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STRUCTURAL AND MODAL ANALYSIS OF THE WHEEL BRAKE MECHANISM OF A CAR

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This research paper explores the structural and modal analysis of a disk-pad brake mechanism, utilizing the robust capabilities of ANSYS and Solid Works simulation software. The criticality of brake mechanisms in vehicular safety necessitates an in-depth study into their dynamic behaviors, particularly investigating the intrinsic modal characteristics that are pivotal for enhancing performance and safety. Leveraging ANSYS's finite element analysis (FEA) and Solid Works' versatile modeling and simulation environments, this study deciphers the nuanced vibrational modes, resonant frequencies, and associated deformation patterns of the brake mechanism under varied operational conditions. Detailed computational models were developed to elucidate the natural frequencies and mode shapes that are intrinsic to the system's operational integrity and reliability. The ensuing data illuminates potential areas for optimization, aiming toward minimizing detrimental vibrational effects, enhancing structural integrity, and ultimately contributing to the evolution of more safe and reliable brake mechanisms.

Keywords: Structural & modal analysis, Disk-pad brake mechanism, Vibrational modes, Resonant frequencies, ANSYS, SolidWorks, System optimization

1. Introduction

The modal analysis provides the natural frequencies at which a structure will resonate [1-3]. In many areas of engineering, these natural frequencies are essential. In the automotive industry, suspensions and brake mechanisms are usually adjusted to have the racing cars and passenger cars the different natural frequencies.

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The operation of brake mechanisms demonstrates the sliding frictional contact and uses complex solutions to prevent unsteady modes (noise, vibration, etc.).

The stick-slip phenomenon causes sustained and friction-induced oscillations between brake discs and brake pads [1]. The coefficient of static friction is higher than the coefficient of sliding friction and the coupling mode (the instability was mainly caused by the geometric parameters that were wrongly chosen) [4, 5].

The classical eigenvalue problem provides the basic equation that is solved in a typical undamped modal analysis for the ANSYS program [5].

$$[K] = \omega_i^2[M], \quad (1)$$

where: $[K]$ - stiffness matrix; i - shape vector (eigenvector) of mode i; ω_i - the natural frequency/pulsation of mode i (ω_i^2 - eigenvalue); $[M]$ - mass matrix between the tire and road surface, braking command and control system, driver physical effort.

To solve this equation, the method of Lanczos vectors in the ANSYS program will be used. In this case, the mechanical component of the brake disc-pad system studied in this paper is considered independent, without mechanical links. Therefore, in the effort to identify the possibility of the resonance phenomenon, a modal analysis of the mechanical component with boundary conditions that are real will be performed [6], 7. The equation that describes the dynamic motion of a system with a single degree of freedom under the influence of a dynamic force, $F(t)$, is as follows:

$$m\ddot{u} + c\dot{u} + ku = F(t), \quad (2)$$

where: m – the mass of the mechanical component (brake disc-pad system); u – displacement; $\dot{u}$ - initial speed at zero time; c – damping of the mechanical component; k – elasticity constant.

In the case of undamped free vibrations, the perturbing force is zero ($F(t)$ = 0) and the damping is zero ($c = 0$). So the equation of dynamic motion is:

$$m\ddot{u} + ku = 0. \quad (3)$$

When performing modal analysis on such structures, the pre-stressing effect must be considered because the state of stress changes the natural frequency of a structure and a static analysis is performed to find out the stress state of the structure [8, 9]. The pre-stress state due to the action of the dynamic force, F modifies the structural stiffness by adding a stiffness matrix, $[S]$ of the tension, σ_0, to the original structural stiffness. The eigenvalue solution is based on the new structural stiffness, according to the relation/equation [8]:

$$[K]\{u\} = \{F\} \underset{\text{A liniar static analysis is performed}}{\rightarrow} \sigma_0 \underset{\text{A stiffness matrix is calculated from the structural analysis}}{\rightarrow} [S] \tag{4}$$

Therefore, the solution is based on the new structural stiffness according to the equation or relation:

$$\begin{gathered}([K] - \omega_i^2[M])\{\Phi\}_i = \{0\} \\ \downarrow \\ ([K+S] - \omega_i^2[M])\{\Phi\}_i = \{0\},\end{gathered} \tag{5}$$

Original free vibration equation is augmented to include term [*S*] (stiffness matrix)

where the following notation was used:

$$\omega_i = \sqrt{k/m} \tag{6}$$

The natural period of vibration (T_i in seconds) is the extent to which a system with a single dynamic degree of freedom completes a complete cycle of un-damped free oscillations [10]. The relationship between period and natural circular frequency (also known as natural pulsation of vibration, in rad/s) is as follows:

$$T_i = 2\pi\omega_i. \tag{7}$$

The natural frequency of vibration, or f_i, is the total number of oscillations a system makes in one second (measured in Hz) and is calculated from the following relationships:

$$f_i = 1/T_i \text{ sau } f_i = 1/2\pi\omega_i. \tag{8}$$

Modal analysis of the brake disc-pad assembly can be performed by dynamic transient, harmonic, random vibration and response spectrum simulation, as schematically illustrated in Fig. 1.

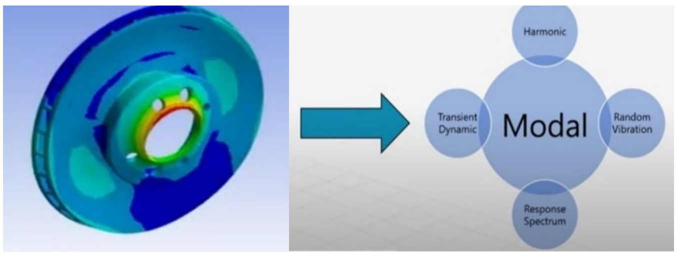


Fig. 1 Modes of modal analysis of the brake disc-pad assembly

2. Materials and methods

A simple brake disc-pad assembly model created with the 3D CAD program SolidWorks2022, Dassault Systèmes, USA is shown in Fig. 2. The disc made of gray cast iron is vented and has a total thickness of 22 mm, with the solid parts (right-left) 10 mm thick, and the brake pads 15 mm thick. The inner diameter of the disc is 61 mm, and the outer one is 260 mm. A prestressed modal analysis is performed on this model using different methods to determine the unstable modes.

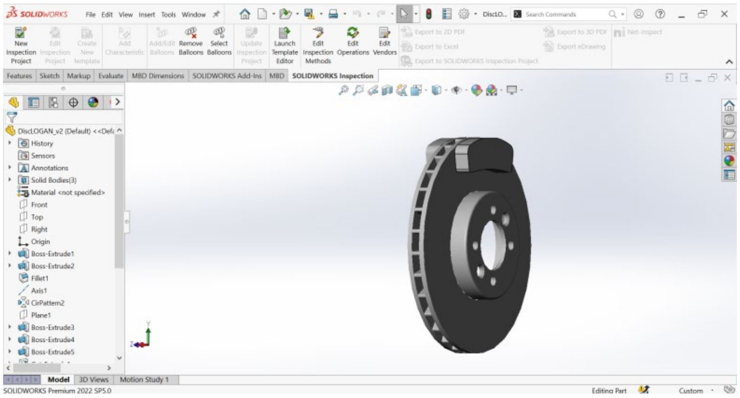


Fig. 2 Brake pads-disc assembly

Issues that occur in the brake disc-pad contact area usually require manual calculations of asymmetric terms from sources such as frictional slip and then introducing the asymmetric terms using special elements such as MATRIX27.

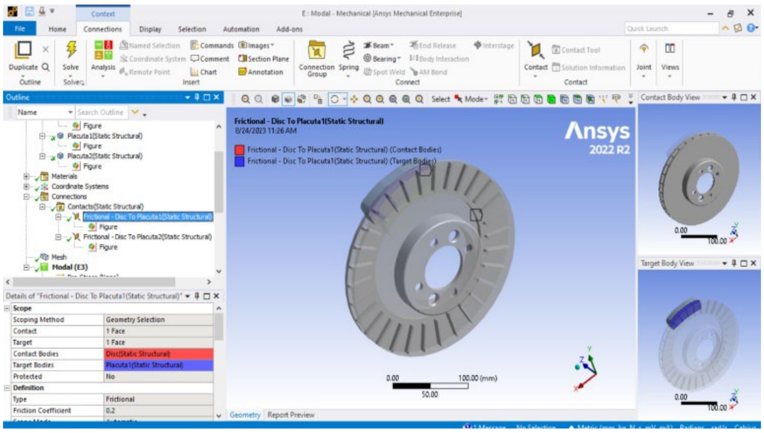
By modeling surface-to-surface contact at the pad-disc interface, 3D contact elements (CONTA17x) offer a more efficient alternative. At the target contact surface, no proper discretization grid is required and no calculations of the unsymmetrical terms are required [8].

The many control options for defining contact pairs offered by surface-to-surface contact elements include contact surface type, algorithm, contact stiffness, and gap/initial penetration impact.

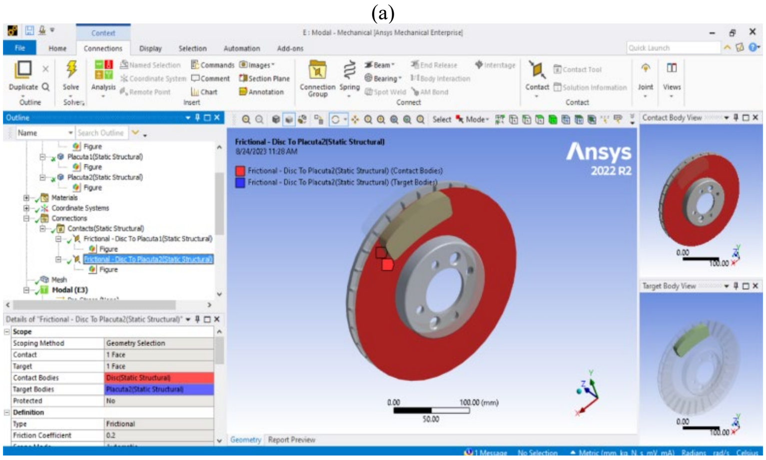
To replicate the frictional sliding contact that happens at the pad-disc interface, surface-to-surface frictional contact pairs with a friction coefficient of 0.3-0.5 are employed to characterize the contact between the brake pads and the disc.

Surface-to-surface contact pairs are utilized to define contact for other parts that will constantly be in contact during the braking process.

Fig. 3 (a) and (b) depict the contact and target pairings for frictional contact on either side of the disc.



(a)



(b)

Fig. 3 Target and contact pairs for friction between brake discs and brake pads: a) the interior face (facing the engine); b) the outside face (facing the wheel).

The enhanced Lagrange algorithm is employed for the friction contact pairs because, throughout the equilibrium iterations, the frictional pressure and stresses are increased and cause the penetration to gradually decrease. Additionally, this technique uses fewer computer resources than the conventional Lagrange multiplier algorithm, which typically needs more rounds to guarantee perfect contact compatibility. When simulating general frictional contact, such as the contact between brake discs and pads, the enhanced Lagrange method is a good choice.

The CMROTATE command, which specifies continuous rotation rates on the contact and target nodes, is used to create the internal sliding motion. The final solution is unaffected by the provided rotation speed, which is solely used to determine the sliding direction.

Brake discs, pads, and all other associated components are joined with 20-node SOLID186 structural solids and with uniform low integration element technology. O finer mesh is used at the disc-pad interface to accurately predict the unstable modes.

The brake disc-pad assembly is obtained with a total of 60351 nodes and 11473 elements. The mesh on the mapped faces is used to control the network on the disc-pad faces. For pads and bodies between discs, there are 8 divisions for edge sizing. As a result, Fig. 4 depicts the finished discretized mesh for the brake disc-brake pad system.

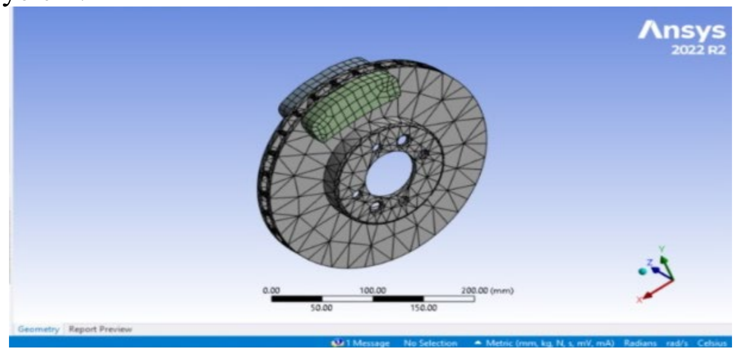


Fig. 4 The finished discretized mesh for the assembly of the brake disc and brake pads

After creating the discretization network, the model was loaded (Fig. 5), restricting the angular (rotational) speed of the disc, the force due to the pressure of the caliper, the moment at the wheel, the reaction due to the action of the wheels on the road track and the inner diameter of the disc hub. A small amount of pressure is applied to both ends of the pad to establish contact with the brake disc and to include preload effects.

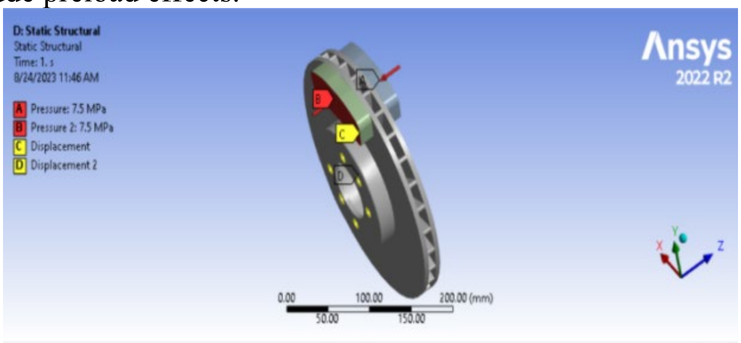


Fig. 5. Conditions at the displacement and loading boundaries

It is mentioned that, the sliding speed between the disc and the pads is variable, and at low and very low speeds, the stick-slip phenomenon appears, due to the elasticity of the brake mechanism elements, so it contributes to the frequency of its. The amplitude of this phenomenon is influenced by the stiffness characteristics of the brake mechanism, the working speed and the frictional behavior of the brake disc-pad material couple.

Naturally, the question of whether each natural frequency is equally significant in any analysis emerges. We may have thousands or millions of degrees of freedom (DOF) for a true physical model, which allows us to find an infinite number of natural frequencies. For example, if we consider the scenario of disc brake pads, where there are numerous DOFs, or frequencies and natural modes, this does not mean finding them all because they are not equally important.

The high frequency modes can typically be neglected. Additionally, not all modes contribute at the same level to the deformation of the structure under dynamic load. The mode participation factor and the effective mass are two straightforward scalars that we utilize to characterize the most significant frequencies or natural modes. As a result, the participation factor, γ_i, and effective mass, M_{ef} for module, i will be as follows:

$$\gamma_i = \{\Phi\}\{\Phi\}_i^T[M]\{D\}\,, \tag{9}$$

where: $\{\phi\}$ – mode shape; $[M]$ – mass matrix; $\{D\}$ – the direction of the excitation vector, and the effective mass of module, i is:

$$M_{ef,i} = \gamma_i^2, \tag{10}$$

that is, the effective mass, M_{ef} is equal to the square of the participation factor, γ_i. The effective mass, M_{ef} is also referred to as participation factor, γ_i, in some specialized writings. Participation factor and effective mass have similar roles in modal analysis.

The quantity of mass traveling in each direction for each mode is measured by the effective mass, M_{ef}, and the participation factor, of the mode. The vector $\{D\}$ denotes the direction in which the participation factor, is calculated. A high number in one direction denotes the presence of forces that will stimulate that mode.

So, the quantity of mass traveling in each direction for each mode is measured by the effective mass, M_{ef}, and the participation factor, of the mode. The vector $\{D\}$ denotes the direction in which the participation factor, is calculated. A large value in one direction indicates that the mode will be excited by forces in that direction.

A modal analysis is defined as "free" for a structure without boundary conditions and indicates that the structure is free of load and without boundary conditions. Also, the structure is free to have rigid motion in the 6 degrees of freedom in three-dimensional space.

The first six natural frequencies should theoretically be zero for a free modal analysis. Zero natural frequency denotes the ability of the structure to move rigidly in the absence of excitation. Since there is no boundary condition, this is to be expected. By using finite element software, for example, to solve the free modal analysis numerically, one may discover that the first eight natural frequencies are not exactly zero but should be values very close to zero.

3. Results and discussions

Due to the comparatively low preload, unstable mode predictions for the brake disc-pad assembly utilizing both techniques were very close. Unstable modes were anticipated to occur at 6474 Hz by the uncompressed linear modal solution and 6458 Hz by the completely nonlinear perturbed model. The shapes of the modes for the unstable modes (Figs. 6 and 7) suggest that the way to bend the pads and disc shows similar characteristics. Due to friction and the stick-slip phenomenon, these bending modes couple and produce noise (creaking).

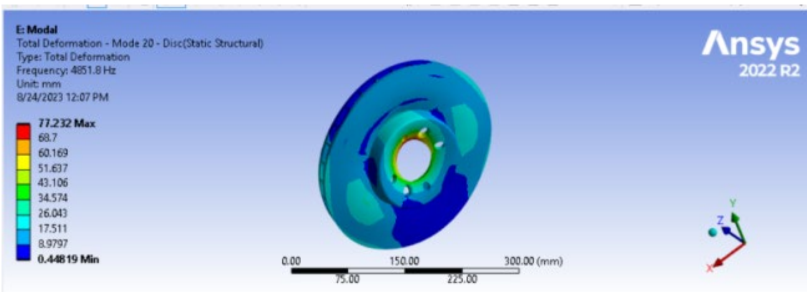


Fig. 6 Form of the modulo 20

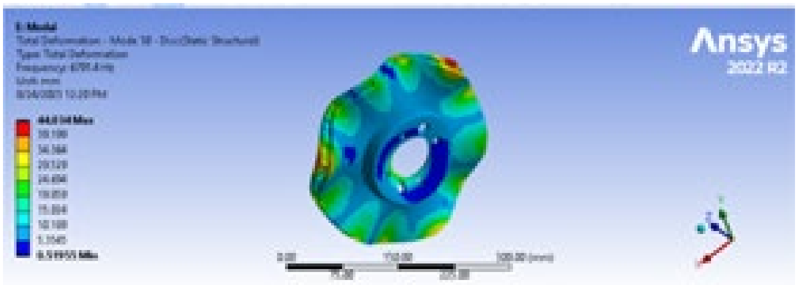


Fig. 7 Form of the modulo 30

a) Determining how each component will behave modally

When anticipating the brake noise, it is important to determine the modal behavior of the individual components (disc and pads). A modal analysis performed on the free disk model and free pads provides insight into potential coupling modes.

An examination of the results obtained from the modal analysis of a free disk and of a pad shows that the second bending mode of the pad and the ninth bending mode of the disk can couple to create dynamic instability in the system.

These bending modes can couple to create an intermediate lock, which generates noise at a frequency near 6470 Hz, a value determined by carrying out some preliminary virtual experiments (simulations with the ANSYS program) in which 30 vibration modes (of which two are presented in Figs. 6 and 7) were extracted, using a fully nonlinear disrupted model (which is not the subject of this paper).

b) Pre-stressed modal analysis

Structures are typically built to function under tension. The effect of prestressing must be taken into account when performing modal analysis on such structures because the state of stress alters a structure's natural frequency.

First, a static analysis is conducted to determine the structure's stress state. This analysis adjusts the structure's structural stiffness by adding a tension stiffness matrix (see equation/relation (4)) to the initial structural stiffness. According to relations/equations (4), the new structural stiffness serves as the foundation for the eigenvalue solution. By replicating the disk's rotation at the speeds shown in Table 1, the natural vibration frequencies of the disk were discovered [5, 11].

Along with the maximum speed required by the design, 5 driving speeds for the motor vehicle were selected as being the most common. The mechanical structure reaches resonance at 6 frequencies in each of the cases examined, with the first being crucial in determining the effectiveness of the structural integrity of the disk (see Table 1). It can be observed (from Table 1), that in the range of angular speeds analyzed of the disk, the resonance frequencies are above the value of 400 Hz (see and Table 2) which is the equivalent of more than 1650 km/h of the vehicle, the speed that cannot be reached by passenger cars.

Table 1

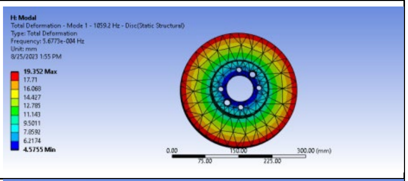
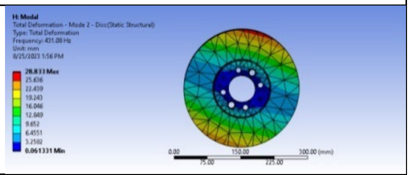
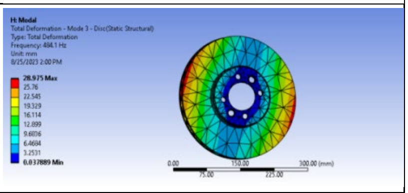
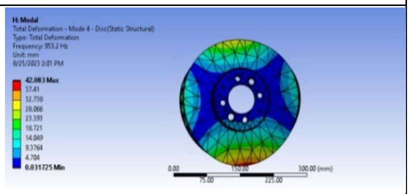
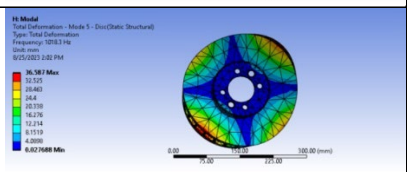
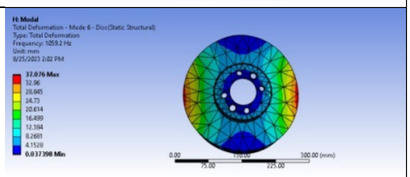
Scenarios analyzed at different vehicle speeds

Analyzed scenario	Angular velocity (1/s)	Travel speed (km/h)	Resonance frequency (Hz)
1	60.38	50	5.68E-04
2	96.60	80	431.08
3	150.9	125	484.1
4	217.4	180	953.2
5	265.7	220	1018.3

The disc can deform up to 4.7 mm from its original shape if resonance occurs, depending on the resonance frequency (maximum values are found for resonance modes 4, 5, and 6; see Table 2), at which point the part's maximum stress is 509.65 MPa, as shown in Fig. 8.

Table 2

Total deformation for different resonant frequencies

No. of vibration mode case	Resonance frequency (Hz)	The total deformation
1	5.68E-04	
2	431.08	
3	484.1	
4	953.2	
5	1018.3	
6	1059.2	

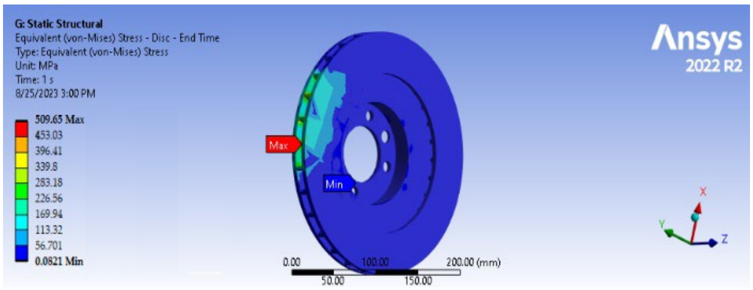


Fig. 8 Disc stress state, vibration mode 2, scenario 2

Given that the material's breaking strength is 325 MPa, the disc won't break in this scenario. The two contacting surfaces (disc-pads of brake), in a frictional contact can support shear stresses, up to a particular magnitude at their interface, before begin to slide relative to each other.

The equivalent shear stress, τ_{fr} in the Coulomb friction model is defined as a percentage of the contact pressure, p, at which sliding on the surface starts to occur, that is:

$$\tau_{fr} = \mu p + c_{coh}, \tag{11}$$

where: μ is the friction coefficient and c_{coh} is the coefficient defining the sliding resistance of cohesion.

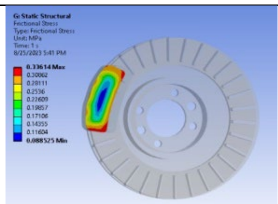
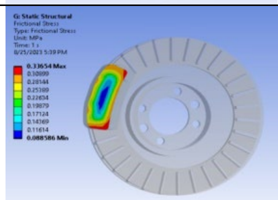
The two surfaces will glide against one another after the shear stress is overcome. This state lasts very little in time, is repeated periodically and is known as jerky slippage (stick-slip). Sticking-sliding (the stick-slip phenomenon) calculations determine when a point of the contact surfaces goes from sticking to sliding or vice versa.

Additionally, a rise in the contact pressure and/or relative speed between the contact surfaces implies an increase in the amplitude of the stick-slip phenomenon. Apart from the Coulomb friction model, one can also define his own friction model. However, "There is a demand to add frictional stress"; frictional stress is an output quantity that cannot be added, however there is a provision to add the maximum contact friction with stress units (max).

Regardless of the normal pressure, if the frictional stress surpasses this threshold, bodies start to slide. Tables 3 and 4 display the frictional stress for various COF values and travel speeds.

Table 3

Frictional tension at different vehicle speeds

Analyzed scenario	Angular velocity (rad/s)	Value of stress due to friction (MPa)	Stress due to friction
1	60.38	0.3361	
2	96.60	0.3365	

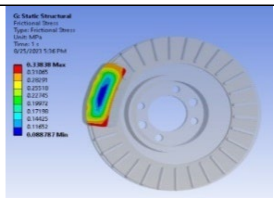
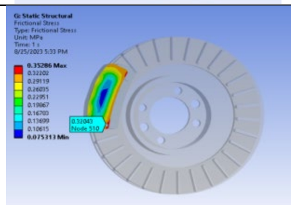
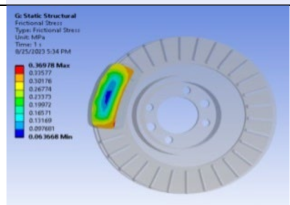
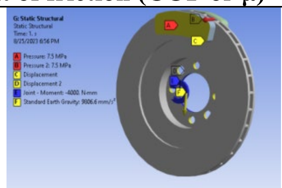
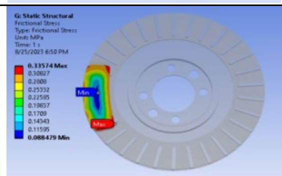
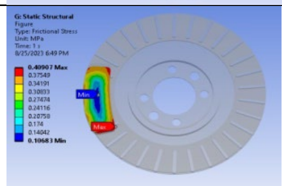
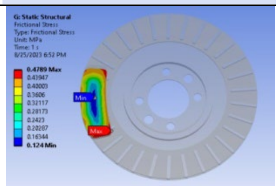
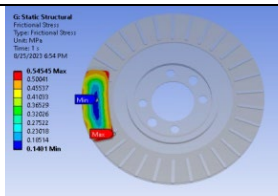
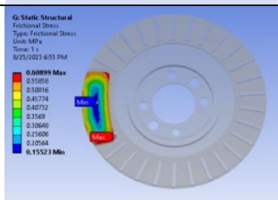
3	150.9	0.3383	
4	217.4	0.3528	
5	265.7	0.3697	

Table 4

Frictional stress at different values of the coefficient of friction (COF or μ)

Analyzed scenario	**Friction coefficient of (COF or μ)**	**Stress due to friction (MPa)**	
1	0.2	0.3357	
2	0.25	0.409	
3	0.3	0.4789	

4	0.35	0.5454	
5	0.4	0.6089	

4. Conclusions

The simulations' results confirm the possibility of introducing the modal analysis in the conception and manufacturing cycle, respectively in the development of experimental research related to the mode of vibration of the brake disc-pads couple as well as their optimization from this perspective.

From the estimated values obtained from the simulations, it emerges that the speed of rotation of the disc at which it vibrates at the resonance frequency (equivalent to more than 1650 km/h) is much higher than the speed at which it can reach in operation. This fact indicates a good structural integrity of the chosen disk.

The simulation program was created in the ANSYS Workbench environment to investigate the structural properties possible and the tribological behavior of a car's brake mechanism.

To build a brake mechanism with a more effective disc and pad structure, reference values for the structural choice of the materials employed are provided by the findings of the modal analysis method, to which it can also be added the finite element analysis method.

Additionally, the outcomes of the numerical research demonstrated that a rise in the contact pressure and/or relative speed between the contact surfaces implies an increase in the amplitude of the stick-slip phenomenon.

Using tools like ANSYS Workbench has been shown to significantly increase work efficiency and decrease design time, and brake disc wear is outpaced by brake pads wear several times.

Significant benefits of using the ANSYS Workbench R16 software in the research process have been shown. Not only did it shorten the duration of the planning process, but it also provided a reliable platform for the experimental validation of the methodologies used in these tests.

More, it has highlighted a notable concordance between theoretical data and experimental results, validating the reliability and accuracy of the data obtained. This demonstrates how the use of some cutting-edge technologies can significantly increase accuracy and efficiency in the development of more reliable and secure navigation systems.

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