the following form:

$$\{\mathbf{u}\}=\{\Phi\}\sin\omega i \tag{2}$$

Where: Φ is the eigenvalue of vector and ω is the eigenvalue angular frequency.
By replacing the solution in the movement equation and by simplification get:

$$([\mathbf{K}]-\omega^2[\mathbf{M}])\{\Phi\}=0 \tag{3}$$

Relying on modal analysis, the normal solution to the problem of natural vibration modes is reduced to:

$$\det([\mathbf{K}]-\omega^2[\mathbf{M}])=0 \tag{4}$$

In this case, equation (3) is reduced to:

$$[\mathbf{K}-\omega_i^2\mathbf{M}]\{\Phi_i\}=0 \tag{5}$$

Where i = 1,2,3…
Each specific value $\lambda_i=\omega_i^2$ and eigenvalue of vector Φ_i define a free oscillation mode of the structure.
The relation between certain values λ_i, frequencies f_i and eigenvalue angular frequency ω_i is:

$$fi=\frac{\omega_i}{2\pi} \qquad \omega_i=\sqrt{\lambda_i} \tag{6}$$

ANALYSIS OF THE NATURAL VIBRATION MODES OF THE VENTED BRAKE DISK

The analysis of the natural vibration modes of the vented brake disk relies on the CAD model of the car part - figure 2, and the finite element method was used as a calculation technique.

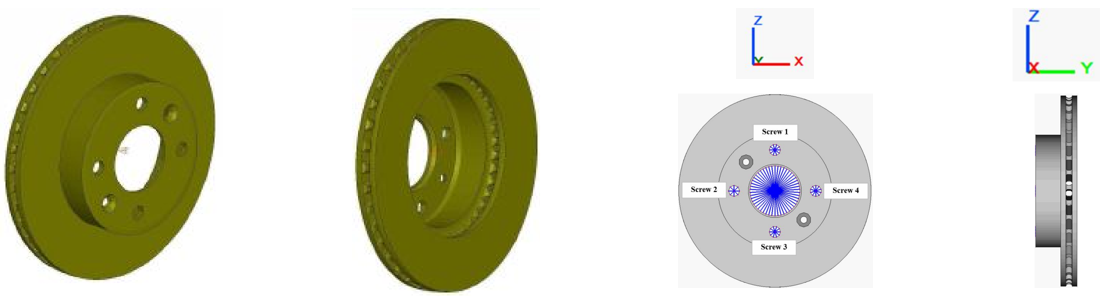


Figure 2. The vented brake disk studied.

Nastran SOL 103 software was used as CAD, in order to make the model and to perform the calculation of modal analysis.
Calculation of modal analysis was done by taking into account the natural vibration modes that have high deformation energies. Natural vibration modes with high deformation energies can cause the car part to break if the elastic limit of the material is exceeded, or can generate noises through its resonance with other parts (e.g. the break plate together with the disk make a friction couple). This is the reason why modal analysis was done by calculation and the results compared with them ones, found out experimentally.

RESULTS OBTAINED BY USING MODAL ANALYSIS CALCULATION FOR THE VENTED BRAKE DISK

The calculation of natural vibration modes under internal limit conditions, in the “free-free” situation, was done by applying a mesh to the parts. On a frequency up to 5,500 Hz, ten vibration modes of the vented disk under analysis were found. The features of Type α and Type β, as well as the masses of the models analyzed are shown in table 1.

Table 1. Characteristics of the materials and masses studied.

Material	Young Module	Density	Poisson coefficient	Masse of the model [kg]	
	MPa	kg/m3	-	initial	Reduced width of wings
Type α cast iron	121000	7300	0,29	4,197	4,025
Type β cast iron	102200	7200	0,29	4,139	

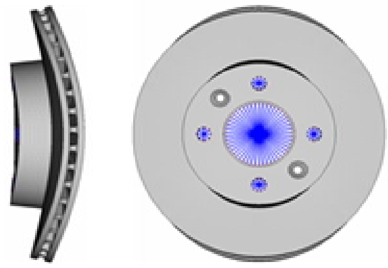
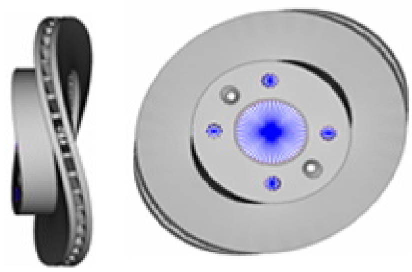
The results of natural frequencies by calculation are presented in table 2, and figure 3 shows the images corresponding to each natural vibration mode.

Table 2. Specific frequences of the brake disk in different configurations.

Mode	Frequency[Hz]		
	Initial type α cast iron disk	type β cast iron disk	Reduced type α cast iron disk
1	1633	1511	1510
2	1638	1516	1513
3	3113	2880	2890
4	3506	3244	3192
5	3508	3247	3196
6	3779	3497	3488
7	3782	3500	3491
8	3889	3599	3519
9	3903	3612	3532
10	5510	5099	4968

Definition of deformation energies is relevant information for the identification of the zones that have to be modified in order to eliminate or diminish the problems caused by vibrations. The zones on the brake disk that have significant deformation energy are those most affected by the deformation of the considered module. The modification of these zones will have as a result the modification of frequencies and of the deformations induced by those.
For this study, the values of brake disc natural frequency calculated in different configurations, serve as controls values for subsequent calculations (calculation of Frequency Response Function) and serve also to validate them against their natural frequencies, experimentally identified.

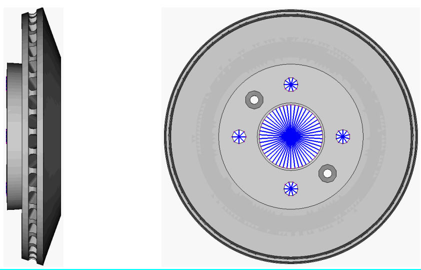
The flexion of the active surface of the disk break is an example of its deformation mode during braking.

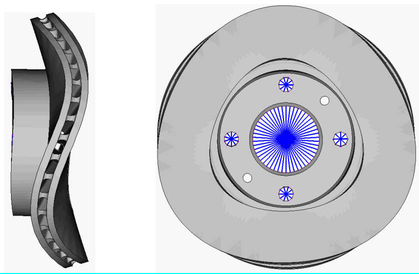
Mod 1 - 1633 Hz

Mod 2 – 1638 Hz

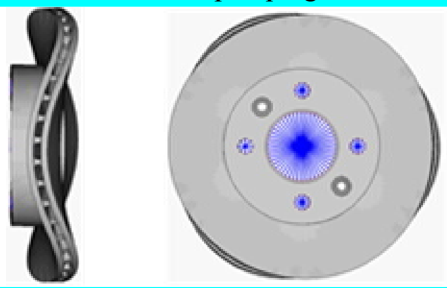
Track flexion on one diameter



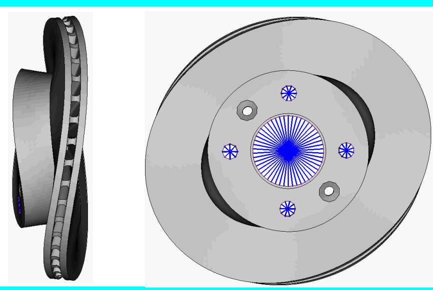
Mod 3 – 3113 Hz
Track pumping



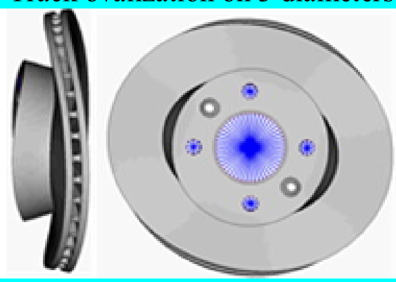
Mod 4 – 3506 Hz
Track ovalization on 3 diameters



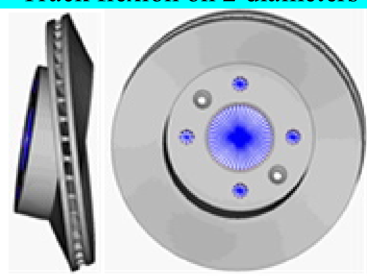
Mod 5 – 3508 Hz
Track ovalization on 3 diameters



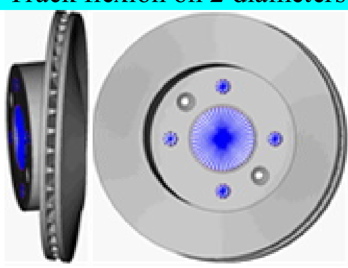
Mod 6 – 3779 Hz
Track flexion on 2 diameters



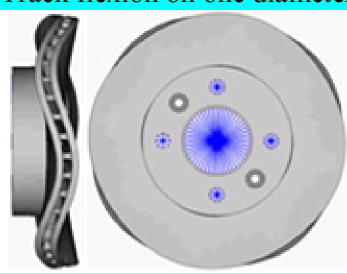
Mod 7 – 3782 Hz
Track flexion on 2 diameters



Mod 8 – 3889 Hz
Track flexion on one diameter



Mod 9 – 3903 Hz
Track flexion on one diameter



Mod 10 – 5510 Hz
Track ovalization on 4 diameters

Figure 3. The visualization of the natural vibration modes of the disk.

EXPERIMENTAL RESULTS REGARDING THE IDENTIFICATION OF THE NATURAL FREQUENCIES OF THE VENTED BRAKE DISK

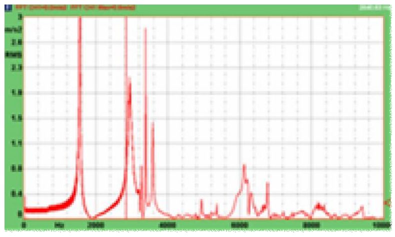
For experimental purposes there has been used a measurement chain composed of the following elements:

- accelerometer PCB type, model 353B04, sensitivity 1.011mV/m/s^2;
- measurement amplifier;
- Hammer.

Software settings:

- accelerometer, filter below 1Hz, FFT analysis up to 10,000Hz;
- FFT analysis was set on 3,200 calculation points.

The Soundbook integrated system was used to this purpose, together with SAMURAI (SINUS Messtechnik Gmbh) software. The latter is targeted on data collection, on analyzing the signals and on the command of various machines and external equipment.
The hammer with the help of which the car part is hit in order to induce vibrations is specially designed for measuring vibrations, and there is a force distributor incorporated in it.
Several experimental measures were performed, with the transducer being placed in different positions on the disk. Figure 4 is represented the way as the distributor is assembled on one side of the disk, and the results got by applying the force to the side opposite the position of the transducer**.**



The equipment used for measurement Positioning of the distributor Diagram of measured frequencies

Figure 4. Measurement of natural frequencies of the vented brake disk.

In all the cases studied, the first natural (fundamental) frequency is around 1,590 Hz, a value close to the one calculated (1,663 Hz). This value is not influenced by the position of the transducer or the direction of the hit. The lower value of the measured frequency, compared with the value obtained by calculation, is due to the structural damping of the part. The difference between the value of the first natural frequency measured and the calculated one is less than 5%.
The experimental results confirm the fact that the calculation model used is accurate and can, therefore, be used for other types of dynamic calculation as well.

ACOUSTIC STIFFNESS CALCULATIONS USING THE FUNCTION OF RESPONSE IN FREQUENCY (FRF)

The Frequency Response Function calculation completes the current study, providing a graphical overview of the vibrational behavior for the disk brake solutions studied. The Function of Response in Frequency (FRF) characterizes the dynamic (or acoustic) rigidity of a system in certain points and is defined by the acceleration induced to a point by unitary excitation. In practice, the respective system is excited with the help of a vibrator or a hammer (see preceding paragraph). Both, the effort applied and the induced response to vibration, are measured simultaneously.
FRF is an important parameter as it allows the evaluation of the force injected in the structure, which is afterwards transmitted to the mass of the structure. It is very important to know it in the excitation or fastening points because it is there that the vibrational transfer is achieved. The modal analysis allows measuring the natural frequencies of a structure and also avoiding resonance, already knowing the excitation frequencies. FRF will help measure the level of vibration in the structure.
FRF, represented in decibels, is expressed by equation [4]:

$$R\gamma(\mathrm{dB}) = 20\mathrm{x}\log\left(\frac{\gamma}{F}\right) \quad [1/\mathrm{kg}] \tag{7}$$

where: γ is the acceleration [m/s^2] and F is the excitation effort [N].

In this case, it is necessary to identify the frequency in the acceleration, at the desired point in the desired direction. Plot then the curve in logarithmic representation.

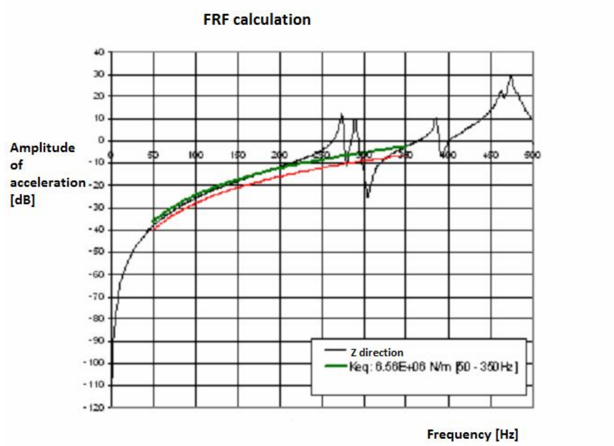


Figure 5. Graphic representation of a FRF curve.

The curve in Figure 5 represents FRF in the Z direction of a certain point on the disk. The peaks represent the natural vibration modes of the disk.
At natural frequencies, the response of the structure is amplified by resonance. In order to assess the value of FRF either calculated (in case of the present study) or measured, will analyze the curves resulting from the calculation in the range of frequencies audible spectrum. The results will be reported to the FRF brake disc curve that is taken as the benchmark, being silent (type α initially).
For acoustic vibrio validation, FRF is calculated only when the first natural frequency is higher than 30Hz, the inferior values being mechanical deformations without significant impact on the acoustic behavior. The frequency interval for which FRF is calculated varies from 0 Hz to a frequency equal to twice the 5th natural frequency calculated. Beyond this value, the acoustic effects is null because it is damped structurally in the mass of the structure.
The calculation is made in ANAMODE [4] and [5] and is based on the modal analysis calculation (NASTRAN-SOL 103).
For present study, applying of efforts for calculating FRF is done:
- in the 4 fastening points of the disk, on the directions : X, Y, Z;
- in the center of the disk (the X, Y and Z directions).

RESULTS OF THE FRF CALCULATIONS FOR THE VENTED BRAKE DISK

The comparative calculation of FRF for the vented brake disk made from different materials (of types α &β) and with modified geometry due to process (α type), is showed in Figures 6-10.

Type α brake disk is analyzed both in the initial version and simulating wear of the mold.
The compared analysis of acceleration amplitudes and of the frequencies to which they occur, for types of vented disk α and β, shows the quasi-identical behavior of the brake disk regarding the acceleration amplitudes in the excitation point, the difference being given by the level of frequency.
The same behavior is pointed also on the comparative analysis of amplitudes and frequencies of the initial type α cast iron vented brake disk and simulating the mold wear.
It is obvious that the gap is due to the nature of the materials used - in the first case - as well as the modified geometry of the part, in the second case. This is due to the rigidity of cast iron disc type α, more advanced in this respect than cast iron disc type β.

In the case of type α cast iron brake disk whose geometry was modified (reducing cooling fins thickness 2 mm as a result of the wornout mould), the frequency response analysis proves the influence of geometry on the vibrational behaviour of the structure (by modifying the frequencies of highest amplitude accelerations). This is the reason why this parameter should be kept under control within the car part Quality system.
The calculation of natural frequency, respectively the dynamic stiffness of the type α cast iron brake disk in relation to type β one, shows that the curves are shifted in proportion of approx. 10%.
The calculation of natural frequency and of the dynamic stiffness for the initial type α cast iron brake disk and for one which that simulate the geometric deviation due to mold wear, shows that the curves are shifted in proportion of approx. 2% the area of the fastening screws and max. 10% in the center of the disk - see the graphs of Figures 11 and 12.

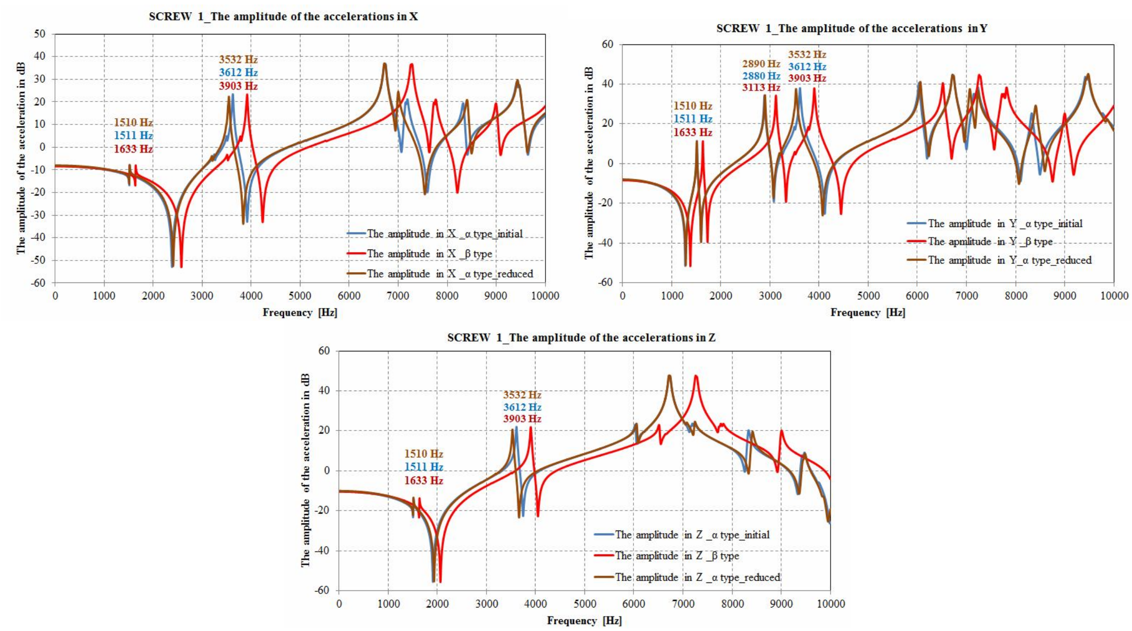


Figure 6. Comparative analysis of FRF for vented cast iron discs type α/β/type α reduced when effort is applied.

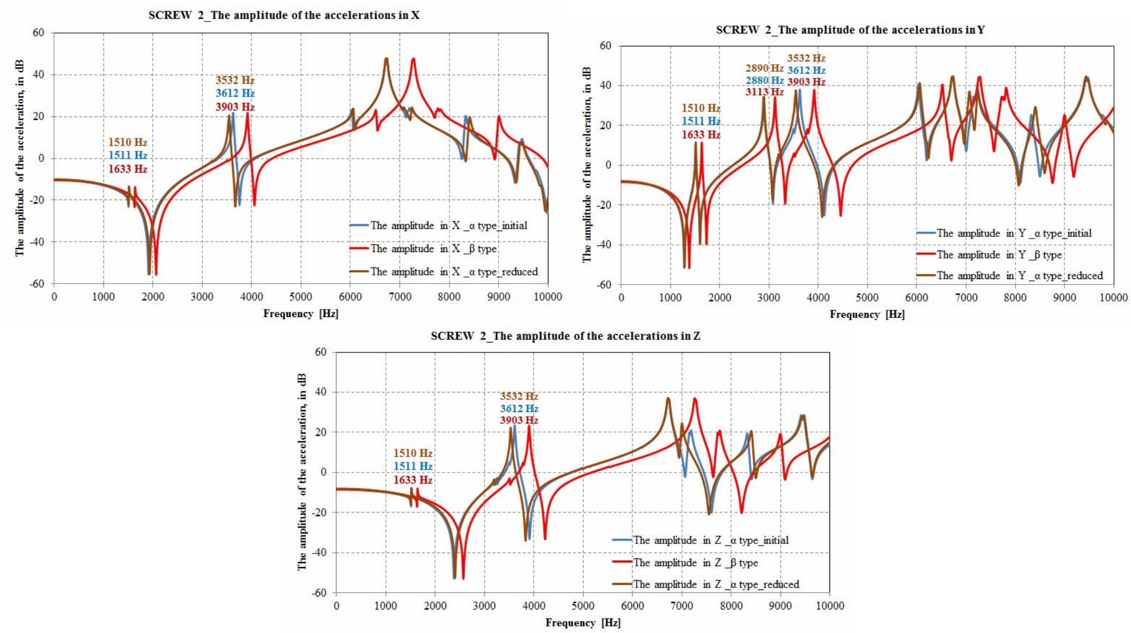


Figure 7. Comparative analysis of FRF for vented type α/β/type α reduced cast iron disks when effort is applied in fastening screw 2, on the 3 directions (vehicle coordinates).

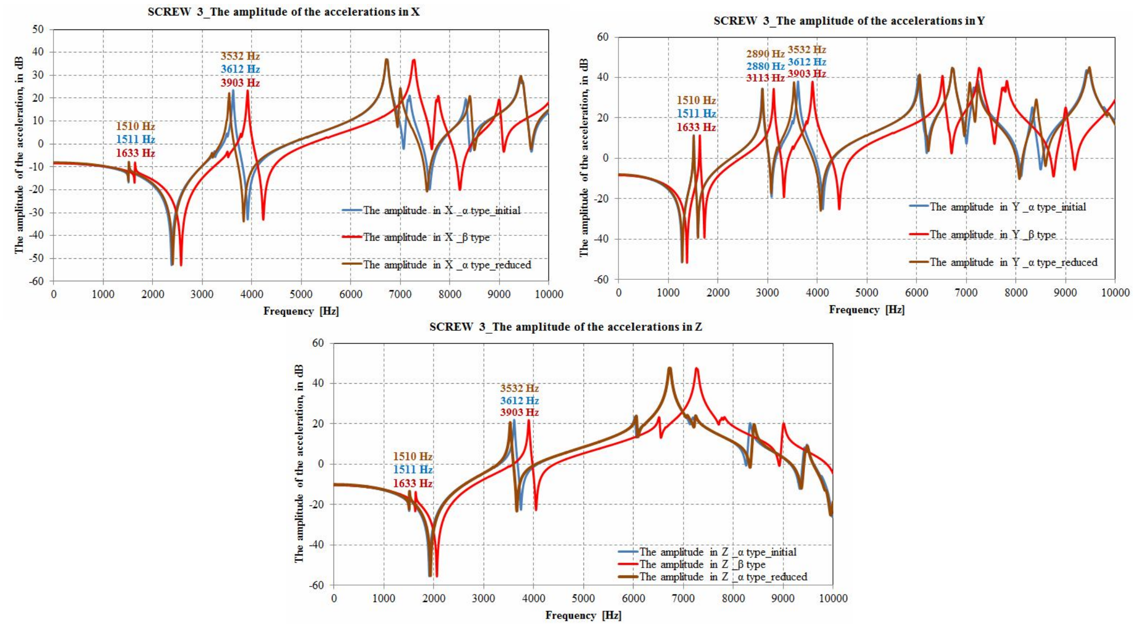


Figure 8. Comparative analysis of FRF for vented type α/β/type α reduced cast iron discs when effort is applied in fastening screw 3, on the 3 directions (vehicle coordinates).

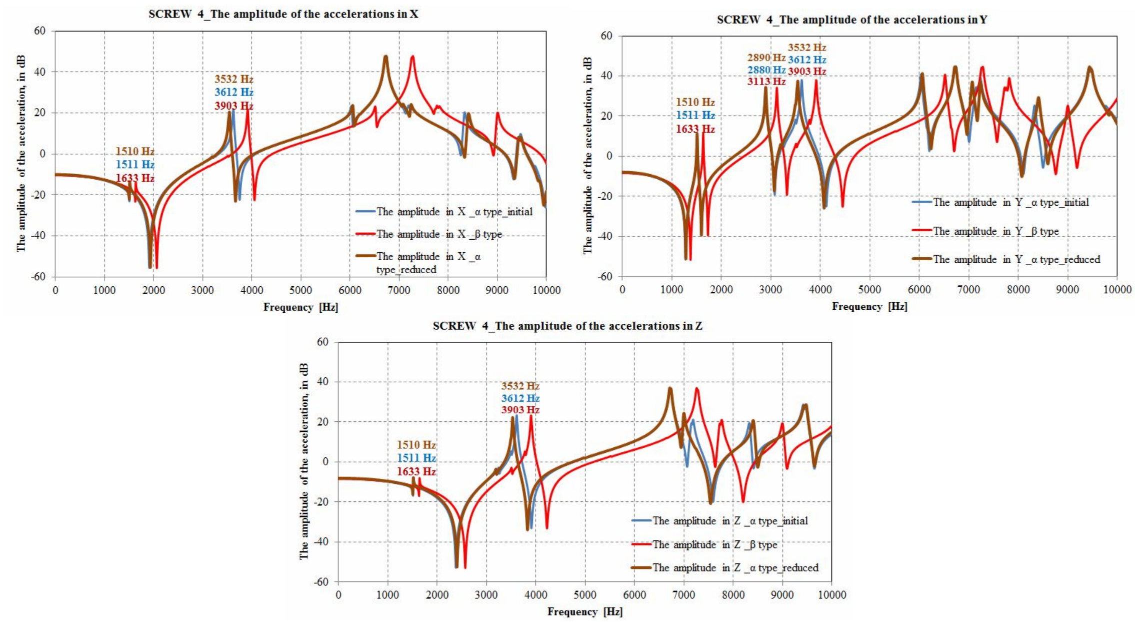


Figure 9. Comparative analysis of FRF for vented type α/β/type α reduced cast iron discs when effort is applied in fastening screw 4, on the 3 directions (vehicle coordinates).

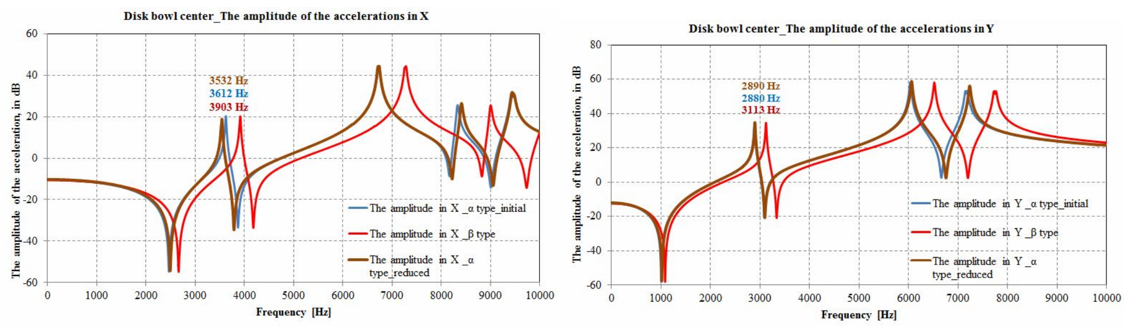


Figure 10. continued...

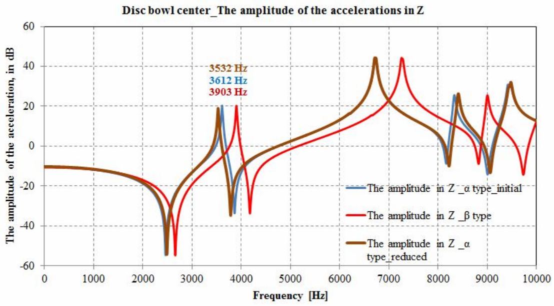
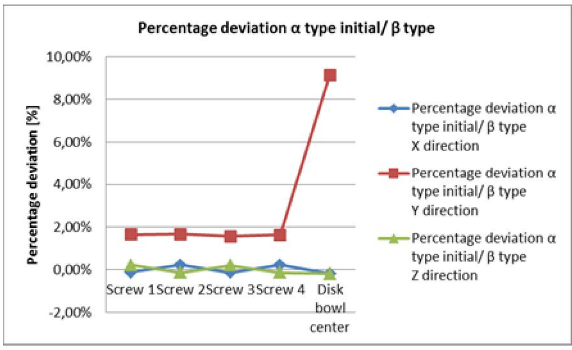


Figure 10. Comparative analysis of FRF for vented type α/β/type β reduced cast iron disks, when effort is applied in the disk bowl center, in 3 directions (vehicle coordinates).



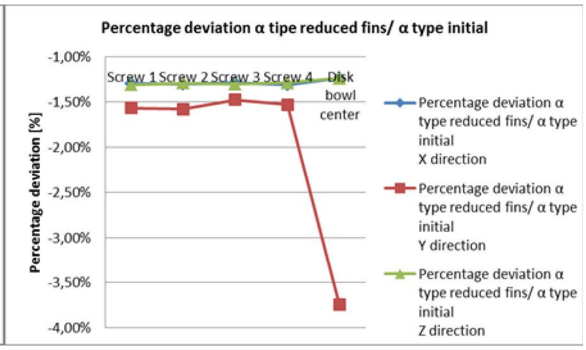


Figure 11. Difference in percentage regarding dynamic stiffness for cast iron vented disc α type compared to β type one.

Figure 12. Difference in percentage regarding dynamic stiffness for cast iron vented disk α type with reduced fins compared to one α type initial.

CONCLUSIONS

Their natural frequencies are a good indicator of the vibrational behavior of disks, including disk brakes. Analysis of calculation results show their own different natural vibration modes and different materials change their natural vibration modes for the same material, through the evolution of part geometry (where iron disk type α).

This allows consideration of eigenmodes as an important parameter in defining a brake disk in terms of sound.

Vibrational behavior of a new brake disc can be prefigured in an early stage of development.
To do this, we have to identify the dynamic stiffness methodology presented in this study, for a disk that was previously analyzed for noise -developed the coupling disk / plate and therefore was declared satisfactory. The new disk to be developed, either in terms of geometry or material, or both, is calculated to define its own frequencies, respectively, the dynamic stiffness. Comparison of dynamic stiffness will enable the new product compared to the reference product and anticipation vibrational behavior, respectively predisposition to produce noise when braking.
Once defined, valuable documents are enrolled for the new product, such as frequencies and allowed tolerances.

In the case of an existing brake disk, the determination by calculating its vibrational behavior is helpful in maintaining a good level of quality. As revealed in this study, the decrease of 2 mm thickness of the cooling fins vented disk brakes led to a change of their frequency and of the dynamic stiffness of the disk. It can be concluded that introducing a well-defined value of the frequency tolerance , it is possible to control the vibration behavior of the disk brake and to avoid the possibility of brake disks made by altering the parameters of the manufacturing process, to become " noisy " when braking.

Observations revealed in this study are of great interest by creating the possibility of accurate assessment of the contribution in defining predictability brake disc noise source on the friction coupling disc / pad.

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NOMENCLATURE

C damping matrix

K stiffness matrix
M mass matrix
u vector
$\dot{\mathbf{u}}$ vector of velocity

$\ddot{\mathbf{u}}$ vector of acceleration
Φ eigenvalue of vector
ω angular frequency
Rγ Frequency Response Function
γ acceleration