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TOPIC:
A STUDY ON INFLUENCE OF MATERIAL AND SHAPE ON VIBRATION AND NOISE OF AUTOMOBILE BRAKE DISCS UNDER DIFFERENT OPERATING MODES
***
(NGHIÊN CỨU ẢNH HƯỞNG CỦA VẬT LIỆU, HÌNH DẠNG ĐẾN DAO ĐỘNG VÀ TIẾNG ỒN TRÊN ĐĨA PHANH Ô TÔ Ở CÁC CHẾ ĐỘ VẬN HÀNH KHÁC NHAU)

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ABSTRACT

Topic: Study on the Influence of Material and Geometry on Vibration and Noise of Brake Discs under Various Vehicle Operating Conditions

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In this study, we aim to investigate, evaluate, and discuss the effects of brake disc geometry and material selection specifically CastIron, ALSiC20%, and AISI1650 on vibration and noise characteristics. The object of this research is an automotive disc brake system operating under two typical conditions: at vehicle speeds of 70 km/h and 120 km/h.

The analysis is conducted using the HyperWorks software suite to perform simulation-based calculations. The research results will highlight the influence of material properties on vibration and noise behavior, as well as the force transmission patterns observed at various points on the disc surface.

FREFACE

In the context of the automotive industry placing increasing emphasis on user comfort and safety, factors related to noise, vibration, and harshness (NVH) have become particularly significant. The braking system especially the brake disc not only plays a critical role in ensuring vehicle safety but is also one of the most prominent sources of noise and vibration during operation. Stemming from this practical issue, the thesis titled "Study on the Influence of Material and Geometry on the Vibration and Noise of Brake Discs under Different Operating Conditions" was carried out with the goal of comprehensively evaluating the effects of design parameters and material properties on the NVH characteristics of brake discs, and proposing optimization strategies accordingly.

This research was conducted based on a solid theoretical foundation and supported by modern computational tools namely, the finite element method (FEM), implemented through the HyperWorks/OptiStruct software suite. Throughout the course of the project, I received dedicated, insightful, and responsible guidance from Assoc. Prof. Dr. Le Minh Duc. His invaluable support, both in terms of technical expertise and research direction, has been a great motivation for me to successfully complete this thesis.

I would like to express my sincere gratitude to him for accompanying and supporting me throughout the process of learning and research.

CHAPTER 1. OVERVIEW OF STUDY

1.1. Introduction

Currently, the global automotive industry is undergoing remarkable growth in both scale and technology. Trends such as autonomous vehicles and fuel efficiency optimization are reshaping how cars are designed and manufactured. Along with these advances, consumer demands are also increasing not only in terms of performance but also in comfort, smoothness, and durability during use.

The braking system in automobiles is a critical mechanism that ensures operational safety by decelerating or stopping the vehicle through the conversion of kinetic energy into thermal energy via friction between the brake disc and brake pads. Although this process occurs over a short duration, it is subjected to intense thermo-mechanical loads under harsh working conditions, resulting in rapid heat generation and high-amplitude structural vibrations. These factors play a key role in causing negative phenomena such as abnormal brake noise, structural vibrations, reduced braking efficiency, and even fatigue-related failures over the component's lifespan [1], [2].

In real-world driving conditions particularly during emergency braking, long downhill descents, or continuous braking at high speeds the brake disc is not only affected by thermal loads and frictional forces but also influenced by external vibrations transmitted from the suspension system, vehicle body, and internal drivetrain oscillations. These factors can induce resonance, leading to brake squeal, rattling sounds, and acoustic phenomena in the frequency range from 1 to 16 kHz [3]. Recent studies, both numerical and experimental, have shown that brake disc material and geometry (including the number of ventilation holes, grooves, disc thickness, and structural design) significantly impact the vibration characteristics and acoustic behavior of the braking system [4], [5], [6].

Traditionally, gray castiron has been the standard material for manufacturing brake discs due to its excellent heat resistance, thermal stability, and high damping capacity, which help mitigate high-frequency vibrations. However, castiron has a major drawback: high density, which negatively affects dynamic responsiveness and fuel efficiency. In the context of modern automotive engineering that aims to optimize weight, performance, and emissions, replacing conventional materials with advanced lightweight alternatives while maintaining braking effectiveness has become an inevitable trend. Materials such as aluminum matrix composites, titanium alloys, and high-strength alloy steels have shown great promise due to their superior thermo-mechanical properties, high thermal conductivity, durability, and lower weight [7], [8], [9].

In addition to material considerations, the geometric structure of the brake disc plays a decisive role in both heat dissipation and vibration behavior. The inclusion of ventilation holes, gas escape grooves, or radial drillings not only increases the surface area for air contact thereby improving cooling efficiency but also reduces the disc's weight, which contributes to better suspension response. However, these geometric changes can significantly alter the natural frequencies and mode shapes of the disc, potentially affecting noise generation and system stability [6], [10].

One of the key challenges facing brake system designers today is achieving a balance between heat dissipation, vibration reduction, noise control, and braking performance throughout the component's service life. In this context, digital simulation tools and finite element analysis (FEA) have become indispensable for evaluating the thermal, vibrational, and acoustic responses of braking systems in a detailed and efficient manner without relying solely on costly and difficult-to-control physical testing [4], [11].

Based on this practical foundation, the present study focuses on investigating the influence of three typical materials used in modern automotive brake discs (gray castiron, aluminum matrix composite AlSiC20%, and AISI6150 alloy steel) combined with two brake disc geometries (a solid disc and a ventilated disc) on vibration and noise characteristics under various operating conditions. Modeling and simulation are conducted using HyperWorks, one of the leading CAE toolsets for modal analysis, vibration simulation, and system stability evaluation. The analyses include determining natural frequencies (modal analysis), simulating forced vibrations, and assessing resonance behavior as well as noise generation risks in the braking system. The results are expected to provide a valuable reference for selecting optimal brake disc materials and geometries to support the development of high-performance, low-noise, and reliable modern automotive braking systems.

1.2. Background

This study was conducted based on the integrated application of knowledge acquired from specialized coursework throughout the academic program. Specifically, the Finite Element Method (FEM) was employed as the primary simulation and analysis tool, grounded in the theoretical foundation provided by the Advanced Engineering Mathematics II course. Additionally, phenomena such as vibration, resonance, and noise generation in mechanical systems particularly in automotive braking systems were analyzed following the principles learned in the Noise and Vibration (NVH) course.

The combination of theoretical knowledge and numerical simulation enables an in-depth and quantitative evaluation of the influence of physical and design parameters on the NVH performance of brake discs. This serves as a foundation for modeling, developing dynamic analysis problems, and identifying instability mechanisms that lead to noise, thereby allowing the proposal of effective improvement strategies.

Introduction about Finite Element Method.

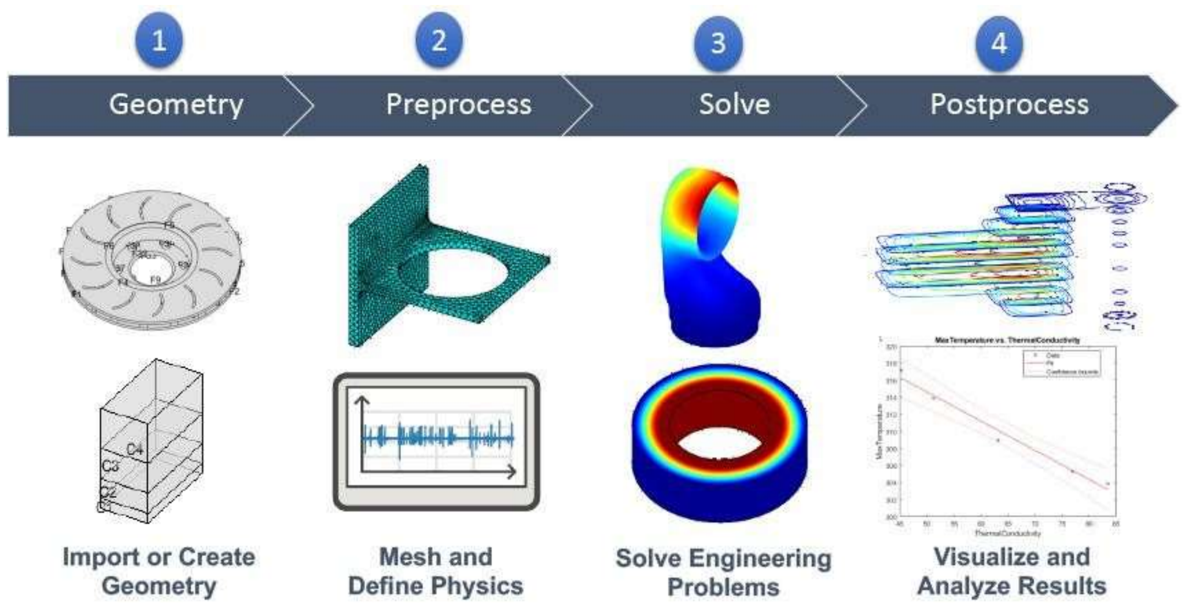


Figure 1.1. What is Finite Element Method.

In the context of rapid advancements in modern science and technology, engineering disciplines are increasingly confronted with more complex problems, while simultaneously facing demands to shorten design cycles and reduce production costs without compromising accuracy. To meet these challenges, the Finite Element Method (FEM) has emerged as a powerful and efficient numerical tool, widely applied in solving a broad spectrum of engineering problems across various fields.

FEM operates based on the principle of subdividing a complex system into numerous smaller elements called finite elements each corresponding to a computational domain with a simpler structure. As a result, the global problem is transformed into a system of algebraic equations that can be solved using numerical algorithms. This method enables accurate simulation and analysis of physical phenomena such as structural loading, stress and strain distribution, dynamics, heat transfer, fluid flow, electromagnetic fields, and even applications in biomedical engineering and biotechnology.

The development of Computer Aided Engineering (CAE) tools has significantly enhanced FEM, enabling engineers and designers to build complex models and conduct high-accuracy simulations in a shorter time. Today, many well-known FEM software packages are widely used in industry and academia, including ANSYS, HyperWorks, ABAQUS, SAP2000, among others. Effective use of these tools or the development of custom numerical programs requires a deep understanding of fundamental principles, modeling techniques, and the core solution algorithms of the finite element method.

Thus, FEM is not merely a mathematical tool but a foundational methodology that empowers engineers to analyze, solve, and optimize real-world engineering designs, thereby driving the advancement of modern engineering and technology.

Introduction about Noise and Vibration.

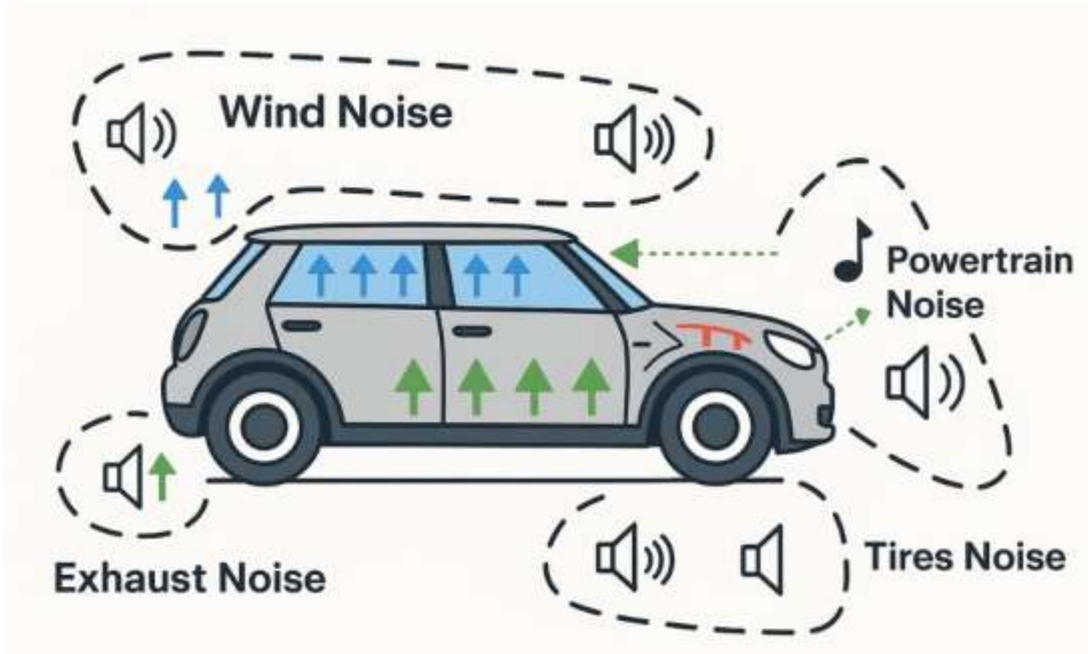


Figure 1.2. Source Noise and Vibration effect to car.

NVH (Noise, Vibration, and Harshness) is a field concerned with the measurement and control of noise and vibration characteristics in vehicles, particularly automobiles. Noise and vibration can be quantitatively measured, while harshness is a more subjective assessment without a standardized measurement scale. Although there is a psychological metric referred to as the harshness level, it often lacks strong correlation with many aspects of perceived discomfort.

Interior NVH refers to the noise and vibration experienced by occupants inside the vehicle cabin. In contrast, exterior NVH refers to the noise generated by the vehicle and perceived from the outside, often considered a primary source. Noise that propagates through the air as a result of turbulent collisions of air particles is classified as airborne noise. Meanwhile, noise that originates from vibrating structural surfaces is known as structure-borne noise.

In this context, noise is defined as any undesirable sound perceived by vehicle occupants or nearby individuals, typically causing discomfort or disturbance.

Several major sources of noise in vehicles.

- Engine Noise and Vibration: Originating from the combustion process inside the cylinder, as well as vibrations caused by rotating components such as the crankshaft and camshaft, and mechanical impacts between moving parts. This is one of the primary sources of vehicle noise, especially noticeable during acceleration or high-load operation.
- Exhaust Noise: Generated as exhaust gases move at high velocities through the exhaust system. If the muffler or silencer system is ineffective, the emitted noise can be loud and unpleasant.
- Noise and Vibration from the Drivetrain: Includes components such as the transmission, driveshaft, and differential. These generate noise due to mechanical vibrations, gear friction, and resonance within the drivetrain system.
- Tire-Road Interaction Noise: When the vehicle is in motion, friction between the tires and the road surface produces noise, particularly on rough surfaces or at high speeds. Uneven tire wear or imbalanced tires can also increase vibration levels.
- Brake System Noise and Vibration: Particularly brake squeal, which occurs when the brake disc and pads vibrate at resonant frequencies, producing a high-pitched and clearly perceptible noise.

Various approaches for reducing vibration and noise in automotive systems.

- Reducing Engine Noise: Optimize the combustion process to minimize knocking and vibrations, use a damping flywheel and balance shafts, improve machining precision, reduce clearances in rotating parts, design a sealed engine cover, and apply better sound insulation.
- Reducing Noise from the Vehicle Body: Design body panels with ribs and corrugations to increase local stiffness and reduce vibration; use elastically mounted engine brackets to isolate vibrations; reinforce weak areas of the body using high-stiffness materials or stiffening plates.
- Using Sound Insulating and Absorbing Materials: Apply acoustic insulation in the passenger cabin, roof, doors, and engine compartment partition; use

vibration-damping liners on the floor and other large surface areas of the vehicle body.

- Increasing Structural Stiffness of Vibration-Prone Components: Use high-stiffness materials for components susceptible to resonance such as brake discs, door panels, and suspension systems; optimize the natural frequencies of systems to avoid matching excitation frequencies from the engine or road surface.

1.3. Objectives

The objective of this study is to comprehensively investigate and analyze the effects of key factors such as braking force, disc material, geometry, and structural configuration on the vibration and noise phenomena generated during braking, particularly in disc brake systems. The study will be conducted under various operating conditions to reflect real-world scenarios that may occur during vehicle operation.

By integrating numerical simulation methods, the research aims to identify the underlying mechanisms that lead to instability and noise generation in braking systems. Based on these analyses, the study will provide technical recommendations regarding disc geometry design, material selection, and optimal operating conditions to effectively minimize unwanted noise and vibrations in disc brakes.

The outcomes of this research are expected to contribute to the improvement of NVH (Noise, Vibration, Harshness) quality in braking systems and serve as a scientific foundation for the development of quieter, more stable, and more durable brake designs in the automotive industry.

1.4. Objective and Methodology

1.4.1. Objective

Brakes are a critical component of an automobile's braking system, playing an essential role in ensuring operational safety. The primary function of the braking system is to reduce the vehicle's speed to a desired level or bring it to a complete stop within the shortest possible time and distance, while also maintaining the vehicle's stationary position on various terrains. Without a braking system or with a malfunctioning one, controlling speed and stopping the vehicle would become hazardous, posing significant risks to both the driver and surrounding vehicles and pedestrians.

Among the various types of braking systems used in modern transportation, especially in automobiles, the most common include disc brakes and drum brakes. Within the scope of this study, the focus will be specifically on exploring NVH phenomena associated with disc brakes (shown in Fig. 1.3).

Figure 1.3. Disc Brake on car.

Disc brakes operate based on the principle of friction. When the driver applies force to the brake pedal, the hydraulic system transmits pressure to the pistons, which in turn push the brake pads against both sides of the rotating brake disc (rotor) attached to the wheel. The friction generated between the brake pads and the disc reduces the wheel's rotational speed, thereby slowing down or stopping the vehicle. However, this process is also a major source of NVH issues, particularly the noise generated when the brake pads contact the disc at high speeds and under strong braking force. Squealing or groaning noises caused by disc brakes not only reduce driving comfort but may also indicate unwanted structural vibrations, which can negatively affect both user experience and the service life of the brake disc.

Disc brake noise is a form of structure-borne noise, originating from mechanical vibrations that propagate through the material. Factors influencing this phenomenon include the geometric design of the brake disc, material properties (such as stiffness, density, damping ratio), and operating conditions. To improve NVH performance, possible measures include altering the disc design (such as drilling holes, adding ventilation grooves, honeycomb structures, or vented discs) or changing the disc material.

In this study, the object of analysis is a common automotive brake disc. The research focuses on investigating the effects of geometric and material factors on the vibration and noise characteristics of the brake disc.

1.4.2. Methodology

In this study, the Finite Element Method (FEM) is chosen as the primary tool for simulating, analyzing, and evaluating NVH phenomena in automotive brake discs. FEM is a powerful and widely used numerical technique in engineering mechanics, particularly well-suited for analyzing nonlinear dynamic problems with complex geometries and diverse boundary conditions. This method allows the research object to be discretized into smaller elements, each with specifically defined mechanical and physical properties. By constructing a physical model and solving the system of motion equations, FEM enables the determination of the dynamic response of the brake disc under excitation loads, thereby providing quantitative data for evaluating vibration and noise behavior.

The NVH analysis of the brake disc is carried out through two main types of problems:

- Normal Modes Analysis: This analysis aims to identify the natural frequencies and corresponding vibration modes of the brake disc, helping to evaluate the modes that may lead to resonance, noise generation, or structural damage.
- Frequency Response Analysis: This analysis determines the brake disc's response when subjected to harmonic loads with specific amplitudes and frequencies, which are common operating conditions in real-world scenarios.

The specific implementation steps are outlined as follows.

Step 1. Geometric Modeling and Data Preprocessing.

The 3D model of the brake disc was designed using CAD software (Autodesk Inventor) based on actual measured data.

After the design was completed, the FEM simulation software suite (Altair HyperWorks/OptiStruct) was used to set up and perform algorithmic calculations for the study. The meshing process was carried out with high-quality elements to ensure the accuracy of the simulation results.

The element size was selected appropriately to match the geometry and the objective of frequency analysis. Material properties such as Young’s modulus, Poisson’s ratio, and density were assigned according to each case study in order to evaluate the influence of material on NVH behavior.

Step 2. Normal Modes Analysis.

Modal analysis involves calculating the natural mode shapes and natural frequencies of a given system. This process does not necessarily provide the full time-dependent response of the system to a specific excitation. The natural frequencies of a system depend solely on the stiffness and mass distribution of the structure, as well as boundary conditions. This is examined based on the fundamental equations of motion [11].

$$Mg + C_x v + Kx = F \quad [1.1]$$

Where:

M is the mass matrix

K is the stiffness matrix

g is the acceleration due to gravity

x is the displacement

C_x is the damping matrix

F is the external force

v is the velocity

Since the external force is zero, the damping force and velocity will also be zero. Consequently, the equation of free vibration can be expressed as:

$$Mg + Kx = 0 \quad [1.2]$$

Step 3. Frequency response analysis.

Frequency response analysis is used to calculate the structural response under harmonic loading. Typical applications include the analysis of noise, vibration, and harshness (NVH) in vehicles, rotating machinery, and power transmission systems.

The loads can be applied as forces or forced motions (including displacement, velocity, and acceleration). These loads depend on the excitation frequency, and the problem is solved through the basic equations of motion [11].

$$Mg + C_x v + Kx = F \quad [1.3]$$

Step 4. Results Evaluation and Analysis.

The results from the two analyses above are synthesized and compared to evaluate the overall NVH characteristics of the brake disc under different conditions:

- Comparing the influence of each material on vibration and noise for the original (solid) brake disc.
- Comparing the influence of each material on vibration and noise for the ventilated brake disc design.
- Comparing the effect of different velocities (braking forces) on vibration and noise of the brake disc.
- Comparing the displacement of key points across different regions on each brake disc type.
- Comparing vibration and noise between the original brake disc and the ventilated disc design.

Based on these comparisons, optimal solutions are proposed to reduce NVH, either through design modifications or by selecting more suitable materials.

CHAPTER 2. DESIGN OF DISC BRAKE MODEL

2.1. Structure and Operating Principle of the Brake System

The initial part of the study focuses on thoroughly investigating the structure and working principles of the disc brake system commonly used on the 12-seat Toyota Hiace vehicle. This process involves examining key components of the braking system such as the brake disc, brake pads, caliper assembly, pistons, brake fluid lines, etc., to understand their relationship to the noise generated during braking (shown in Fig. 2.1).

Simultaneously, after identifying the primary noise contributors namely the brake disc and brake pads actual measurements of geometric parameters and dimensions were conducted. These include the outer and inner diameters, disc thickness, contact area with the brake pads, surface curvature, as well as the geometric dimensions of the brake pads such as length, width, and friction layer thickness. Accurate measurement of these parameters is essential for constructing the numerical model, enabling more precise and effective analysis and evaluation.

Figure 2.1. Disc Brake on Toyota Hiace.

The disc brake model used in this study is a solid disc type, directly sampled from the front brake system of a 12-seat Toyota Hiace vehicle. This type of brake disc features a flat friction surface. However, besides the solid disc, there are several other common design variants of brake discs, including:

- Drilled disc: Features holes penetrating the friction surface to enhance heat dissipation, reduce weight, and remove dust and gases generated during braking (Fig. 2.2).
- Slotted disc: The disc surface is machined with radial or spiral slots that help clean the brake pad surface (Fig. 2.2).
- Ventilated disc: Comprises two disc surfaces connected by internal cooling fins, allowing better airflow and improved cooling (Fig. 2.3).

Differences in the geometric design of brake discs not only affect braking performance and material durability but also significantly influence the vibration and noise characteristics of the braking system. Therefore, comparing these design variants is an important research direction in the NVH analysis of disc brake systems.

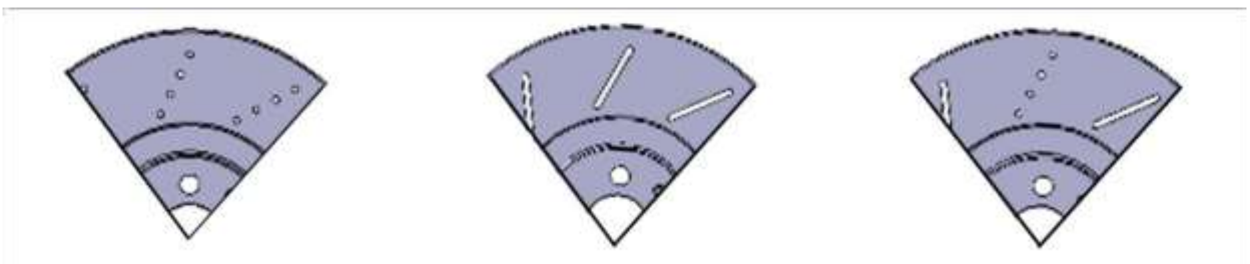


Figure 2.2. Patterns of holes and slits.

Figure 2.3. Ventilation Blade patterns.

To evaluate and compare the vibration performance between the drilled disc brake and the solid disc brake, a new drilled disc brake model was created based on the initially surveyed solid disc model. The design, including the diameter, position, and number of holes, was developed following [6].

2.2. Disc Brake Modeling

Using Inventor software, the 3D model was designed based on the actual model with parameters such as outer diameter, inner diameter, disc thickness, contact area with the brake pad, surface curvature, as well as the geometric dimensions of the brake pad including length, width, and thickness of the friction layer. Combining with the existing study on drilled brake discs, the resulting set of brake disc models is as follows Figs. 2.4- 2.9.

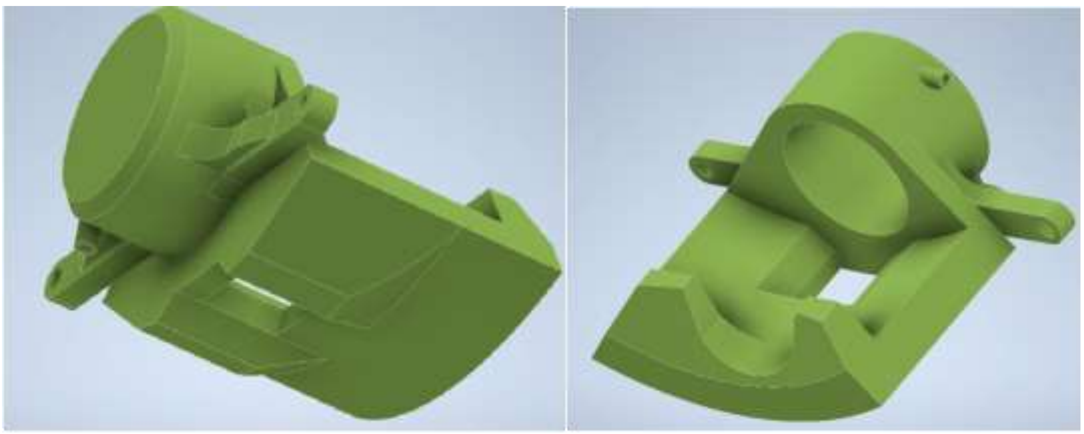


Figure 2.4. Brake Caliper.

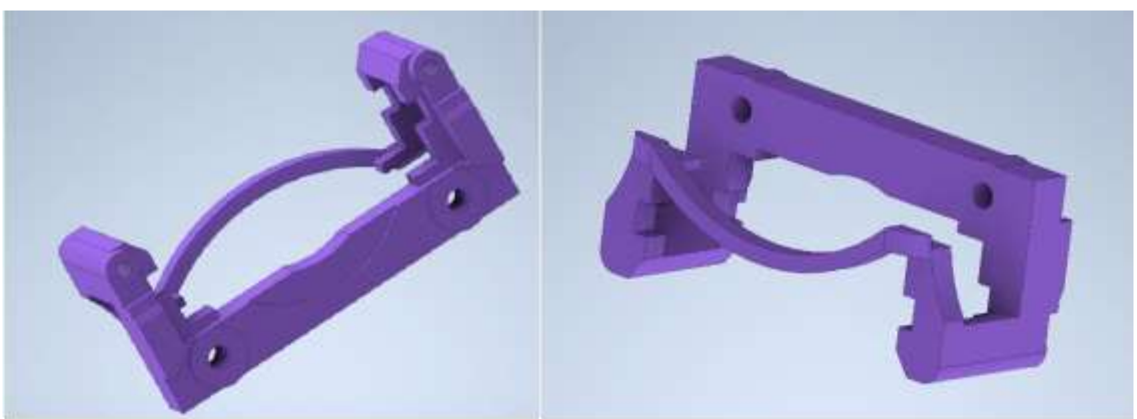


Figure 2.5. Brake Caliper Bracket.

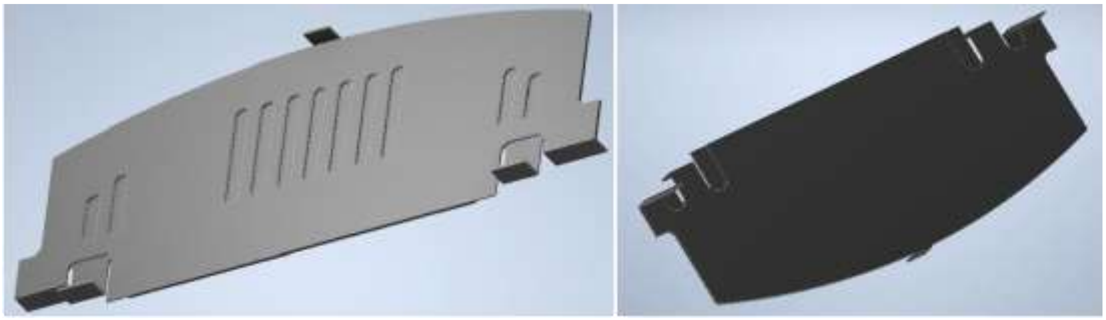


Figure 2.6. Brake Pad Clip.

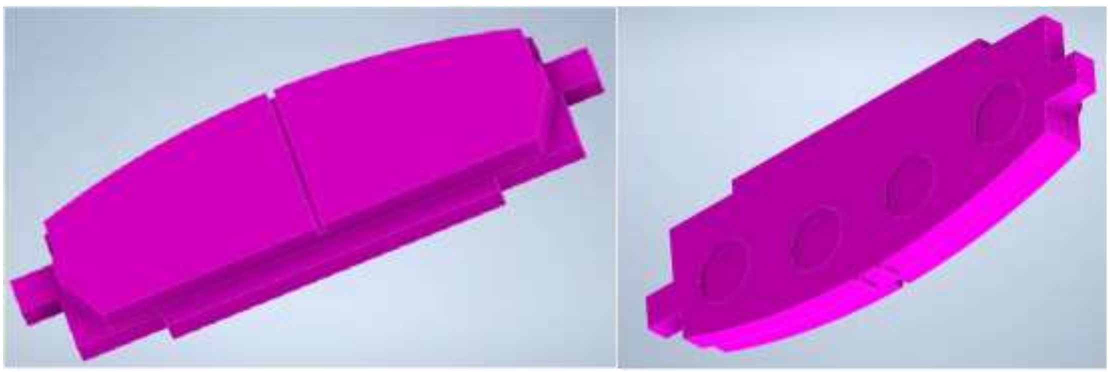


Figure 2.7. 3D model of the Brake Pad.

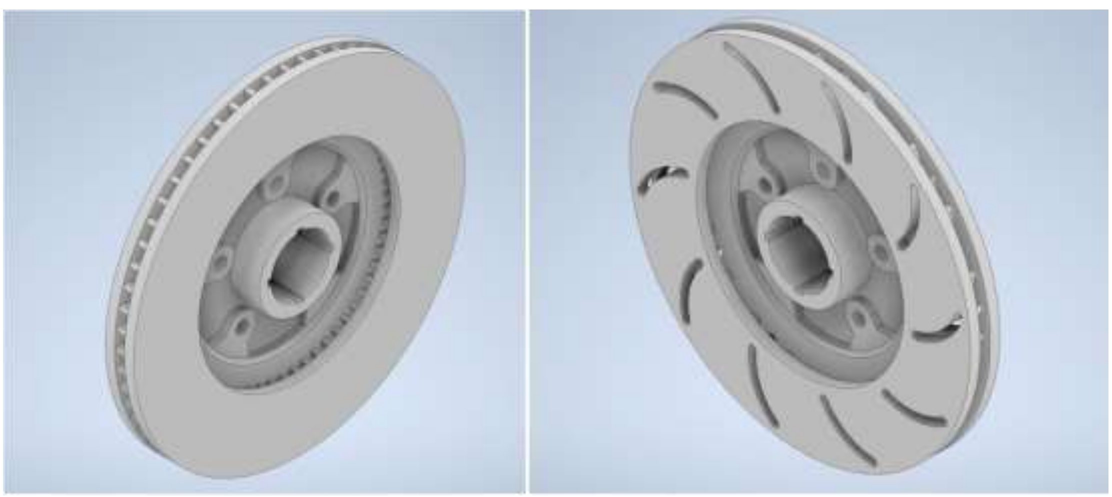


Figure 2.8. Original Brake Disc model and new Ventilated Brake Disc be designed.

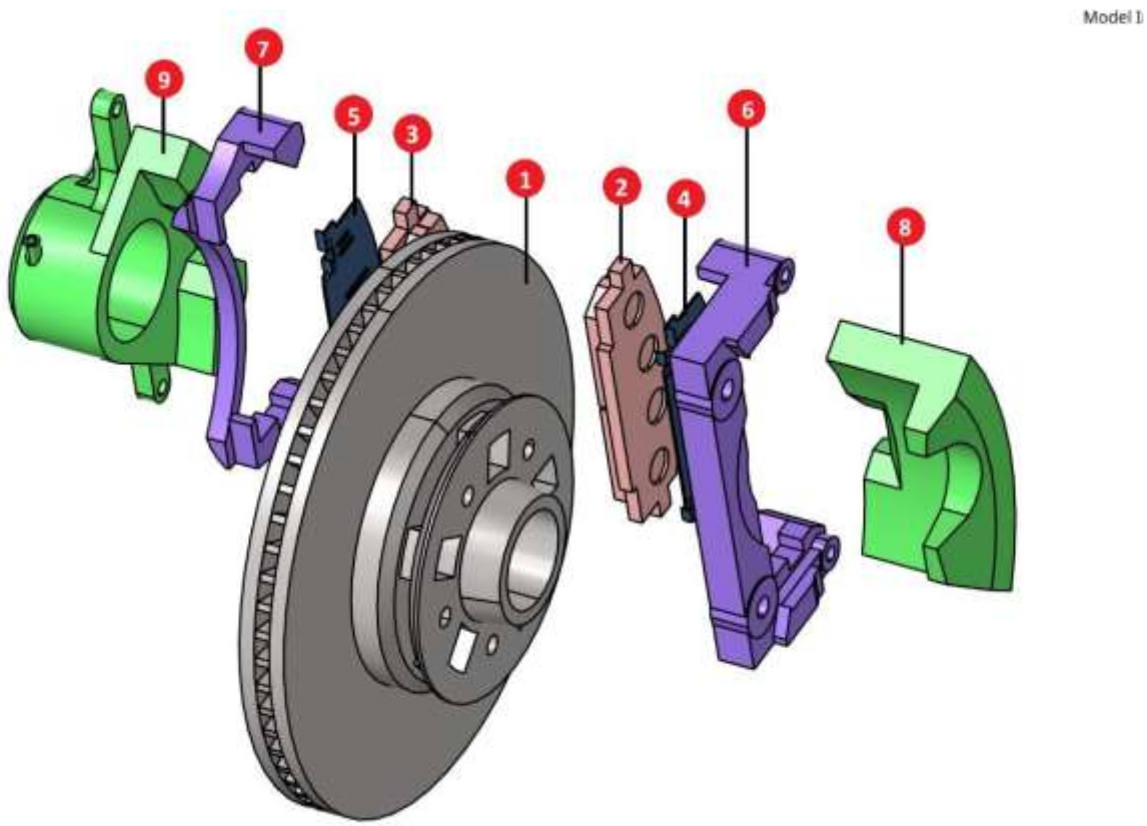


Figure 2.9. Brake Assembly Model.

1. Brake Disc/2,3. Brake Pad/ 4,5. Brake Pad Clip.

6,7. Brake Caliper Bracket/ 8,9. Brake Caliper.

After completing the construction of the standard models, two main analyses in this study were carried out: Normal Modes Analysis and Frequency Response Analysis.

2.3. Simulation with HyperWorks and Definition of Boundary Conditions

2.3.1. Introduction to the Simulation Tool

Figure 2.10. Introduction Hyperworks 2021 software and solver Optistruct.

HyperWorks 2021 is one of the advanced and powerful CAE software tools widely applied in various engineering fields due to its capability for accurate analysis based on the Finite Element Method (FEM). This system comprehensively supports engineering computations, physical simulations, and mechanical design optimization from detailed components to entire systems aiming to achieve engineering performance goals, optimize manufacturing costs, shorten product development cycles, and enhance operational reliability (Fig. 2.10).

Developed on a comprehensive optimization architecture, HyperWorks not only improves design data management efficiency but also allows flexible automation of simulation workflows. It is a powerful simulation solution enabling industrial organizations to make fast and precise design decisions. With tight integration with leading analysis, modeling, and optimization tools in the industry, HyperWorks provides an intuitive working environment, effective reporting, and technical data monitoring support. Adhering to the “open architecture” philosophy, HyperWorks continues to play a pioneering role in the simulation industry with high compatibility with other commercial CAD/CAE platforms. The HyperWorks 2021 version marks a significant milestone, representing a substantial upgrade in Altair’s CAE software ecosystem.

The solver used is OptiStruct: a powerful solver within HyperWorks employed for structural analyses such as linear static and nonlinear analysis, modal analysis, structural optimization, and dynamic response analysis. In this study, the focus is on two main problems: modal (natural frequency) analysis and dynamic response analysis of the brake disc.

2.3.2. Setting Up Boundary Conditions

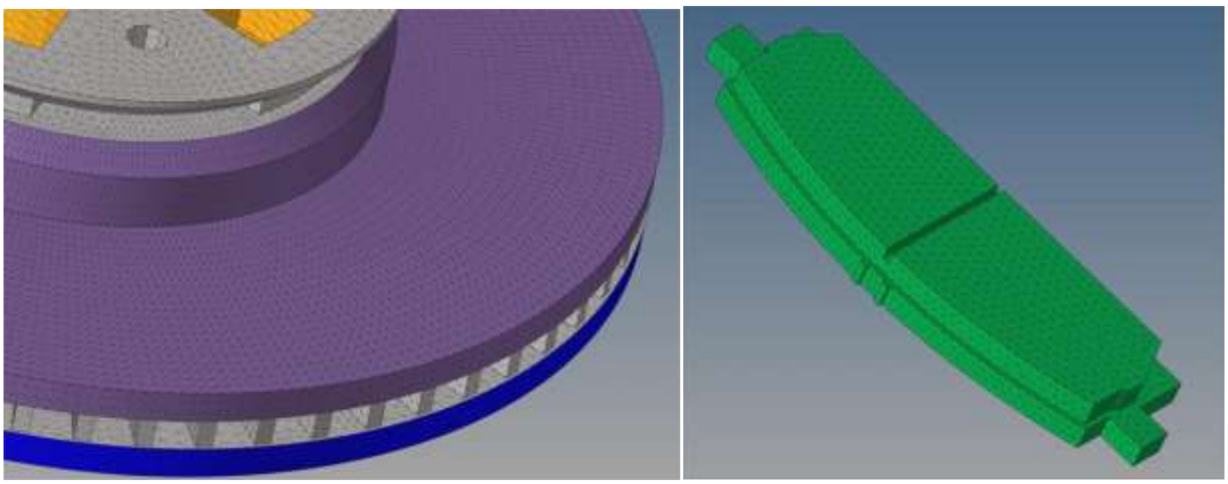


Figure 2.11. Mesh of modal.

First, a 2D surface mesh is generated for the friction surfaces of the brake disc and brake pad, using quad elements with an element size of 2 mm, as shown in Fig. 2.11.

After obtaining a high-quality 2D surface mesh, a 3D mesh is then created for the entire model, employing R-trias (triangular) elements with an element size of 2 mm.

Determination of Materials Used in the Study

The materials used for testing the brake disc in this research include*:* CastIron, ALSi C20% and AISI6150.

Table 1. Material properties.

	Density (RHO)	Elastic Modulus (E)	Poisson's ratio (NU)	Mass of Original Brake Disc	Mass of Ventilated Brake Disc
Unit	g/cm3	Gpa		Kg	Kg
CastIron	7.15	120.5	0.3	9.267	8.259
ALSiC20% (Metal matrix composite)	2.74	99	0.35	3.551	3.164
AISI1650 (Stainless steel)	7.85	205	0.29	10.174	9.067

Create Materials (which define the physical properties of the materials used by elements in the simulation) in OptiStruct and assign the corresponding parameters to each material type.

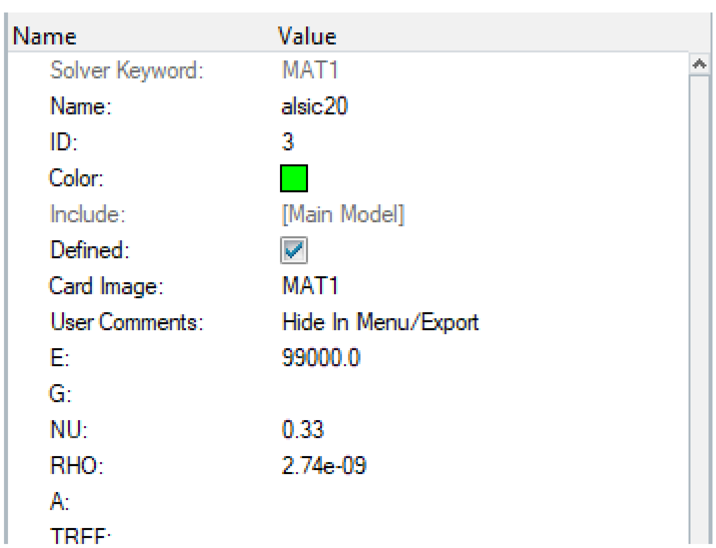


Figure 2.12. Setting Up the Material Tab.

Next, create two types of Properties (Properties define how elements respond under loads based on geometry and material): PSOLID (used for solid elements, 3D solid models without thickness) for the 3D mesh, and PSHELL (used for shell elements, requiring thickness definition, effective for thin structures) for the 2D mesh with thickness T = 0.001.

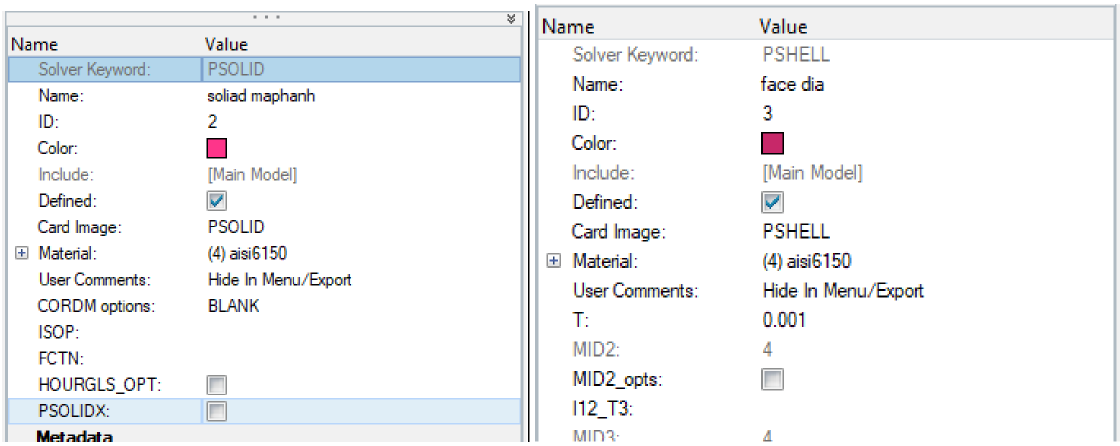


Figure 2.13. Setting Up the Properties Tab.

Next, fix the positions of the bolts on the brake disc and the rotational axis of the brake disc using RBE2, as shown in Fig. 3.14. RBE2 is an important rigid connection element, mainly used to constrain the motion between a master node and multiple slave nodes, where the slave nodes undergo the same displacement as the master node. It is used to represent joints, surface attachments, simulate rigid connections between components replacing bolts, and create various boundary constraints…

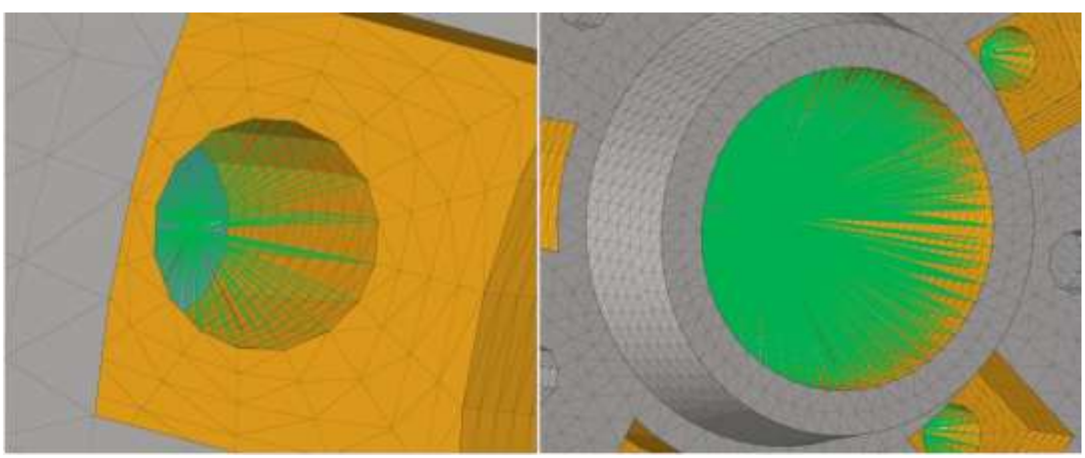


Figure 2.14. RBE2 on bulong and axis of rotation.

Fix the outer surfaces of the two brake pads using RBE3, as shown in Fig. 2.15. RBE3 is used to distribute loads or displacements from a master node to multiple slave nodes without creating a rigid connection between them. It is typically applied when distributing forces or moments from one main point to several others without altering the overall stiffness of the model.

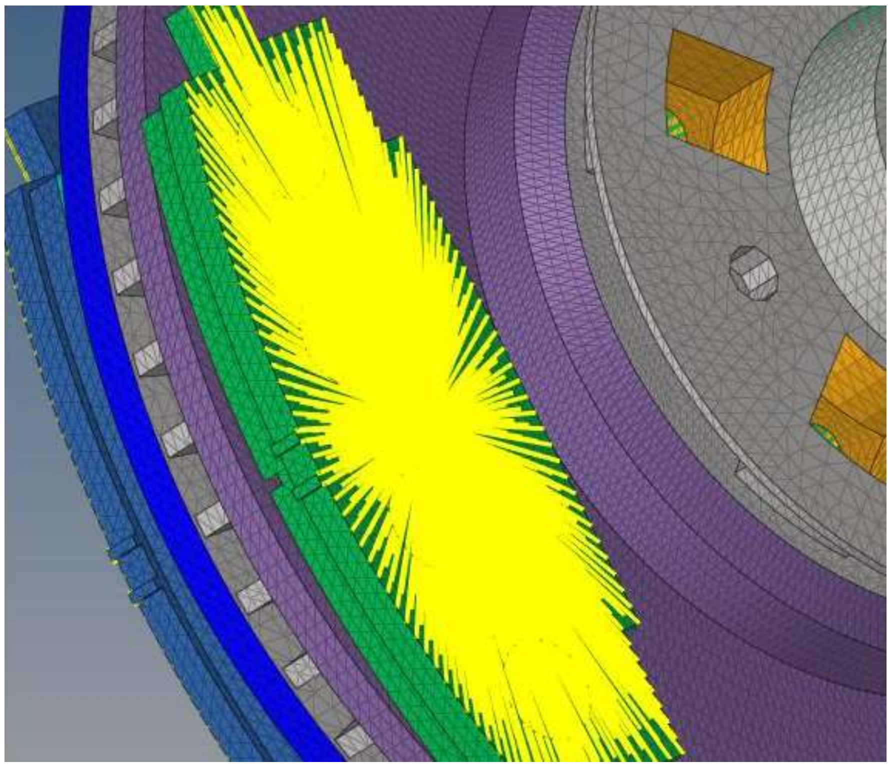


Figure 2.15. RBE3 on Brake Pad.

Create SPCs (Single Point Constraints) to restrict the motion of nodes in the model, that is, to limit the degrees of freedom of a node in one or more specific directions. Apply these SPCs to the nodes associated with the RBE3 and RBE2 elements, as shown in Fig. 2.16.

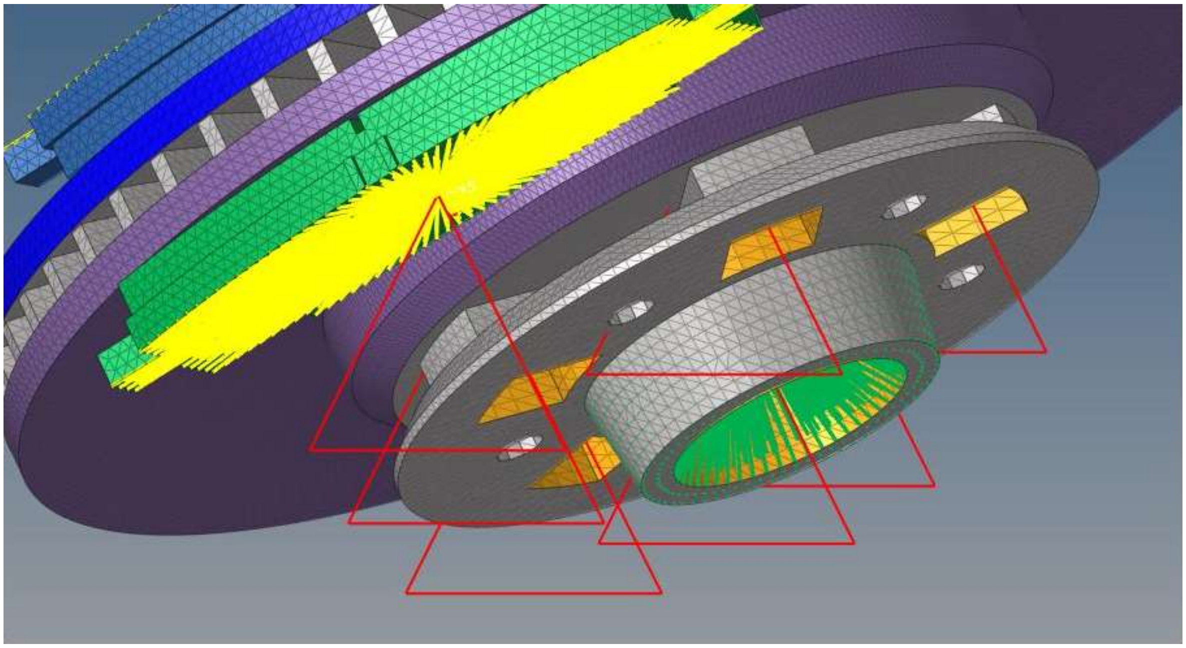


Figure 2.16. Setting SPCs at Nodes Containing RBEs.

Create LOADs by assigning the braking force applied on the brake pads. To determine the magnitude of this force, assumptions are made as follows: The vehicle is fully loaded with 12 passengers, each weighing an average of 70 kg plus 20 kg of luggage. The actual total mass of the vehicle is G=1650 kg.

Therefore, the total mass including passengers and luggage is Gt= 2730 kg.

Assuming the vehicle needs to stop within 60 meters, and other parameters are set under ideal conditions with the vehicle moving straight at a constant speed, the scenarios considered are speeds of 70 km/h and 120 km/h. Based on Newton's laws and the kinematic equations for uniformly decelerated motion, we have:

$$V^2 - V_0^2 = 2as \qquad [3.1]$$

Where: V: final velocity (at point 2) V_0: initial velocity

s: stopping distance (m) a: vehicle deceleration

From the above equation, the braking force acting on each brake pad is determined as follows:

Table 2. Value of Brake force according to vehicle speeed.

Vehicle speed (km/h)	Value of Brake force (N)
70	4300.733025
120	12638.88889

After obtaining the force values, apply them to the two RBE3 points on both brake pads along the Z direction (the direction of the force exerted by the brake pads on the brake disc), as shown in Fig. 2.17.

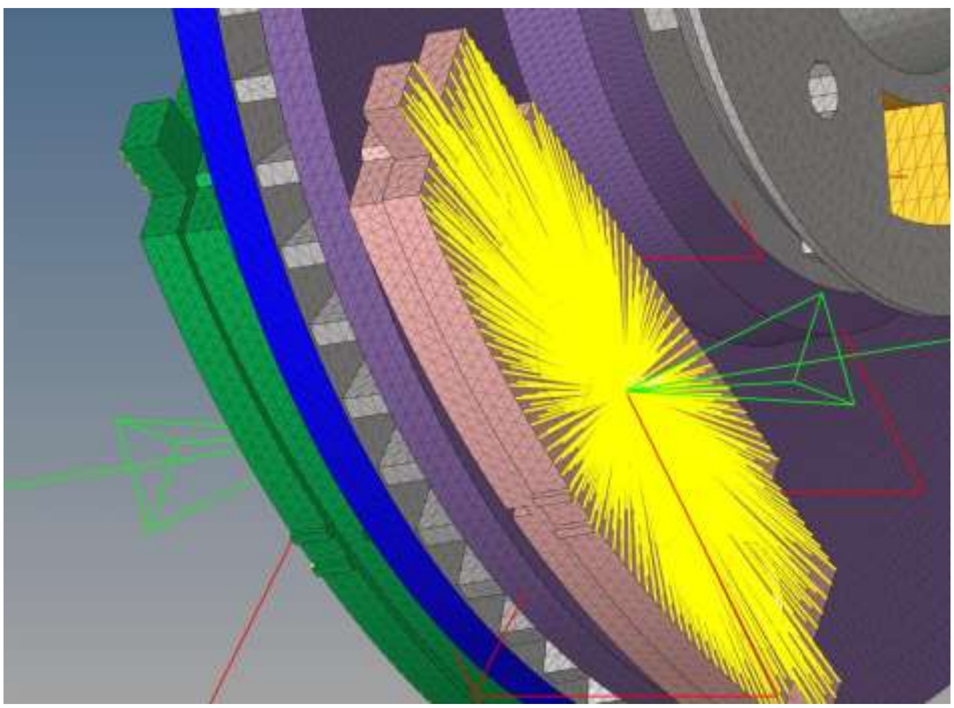


Figure 2.17. Applying Forces to the Two Brake Pads.

Create a TABLED1 (used to describe the relationship between two varying quantities in a tabular form) to define the frequency response table.

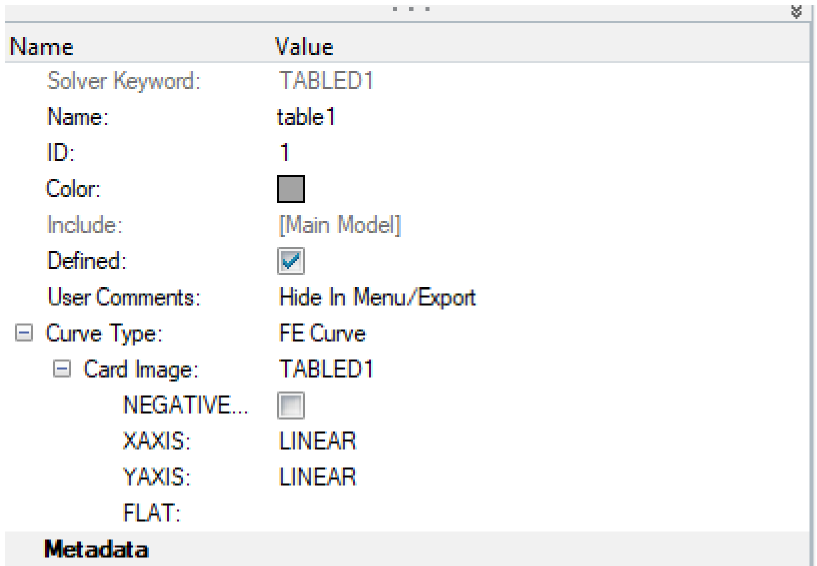


Figure 2.18. Setting Up the TABLED1 Tab.

Create an RLOAD2 (used to define harmonic oscillating loads in frequency response analysis, expressed as a complex function of frequency), assign the LOAD, and define the force according to the frequency data in TABLED1.

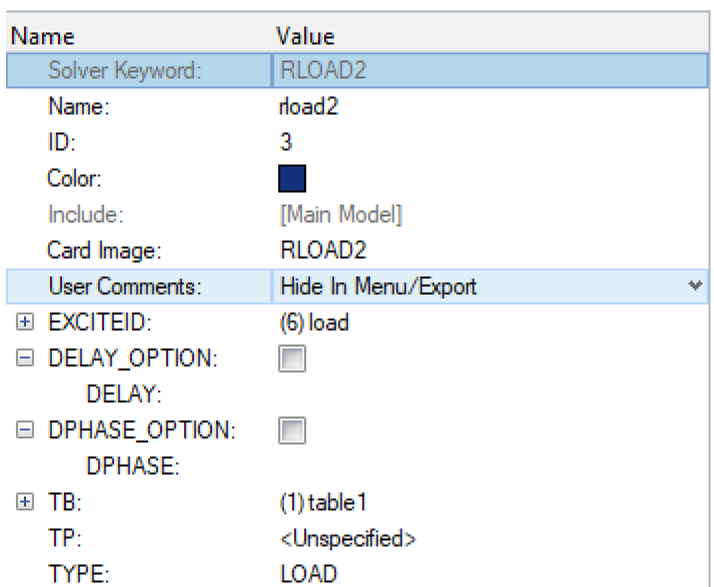


Figure 2.19. Setting Up the RLOAD2 Tab.

Create FREQi (used to define the frequency range for frequency response analysis or modal analysis problems), specify the excitation frequency range using RFEQi (Frequency Independent Intervals). Randomly select the initial frequency range as 5000 Hz with 500 calculation steps.

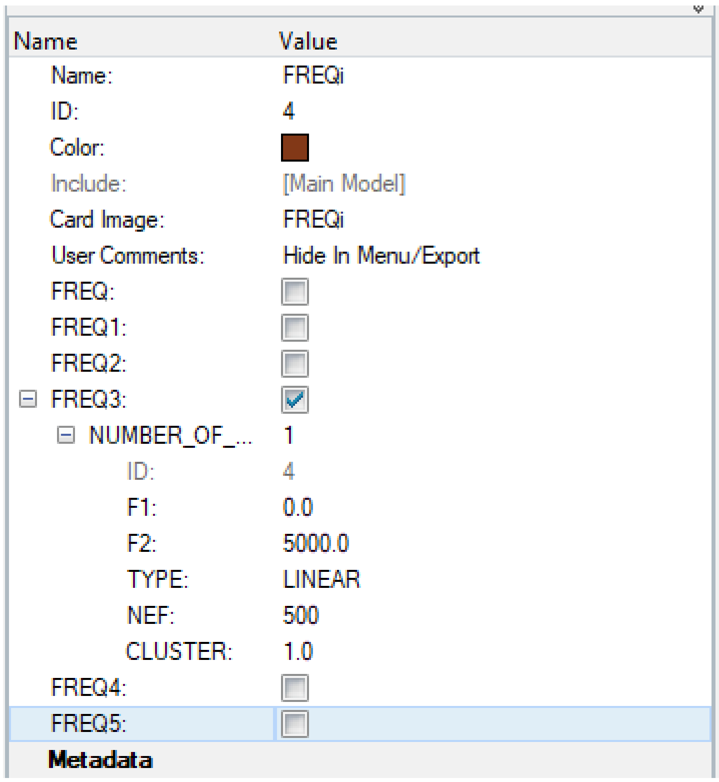


Figure 2.20. Setting Up the FREQi Tab.

Create EIGRL (a card used to define the eigenvalue solver in modal analysis) to specify the solver for the problem, with the frequency range set from 0 to 5000 Hz.

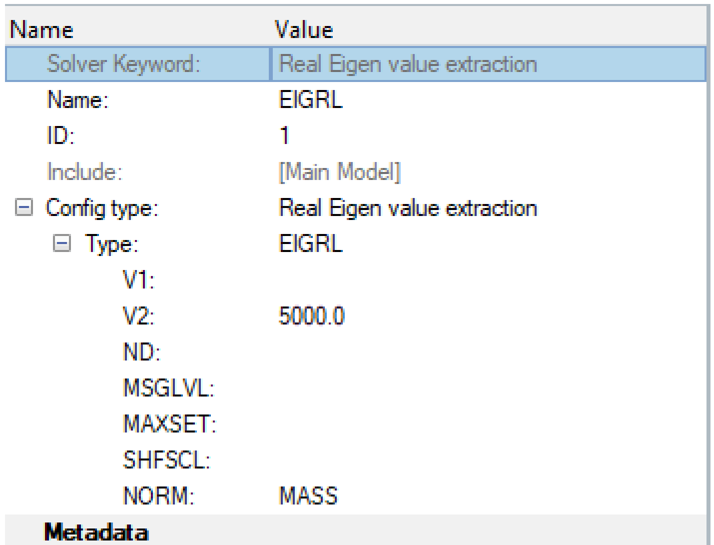


Figure 2.21. Setting Up the EIGRL Tab.

Create CONTACT (used to define and control contact conditions between surfaces or elements in the model) between the two brake pads and the brake disc. Set the contact condition on the mating surfaces of the brake pads and the disc with a Static Friction Coefficient.

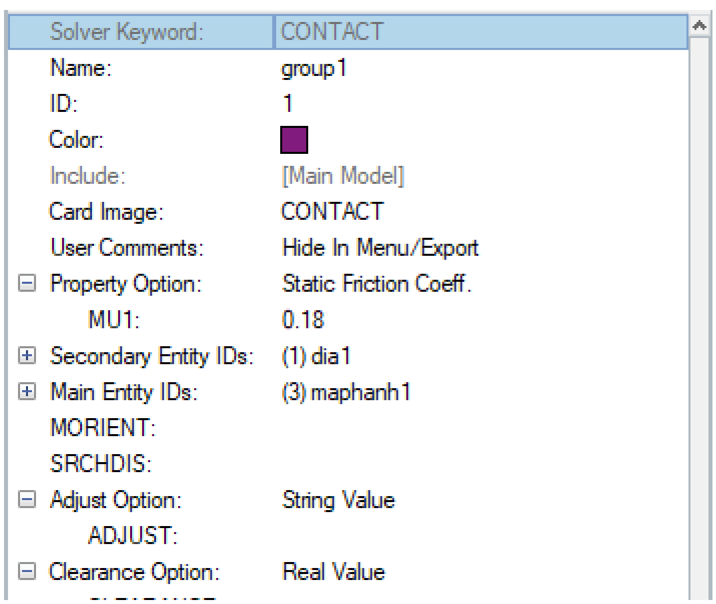


Figure 2.22. Setting Up the CONTACT Tab.

Create LOAD STEPS (used to define the type of analysis, link load and boundary conditions, specify solution methods and algorithms...), and define the Normal Modes Analysis case as follows.

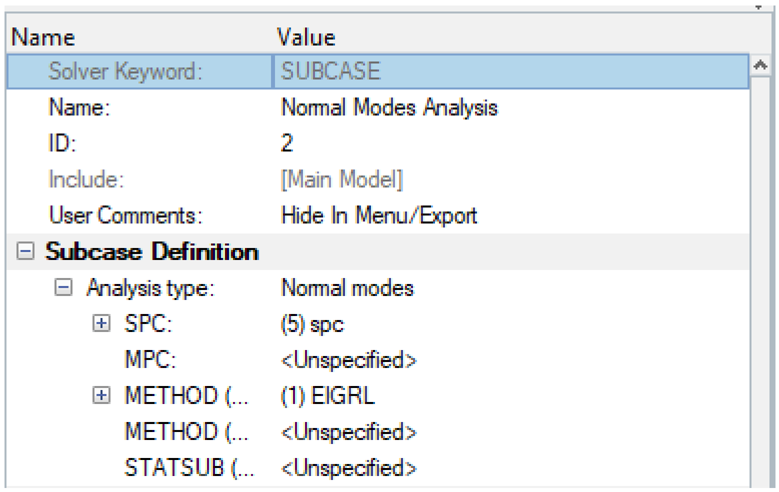


Figure 2.23. Setting Up the LOAD STEPS Tab for Normal Modes Analysis.

Create LOAD STEPS and define the Frequency Response Analysis case as follows.

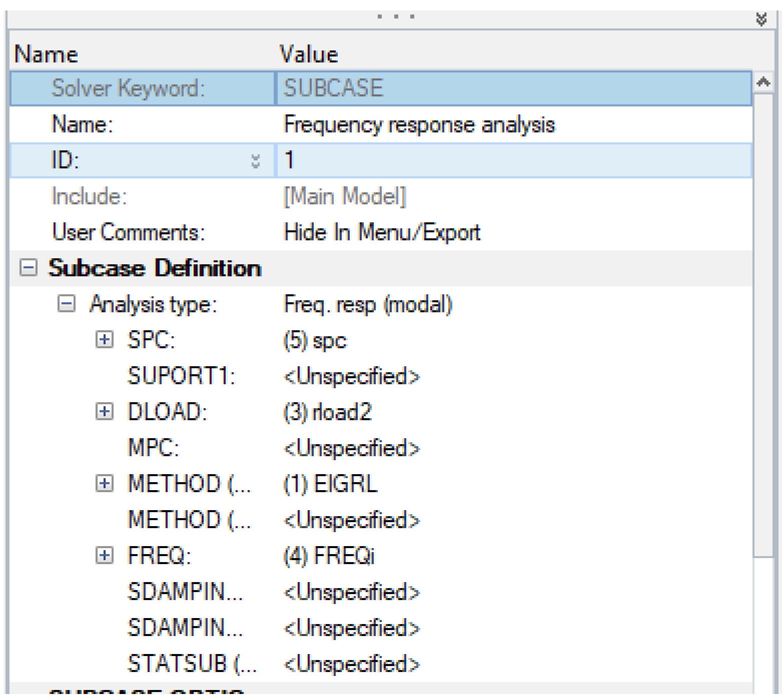


Figure 2.24. Setting Up the LOAD STEPS Tab for Frequency Response Analysis.

Next, define the OUTPUT REQUEST (used to specify the types of results to be saved, optimize output data, and support post-processing).

To evaluate the vibration response, three parameters can be considered: Frequency, Displacement, and Acceleration. Since Displacement and Acceleration are proportional [12], only two parameters need to be selected for measurement and evaluation. Therefore, choose Frequency and Displacement, as shown in Fig. 2.25.

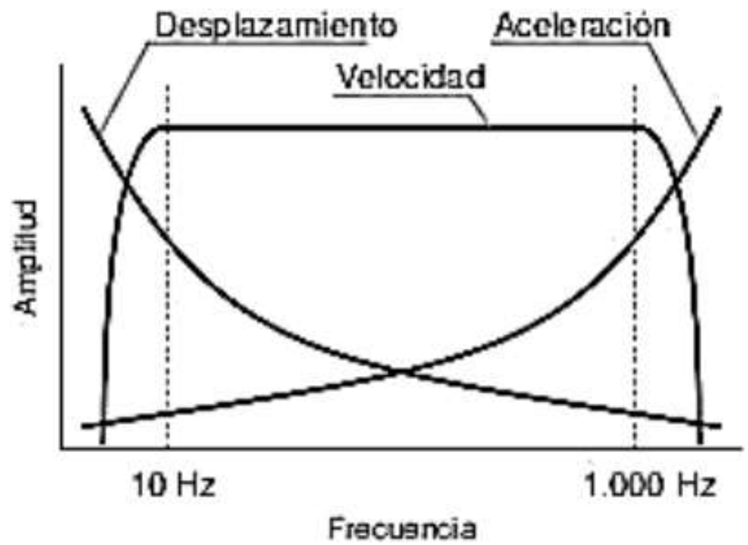


Figure 2.25. Magnitudes en frequency.

CHAPTER 3. RESULTS ANALYSIS

3.1. Meshing Convergence Method

To select the appropriate mesh size that ensures computational efficiency with reasonable processing time, tests are conducted on five different mesh sizes: 0.5 mm, 1 mm, 2 mm, 3 mm, and 5 mm. The goal of this process is to identify the optimal mesh type that balances simulation accuracy and minimizes solution time.

The evaluation is performed by measuring displacement at the same fixed point on the model for all the mesh sizes mentioned above, as shown in Fig. 3.1.

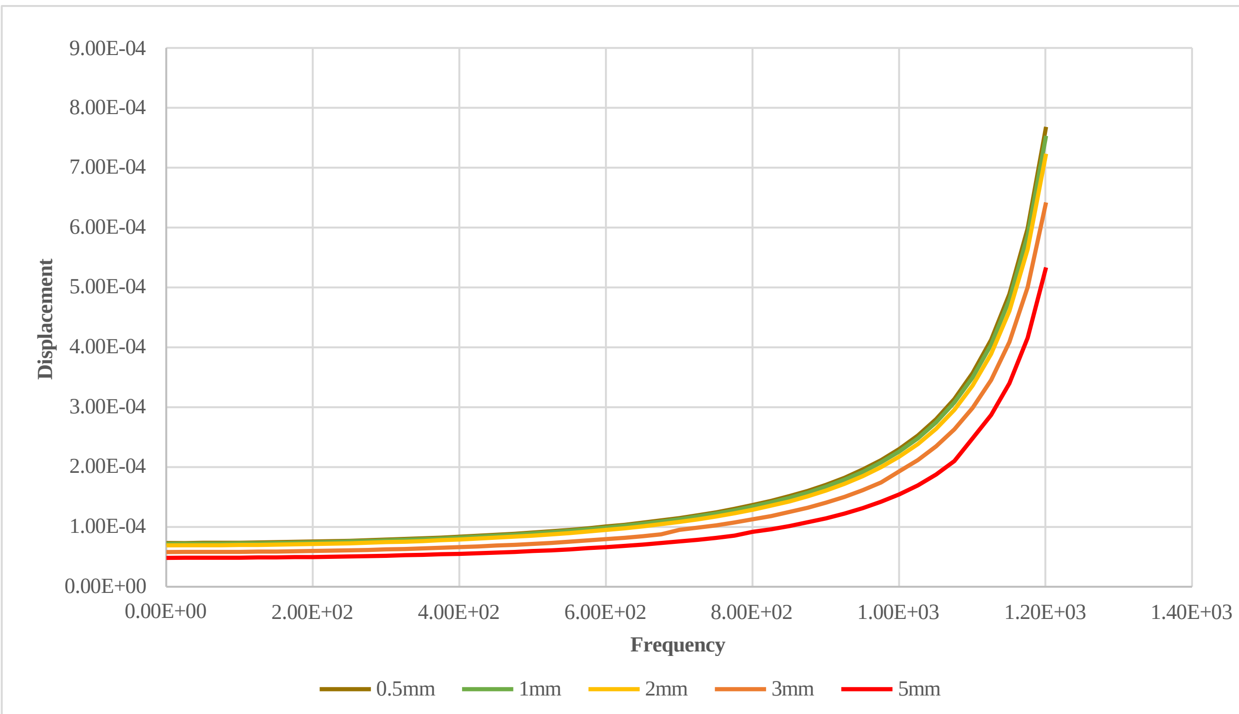


Figure 3.1. Results of different mesh types.

The results show that when the mesh size decreases from 2 mm to 0.5 mm, the measured values exhibit no significant differences, indicating that the solution has reached convergence. This suggests that a 2 mm mesh is sufficiently fine to accurately simulate the vibration characteristics of the system without the need for a finer mesh.

From this, it can be concluded that to significantly reduce computation time while still ensuring reliability and accuracy of the results, choosing a 2 mm mesh size is the most reasonable option. This mesh size will be applied for all subsequent analyses in the study.

Continuing with the selected mesh, the frequency response simulation is conducted over a frequency range of 5000 Hz, with 500 calculation steps during the solution process, as shown in Fig. 3.2.

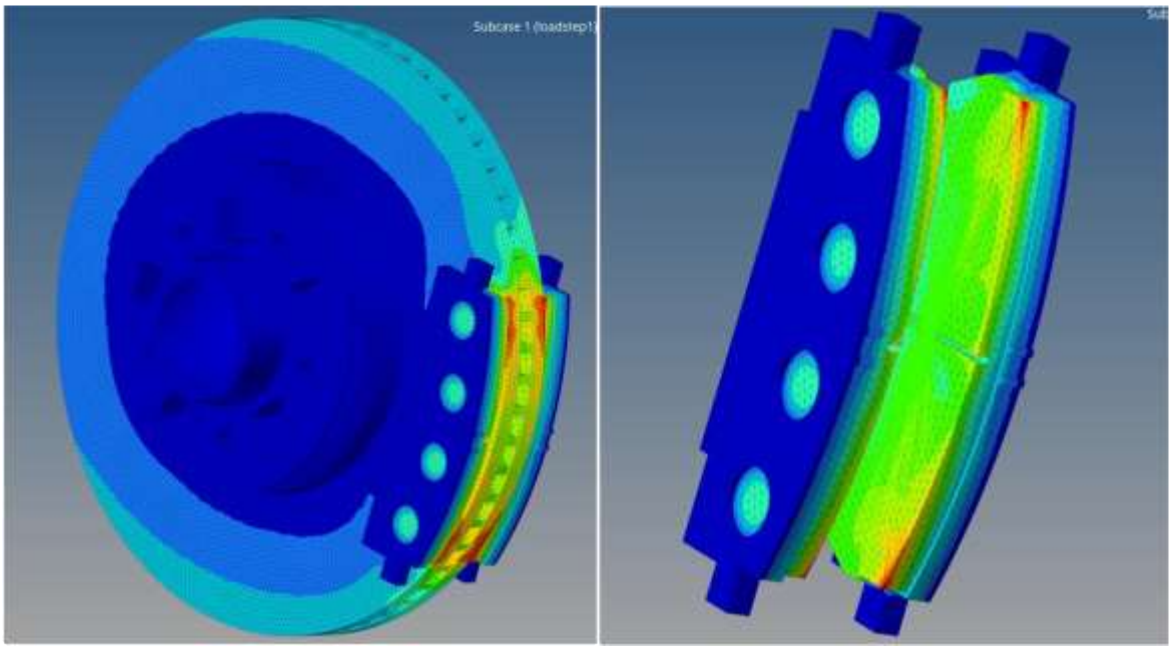


Figure 3.2. Modal H3D results of frequency response analysis.

Next, to further optimize computational resources, additional calculations are performed for two cases with different frequency ranges and numbers of calculation steps. The displacement at the same point is measured in all three cases to select the most appropriate frequency range for the analysis:

- Analysis over a frequency range of 3000 Hz with 300 calculation steps.
- Analysis over a frequency range of 10000 Hz with 1000 calculation steps.

The results obtained are as follows

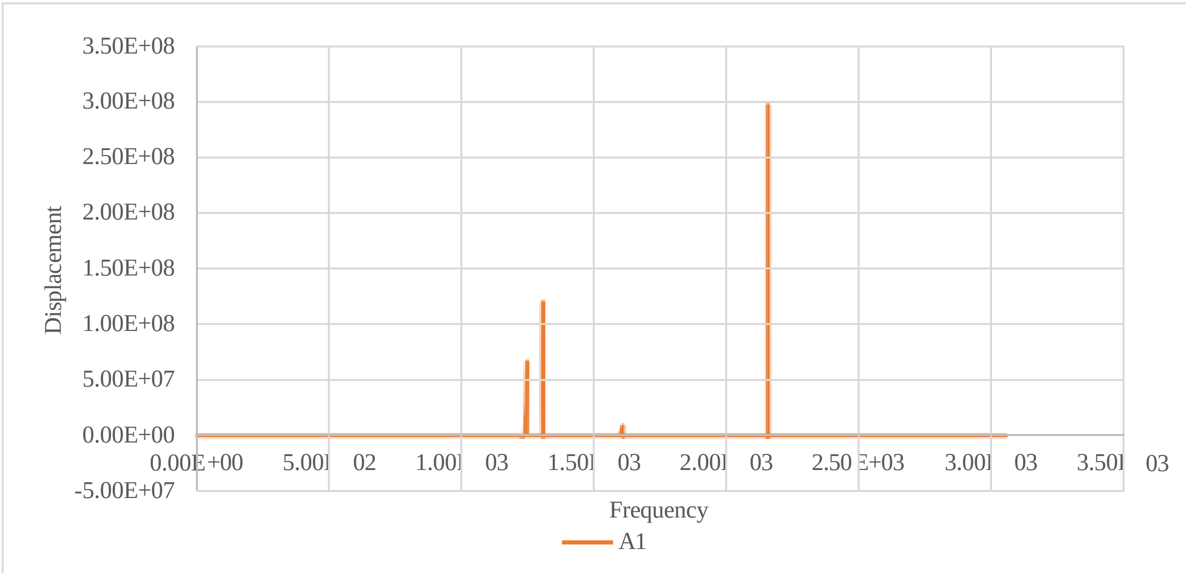


Figure 3.3. Result of frequency response analysis at 3000 Hz with 300 step.

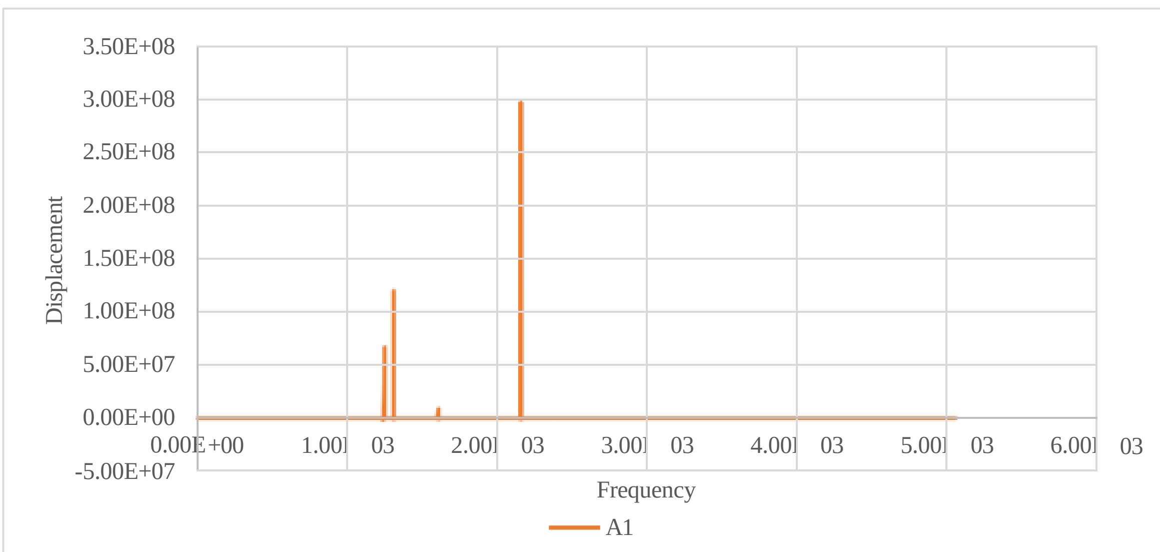


Figure 3.4. Result of frequency response analysis at 5000 Hz with 500 step.

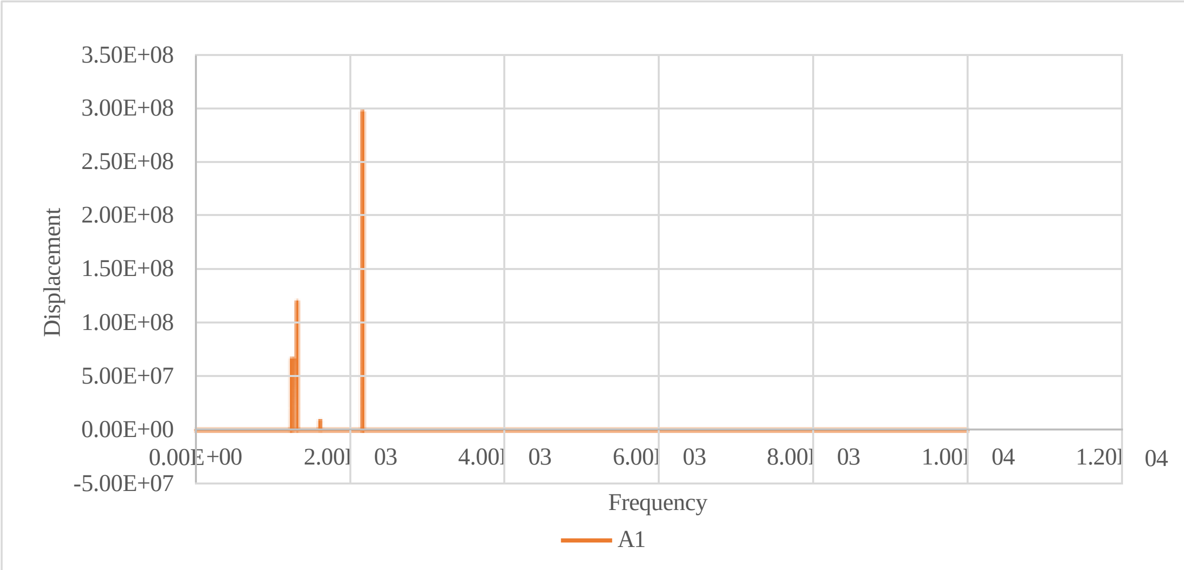


Figure 3.5. Result of frequency response analysis at 10000 Hz with 1000 step.

As shown in Fig. 3.3- 3.5, from the frequency response analysis results at 3000 Hz, 5000 Hz, and 10,000 Hz, it can be observed that the measured values at the monitored point show no significant differences. This indicates that the 3000 Hz frequency range sufficiently covers all the important vibration phenomena within the system. Therefore, to optimize computation time and reduce data processing load, selecting the 3000 Hz frequency range for all study cases is a reasonable and effective solution.

After obtaining the Displacement results from the frequency response analysis, these are compared with the results from the Normal Modes Analysis. When measuring displacement at a random point on the model, the peak amplitudes in the frequency response perfectly coincide with the resonance frequencies identified in the modal analysis. This confirms the high reliability of the model and the calculation method applied.

Subsequently, similar calculations are performed for all remaining study cases to comprehensively evaluate the vibration characteristics of the brake disc under different material properties, geometries, and operating conditions.

For a more detailed analysis of vibration behavior at each measurement point, focus is placed on the frequency range from 0 Hz up to just before the first resonance frequency. This detailed analysis in the low-frequency region helps clarify the fundamental vibration characteristics, thereby enabling the proposal of technical solutions to prevent or minimize resonance effects and noise generation during operation.

Measurements are taken at 3 points with the following positions and coordinates for each disc, as shown in Table 3 and Fig. 3.6.

Table 3. The coordinates of the points are determined.

Original Brake Disc	A1: N101472-Z	A2: N101431-Z	A3: N96252-Z
Ventilated Brake Disc	A1: N116918-Z	A2: N122343-Z	A3: N119621-Z

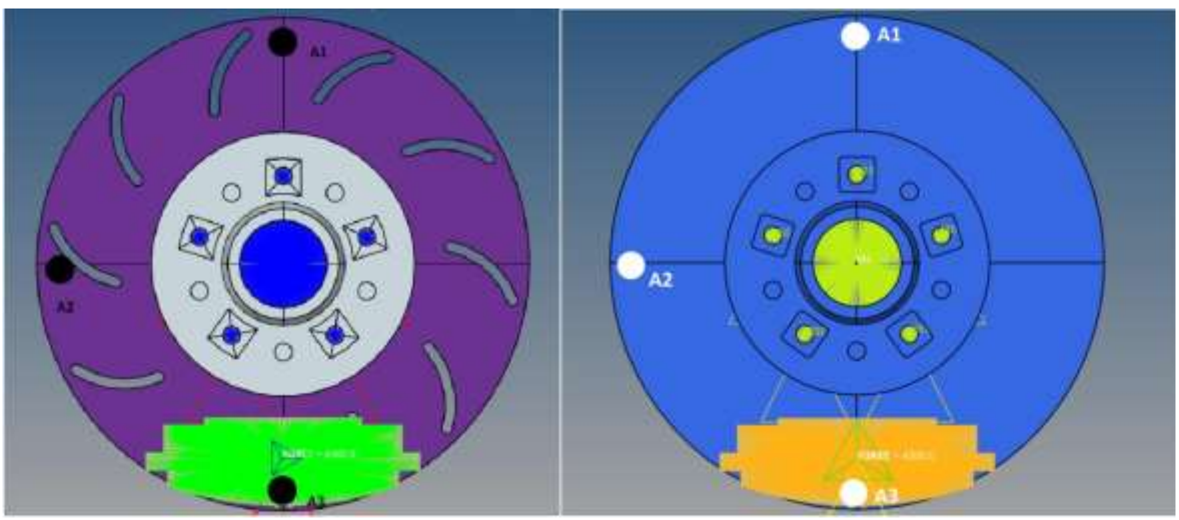


Figure 3.6 Location of A1, A2, A3 on both types of Brake Disc.

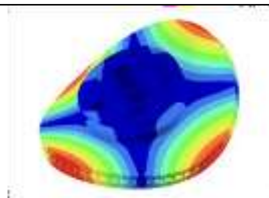
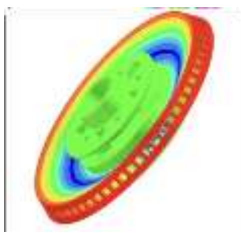
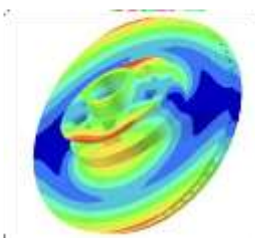
All points are situated near the outer edge of the disc; these three points correspond to the three regions illustrated in the figure above, each located 3 mm from the brake disc's rim

3.2. Results of the Original Brake Disc Case

3.2.1. Normal Modes Analysis

Upon completion of the Normal Modes Analysis, the first mode responsible for resonance in the Original Brake Disc was identified as follows, as shown in Table 4.

Table 4. Results of Normal Modes Analysis of Original Brake Disc.

Original Brake Disc		CastIron	ALSiC20%	AISI6150
Lateral bending mode		1.317e+3	1.827e+3	1.641e+3
Longitudinal bending mode		2.178e+3	3.208e+3	2.707e+3
Torsional mode		2.735e+3	4.023e+3	3.401e+3

3.2.2. Frequency Response Analysis at 70 km/h

a. CastIron.

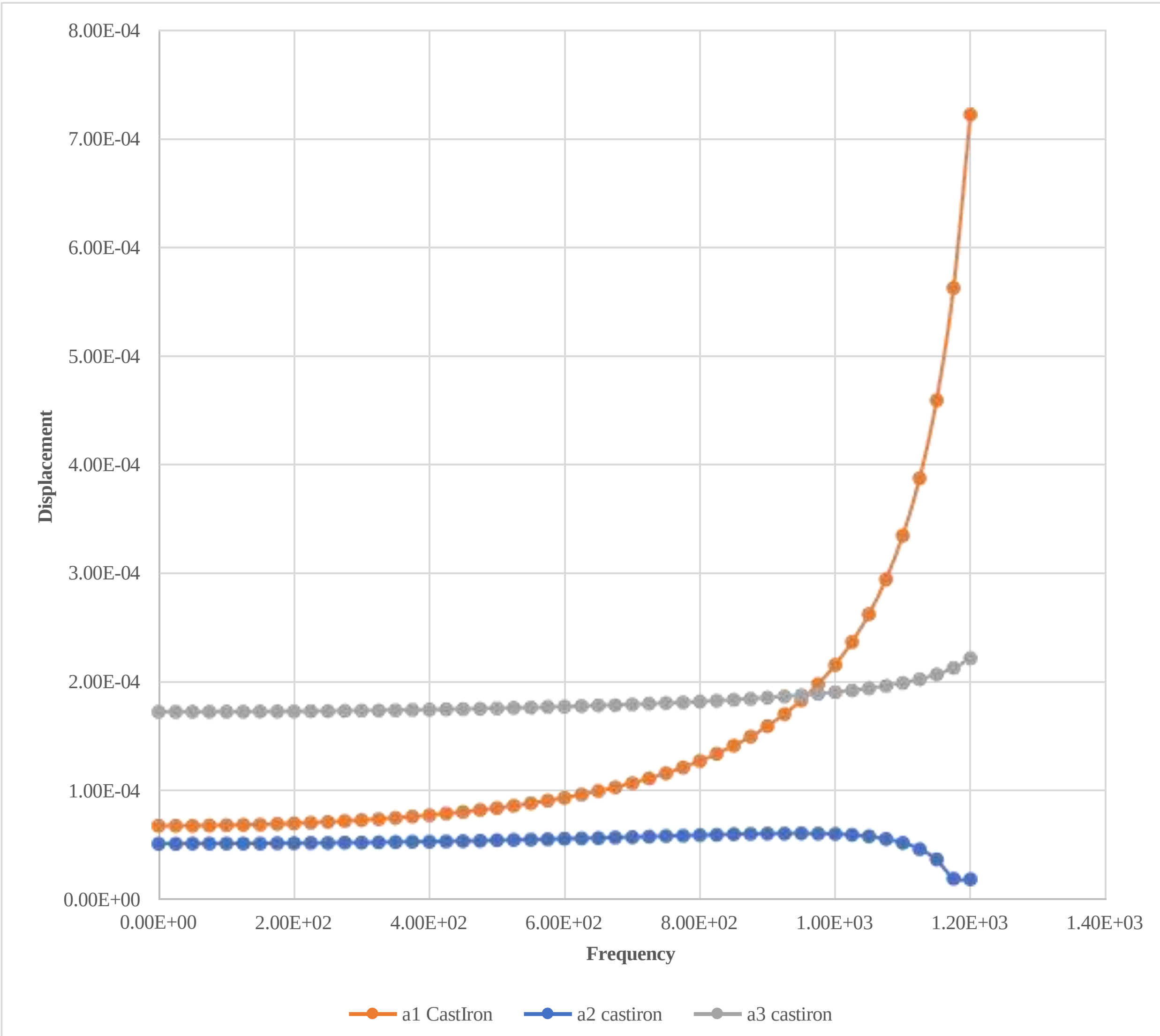


Figure 3.7. Comparison of displacement results for A1, A2, and A3 using CastIron material for Original Brake Disc.

b. ALSiC20%.

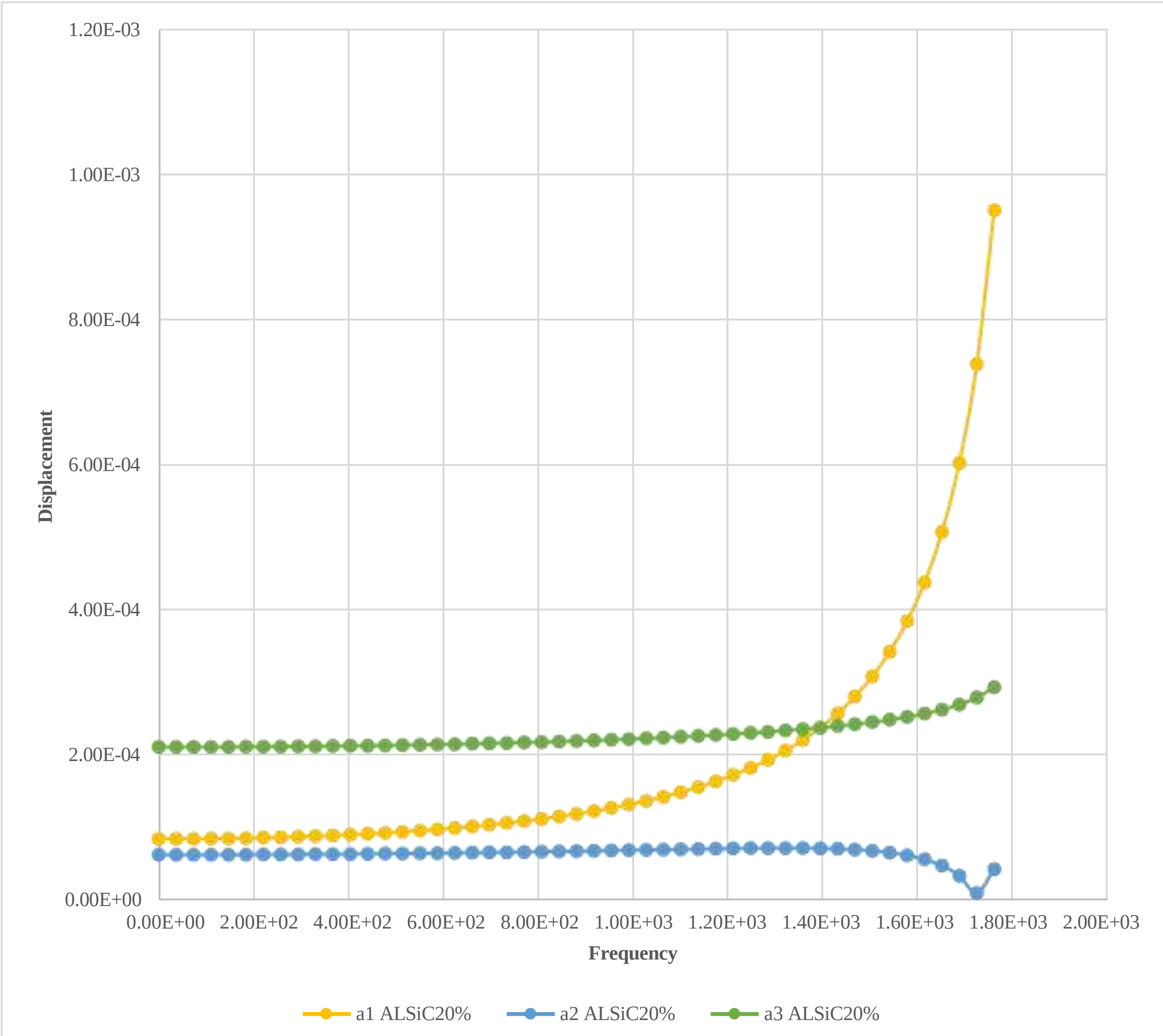


Figure 3.8. Comparison of displacement results for A1, A2, and A3 using ALSiC20% material for Original Brake Disc.

c. AISI1650.

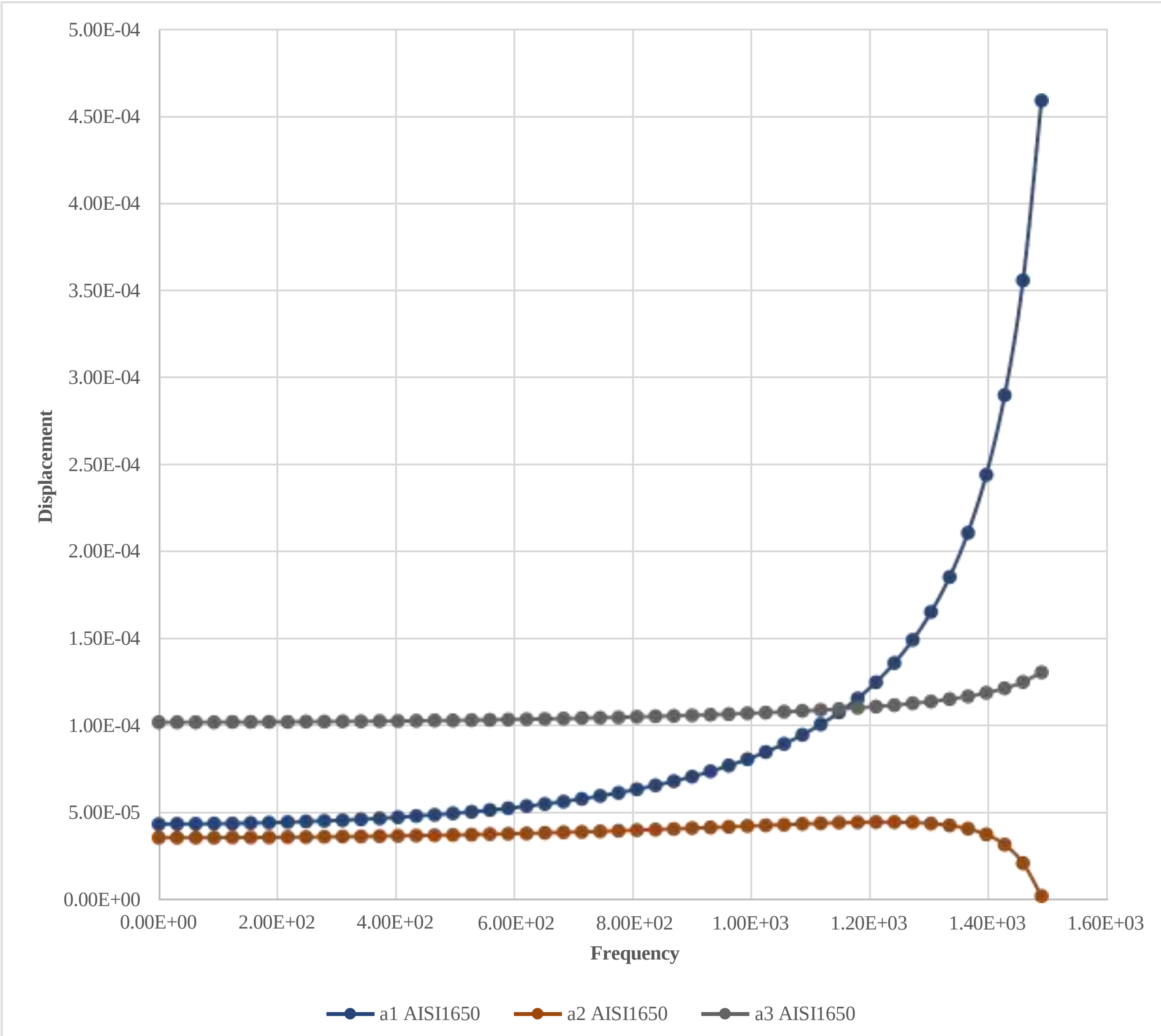


Figure 3.9. Comparison of displacement results for A1, A2, and A3 using AISI1650 material for Original Brake Disc.

Based on the results obtained from the analysis graphs, as shown in Figs. 3.7- 3.9, it can be assessed that the displacement trends at the measurement points A1, A2, and A3 across different materials exhibit a similar pattern. This confirms that although different materials have distinct mechanical properties, the vibration mode shapes and displacement distribution on the brake disc follow a consistent rule governed by the geometry and material characteristics.

Specifically, the magnitude of displacement consistently decreases in the order of point A3, followed by A1, and finally A2. This observation is significant in analyzing the influence of point location on vibration amplitude. Point A3 always shows the highest displacement value, which can be explained by its proximity to the area subjected to the strongest excitation, where braking forces or dynamic loads concentrate, increasing vibration amplitude at this location.

However, it is noteworthy that point A2, although closer to the excitation area than point A1, consistently exhibits a smaller displacement value. This phenomenon contradicts the previous assumption that points nearer to the excitation force would have larger displacements.

To accurately explain this behavior, it is necessary to further examine the position of point A4, with coordinates N96253-Z, to better understand the influence of local geometry and structural properties of the disc in the surrounding regions.

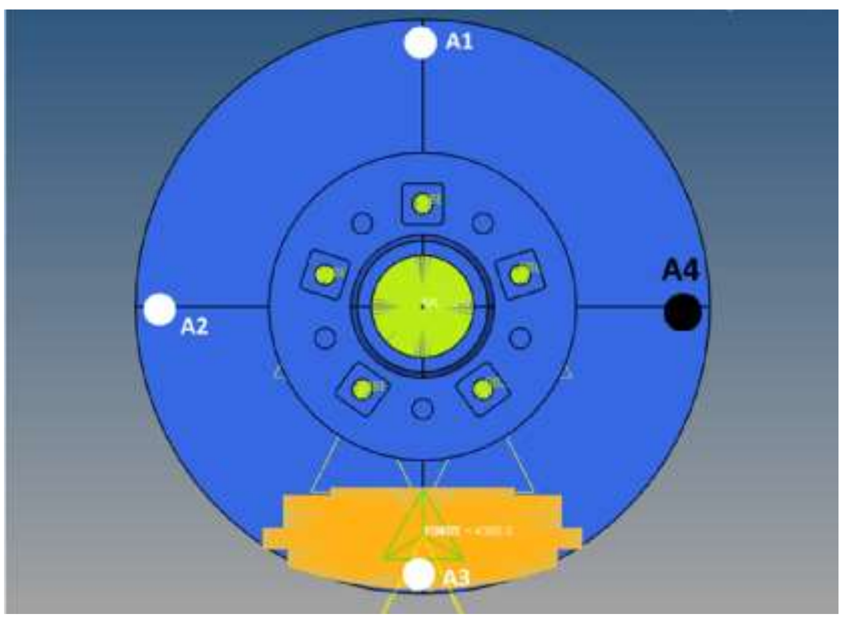


Figure 3.10. Location of A4 on Brake Disc.

Divide the X and Y axes on the disc surface, with the horizontal X-axis containing points A2 and A4, and the vertical Y-axis containing points A1 and A3. The displacement values at point A4 are always similar to those at A2 (confirmed through multiple tests), meaning that points A1 and A3 consistently exhibit larger displacements than A2 and A4.

In other words, the displacement along the vertical Y-axis is always greater than that along the horizontal X-axis, indicating that the model undergoes bending in the horizontal direction.

Cross-referencing with the modal analysis results, the earliest resonance peak corresponds to horizontal bending. Since the measured cases are taken at the first peak for each material, we can summarize and briefly explain the above phenomenon as follows.

Because the first resonance mode causes horizontal bending, and the measurements are taken at the frequency range where this resonance occurs, points along the vertical axis on the brake disc experience higher displacements compared to others. Therefore, the resonance mode shape of the brake disc affects the displacement distribution across different regions.

Now, the question arises: why do points A2 and A4 on the horizontal X-axis have similar displacement values, while points A1 and A3 on the vertical Y-axis differ? To support the explanation that A1 has smaller displacement because it is located farther from the excitation force, we consider two points A1’ (coordinates N101452-Z) and A3’ (coordinates N96232-Z) on the vertical Y-axis, arranged as shown in the following figure. Measurements at these points yielded the following results.

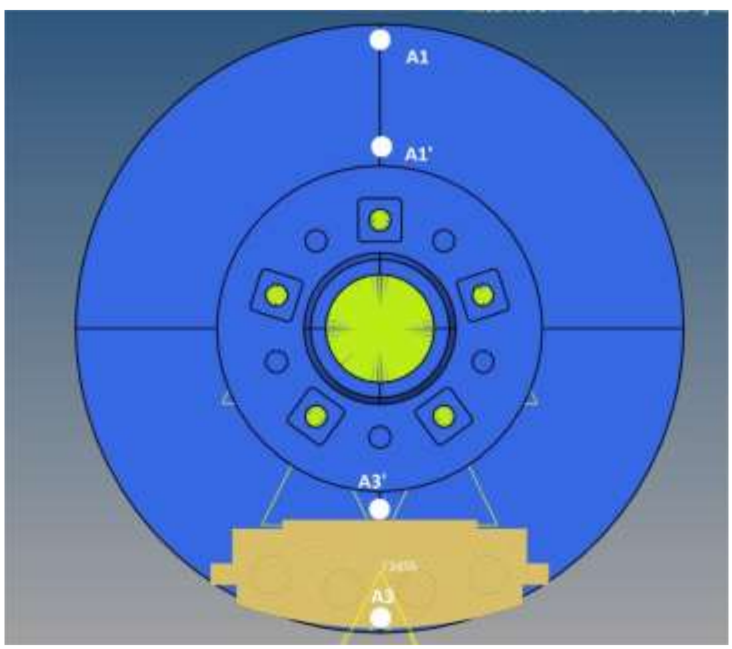


Figure 3.11. Location of A1’ and A3’ on Brake Disc.

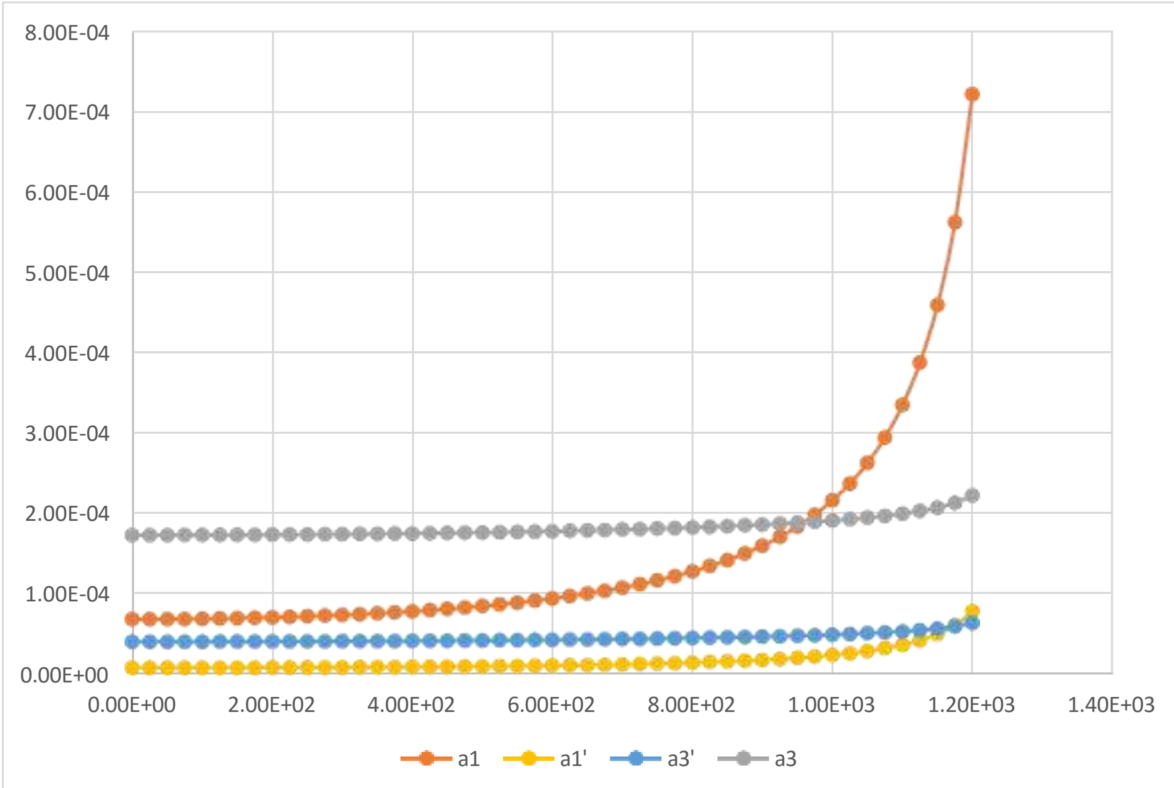


Figure 3.12. Comparison of displacement results for A1, A1', A3, A3' using CastIron materials for Original Brake Disc.

Thus, the above hypothesis is incorrect. The vibration amplitude along the Y-axis tends to decrease progressively as it approaches the center of the brake disc. However, points closer to the applied force still exhibit higher displacement magnitudes. This phenomenon can be explained by the fact that moving closer to the disc center means approaching the connection between the brake disc and the axle shaft. Additionally, the disc structure near the center is denser and heavier, which significantly reduces the displacement values.

Comparison of Three Materials for the Solid Disc Case at speed of 70 km/h

Figure 3.13 shows the comparison of three materials for the Solid Disc Case of 70 km/h. Since the trend of A1, A2, and A3 is similar across all three material types, we only compare one point among the three to evaluate the effect of the material on vibration and noise.

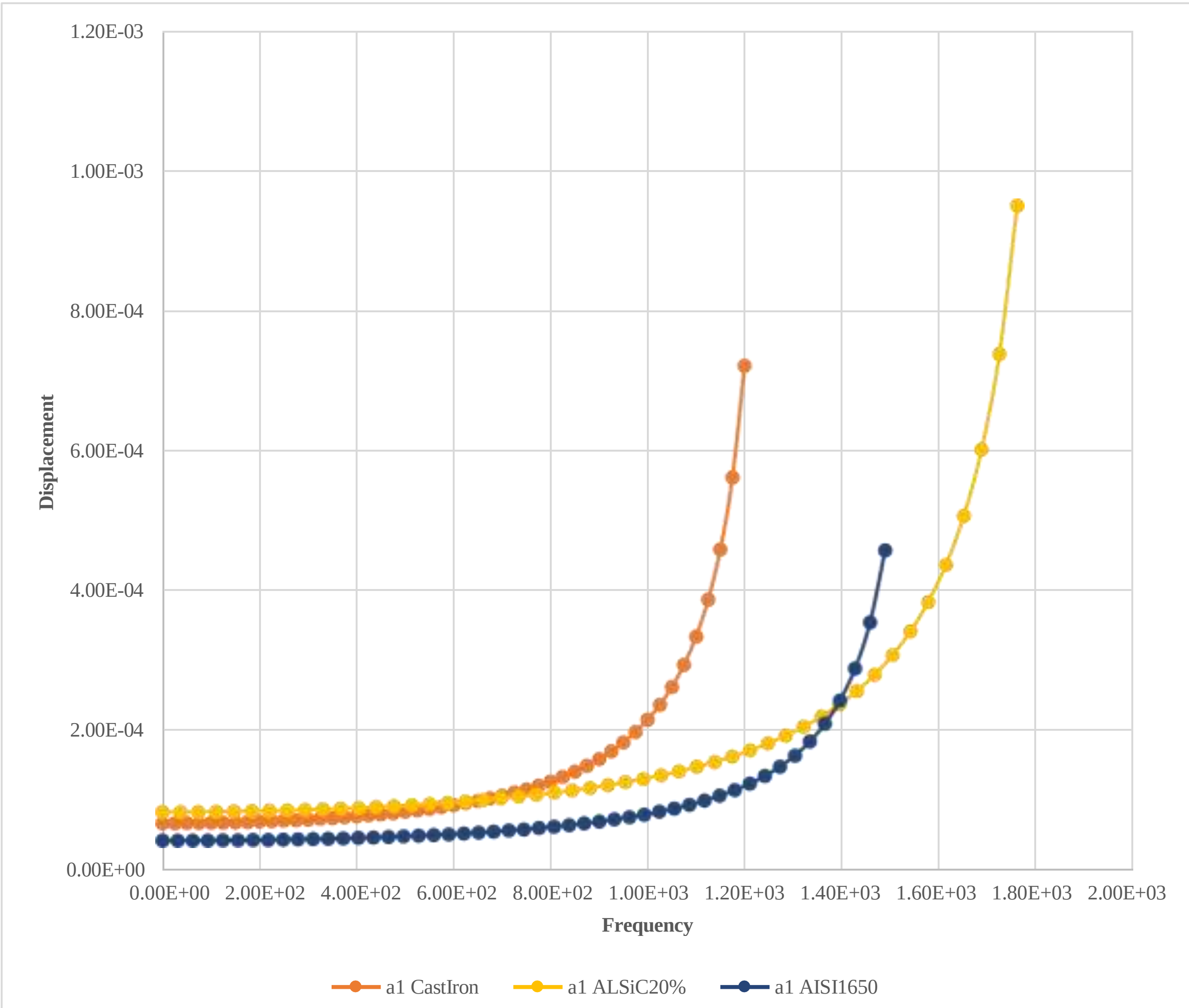


Figure 3.13. Comparison of displacement results for A1 using 3 different materials for Original Brake Disc.

Based on the modal analysis and the material properties table, we have the following.

Table 5. Comparison of Three Materials.

Material	First peak (Hz)	Mass (Kg)	Displacement (mm)
CastIron	1.3e+3	9.267	Middle
ALSiC20%	1.8e+3	3.551	Maximum
AISI1650	1.6e+3	10.174	Minimum

Evaluation of Results

Through the vibration characteristic analysis, the influence of material stiffness and brake disc mass on the system's vibration behavior can be clearly observed.

Stiffness is a factor that directly influences the resonance frequency range as well as the vibration displacement amplitude. Specifically, when the material stiffness increases, the natural frequencies of the brake disc shift to higher frequency ranges. As a result, the likelihood of resonance occurring within the typical operational frequency band is reduced. In other words, the higher the stiffness, the more difficult it is for the disc to be excited into resonance, thereby lowering the risk of excessive vibration and noise during operation. Conversely, with materials of lower stiffness, the natural frequencies tend to fall within the easily excitable range, leading to higher vibration amplitudes and an increased risk of resonance, which may reduce the overall stability of the braking system a result consistent with [3], [4], [11].

The mass of the brake disc also significantly affects the vibration amplitude. As the mass increases, the system's inertia also increases, making it respond more sluggishly to vibrational excitations and thus reducing the displacement amplitude. In other words, the greater the mass, the smaller the vibration (displacement) response, which helps limit structural vibration and noise. However, increasing mass must be carefully considered in relation to fuel efficiency and the system's dynamic responsiveness especially in applications that require lightweight structures and fast braking response as similarly reported in [4], [7], [8].

3.2.3. Frequency Response Analysis at a speed of 120 km/h

All the graphs for the 120 km/h case exhibit the same trend as those for the 70 km/h case (verified through multiple tests), as shown in Figure 3.14. Therefore, we proceed to compare the measurements at point A1 for all three material types under both speed conditions to examine the results obtained.

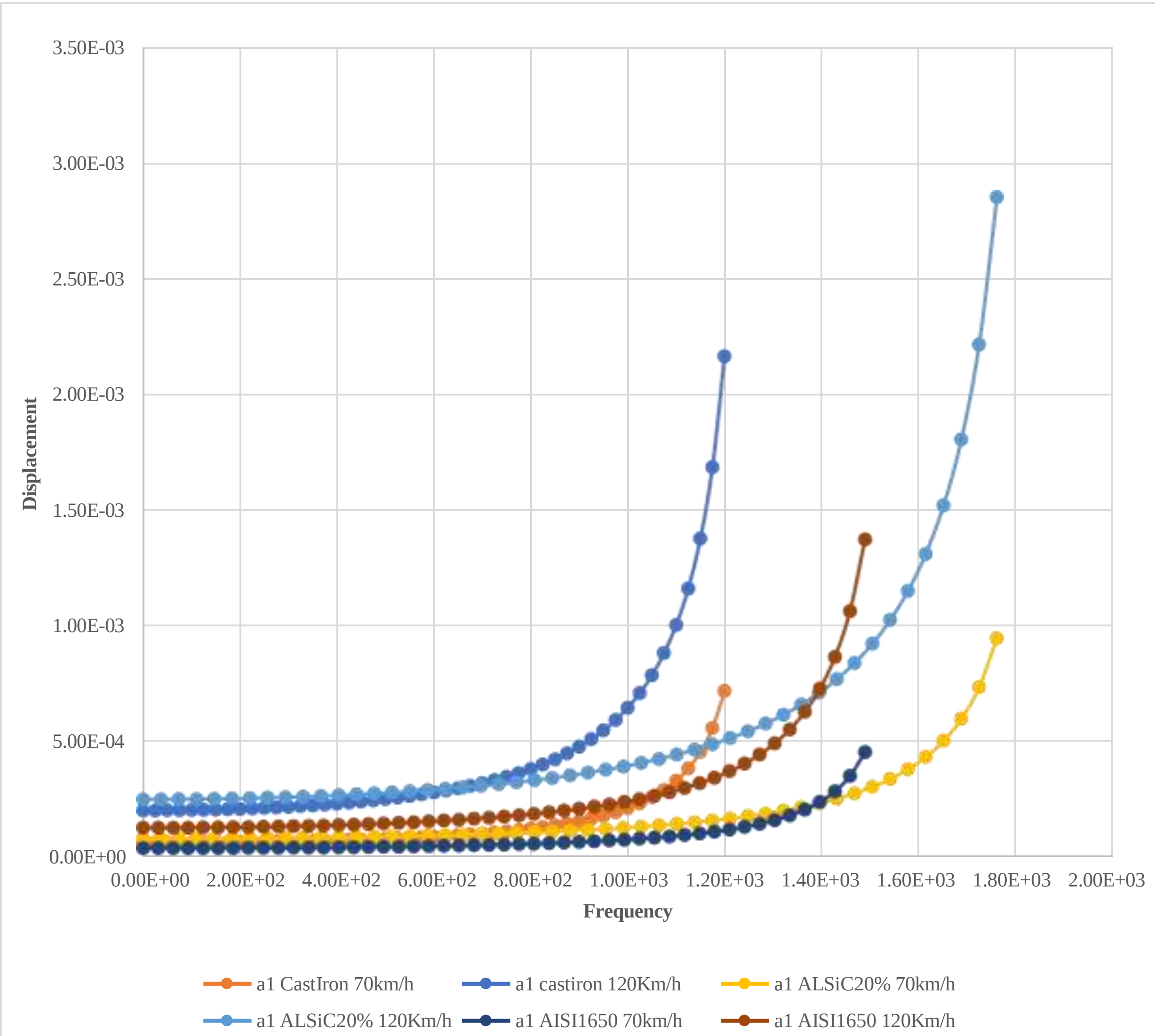


Figure 3.14. Comparison of displacement results for A1 using 3 different materials for Original Brake Disc in 2 case 70 km/h and 120 km/h.

Evaluation of Results.

Based on the analysis of the vibration graphs obtained, it can be observed that the velocity factor or more specifically, the braking force applied to the disc primarily affects the amplitude (magnitude) of the displacement vibrations. In other words, the greater the braking force, the more energy is transmitted into the system, causing the vibration amplitude to increase. However, this does not alter the overall vibration characteristics of the system, including the natural frequencies, mode shapes, or the displacement distribution pattern on the disc surface.

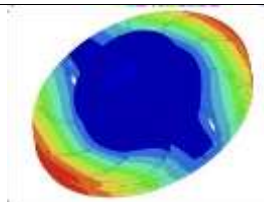
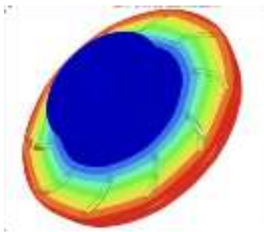
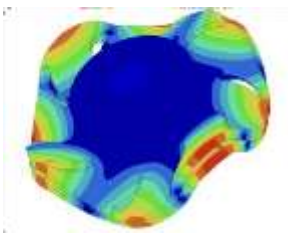
This indicates that dynamic parameters such as velocity or braking torque act as excitation factors, rather than determinants of the system's inherent vibrational characteristics. Even when velocity (or braking force) is varied, the vibration modes remain unchanged; only the system's response level (amplitude) varies in proportion to the intensity of the excitation. This observation is consistent with the findings reported in [3], [11].

3.3. Results for the Ventilated Brake Disc Case

3.3.1. Normal Modes Analysis

After processing the results of the Normal Modes Analysis, we obtained the first mode causing resonance in the Ventilated Brake Disc as follows, as shown in Table 6.

Table 6. Results of Normal Modes Analysis of Ventilated Brake Disc.

Ventilated Brake Disc		CastIron	ALSiC20%	AISI6150
Lateral bending mode		9.314e+2	1.372e+3	1.157e+3
Longitudinal bending mode		1.193e+3	1.762e+3	1.487e+3
Torsional mode		1.914e+3	2.802e+3	2.384e+3

From the results of the Normal Modes Analysis, it is observed that the stiffness of the Ventilated Brake Disc decreases significantly (the first peak occurs earlier) compared to the Original Brake Disc due to the reduction in mass.

As the influence of velocity has been clearly analyzed in previous sections with the conclusion that velocity (or the corresponding braking force) only affects the amplitude of vibration without changing the resonance frequency or mode shape in the following section, to save computational time while ensuring accuracy under realistic operating conditions, we choose to consider only a representative case at a velocity of 70 km/h.

3.3.2. Outcome of the Frequency Response Analysis 70 km/h Case.

a. CastIron.

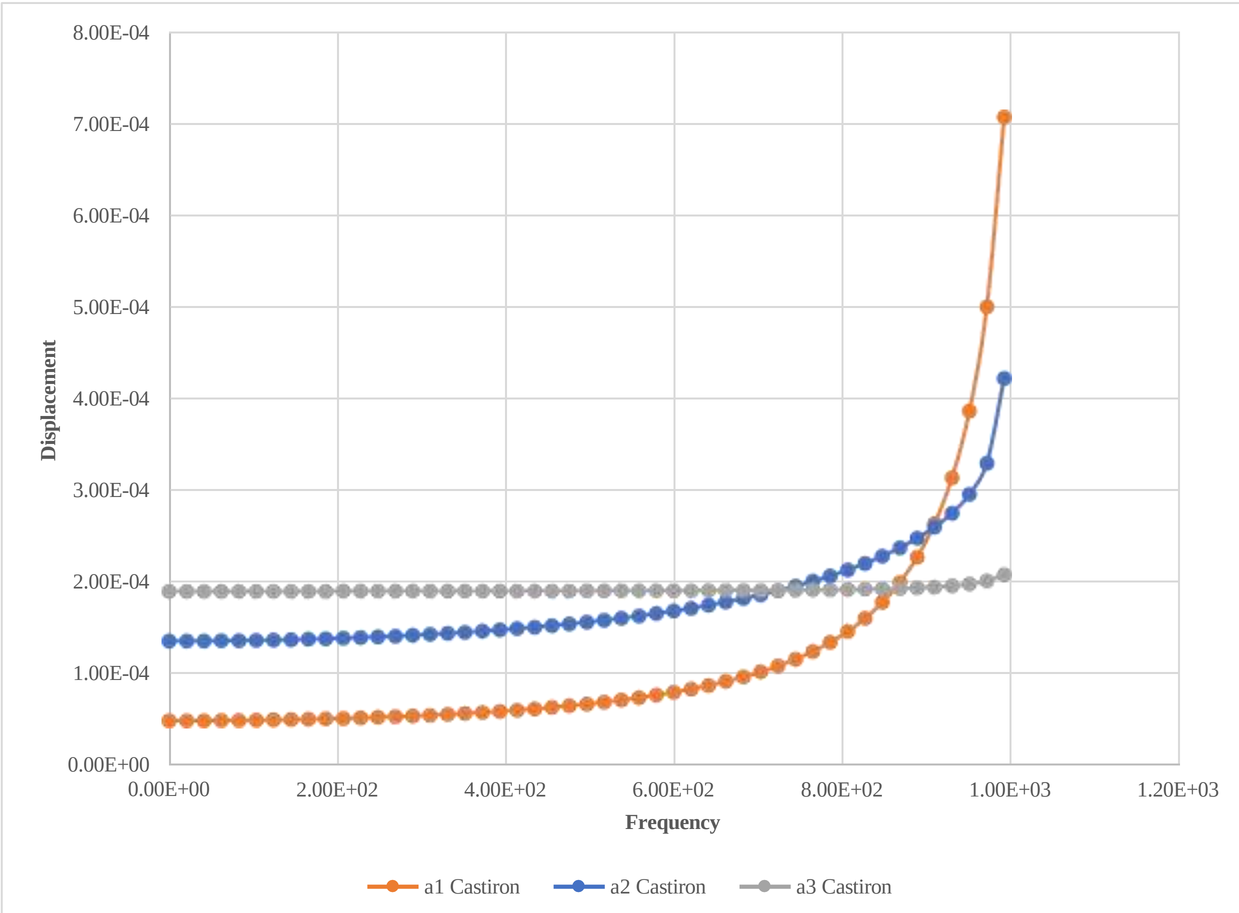


Figure 3.15. Comparison of displacement results for A1, A2, and A3 using CastIron material for Ventilated Brake Disc.

b. ALSiC20%.

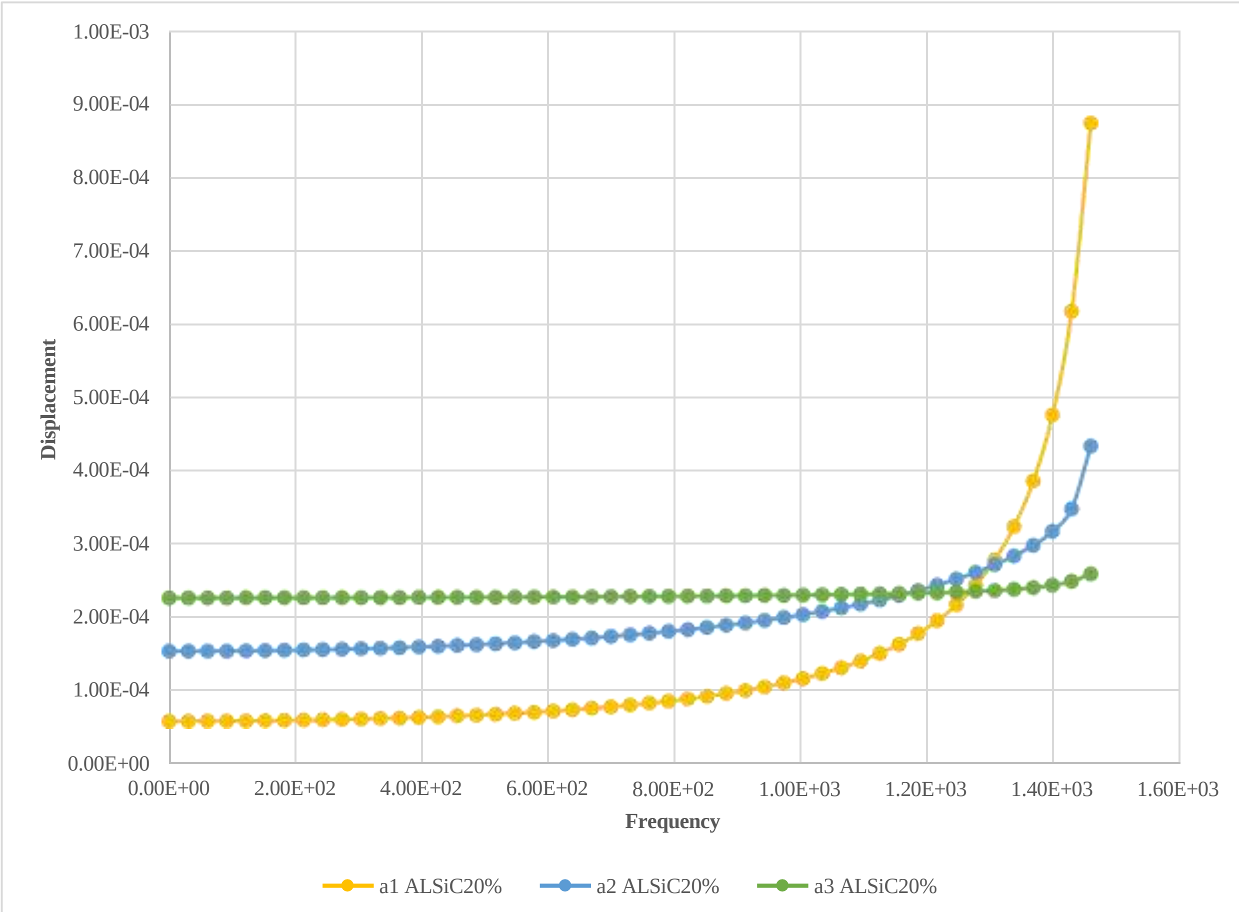


Figure 3.16. Comparison of displacement results for A1, A2, and A3 using ALSiC20% material for Ventilated Brake Disc.

c. AISI1650.

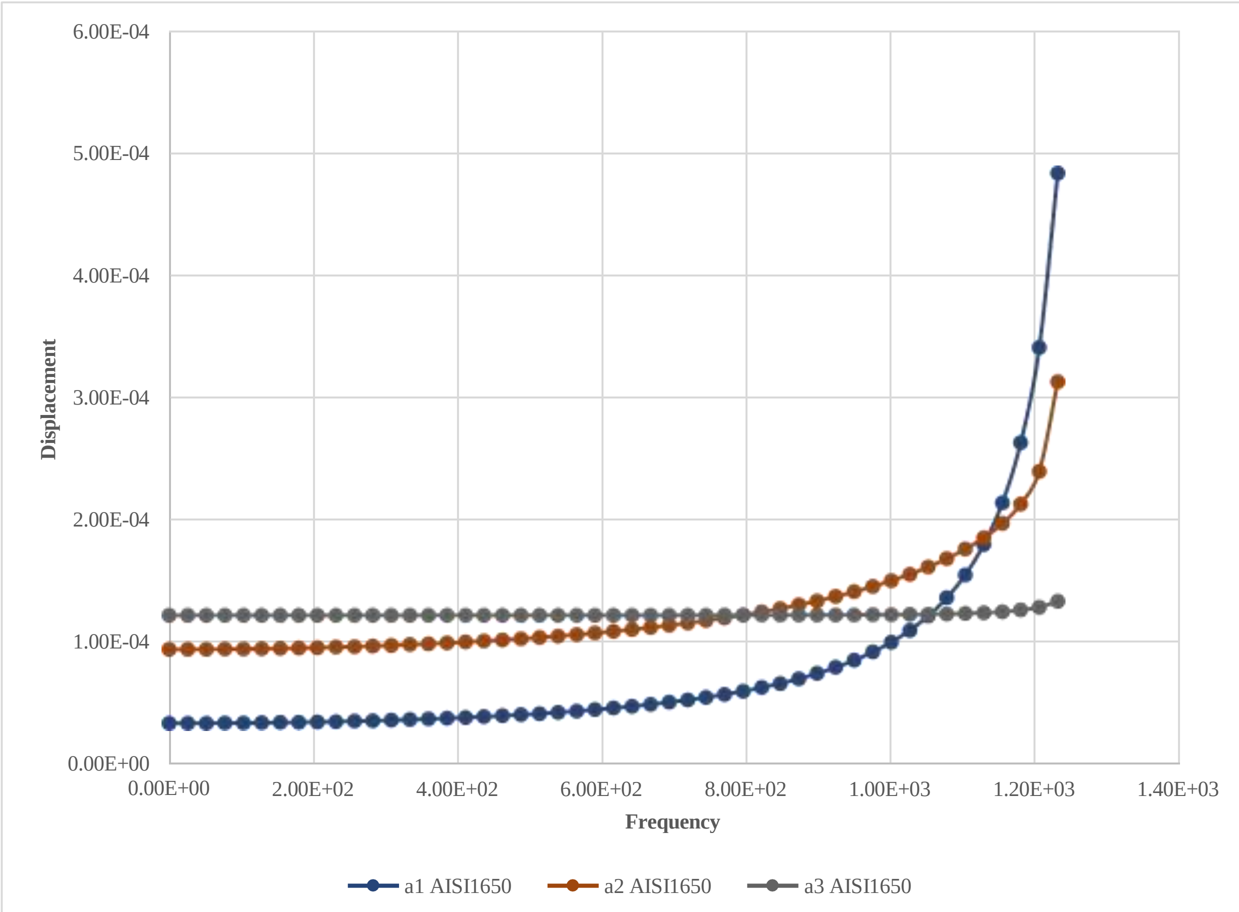


Figure 3.17. Comparison of displacement results for A1, A2, and A3 using AISI1650 material for Ventilated Brake Disc.

Comparison among three types of materials.

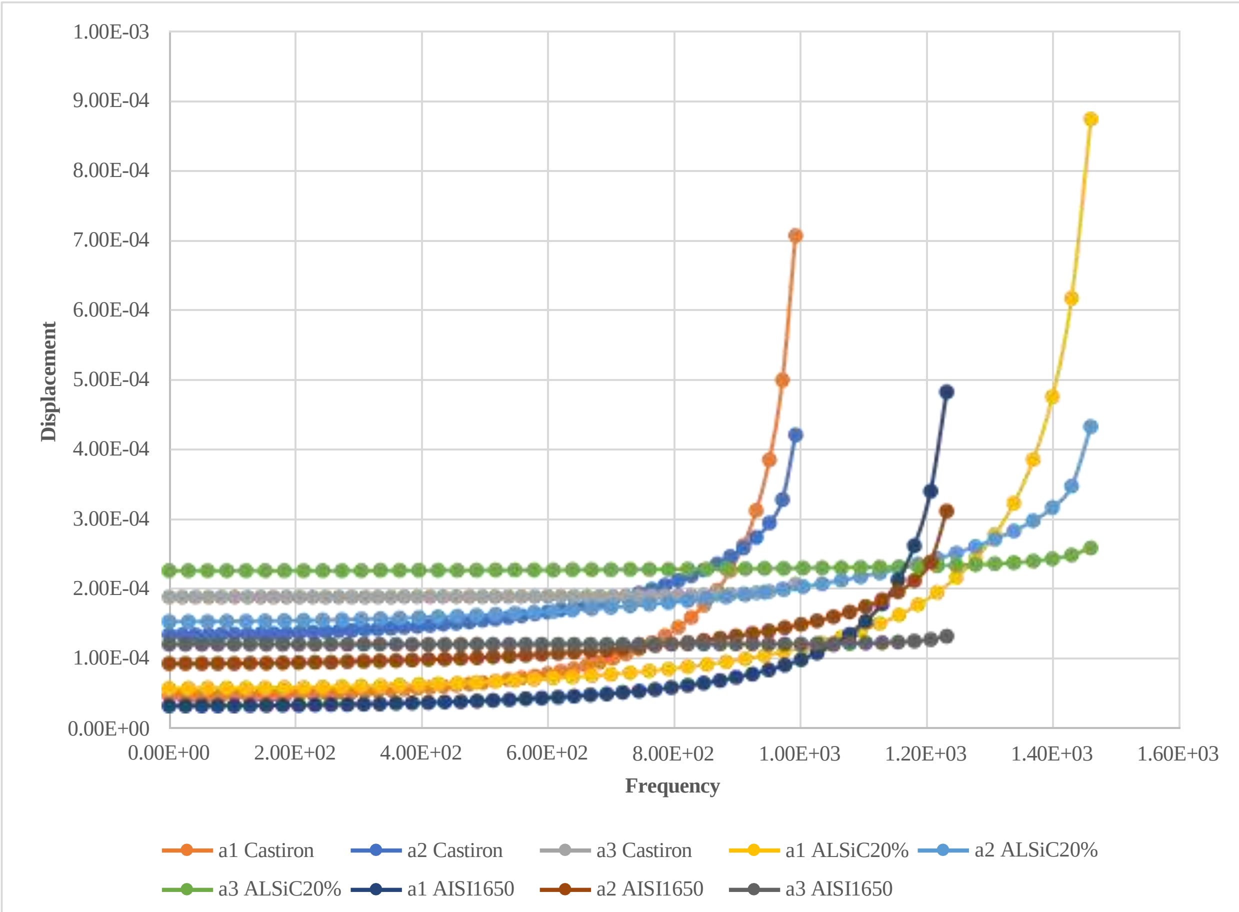


Figure 3.18. Comparison of displacement results for A1, A2 and A3 using 3 different materials for Ventilated Brake Disc.

Evaluation of Results

Similar to the analysis case with the solid disc, when changing the material composition of the brake disc, it is observed that the displacement measurement points at corresponding locations on the model exhibit similar vibration trends across different materials, as shown in Figure 3.15- 3.18. This indicates that the overall vibration pattern of the brake disc is primarily governed by its geometry and boundary conditions, while material changes mainly affect the amplitude and the position of resonance frequencies, without fundamentally altering the “shape” of the vibration modes.

3.4. Comparition Original Brake Disc and Ventilated Brake Disc.

a. CastIron.

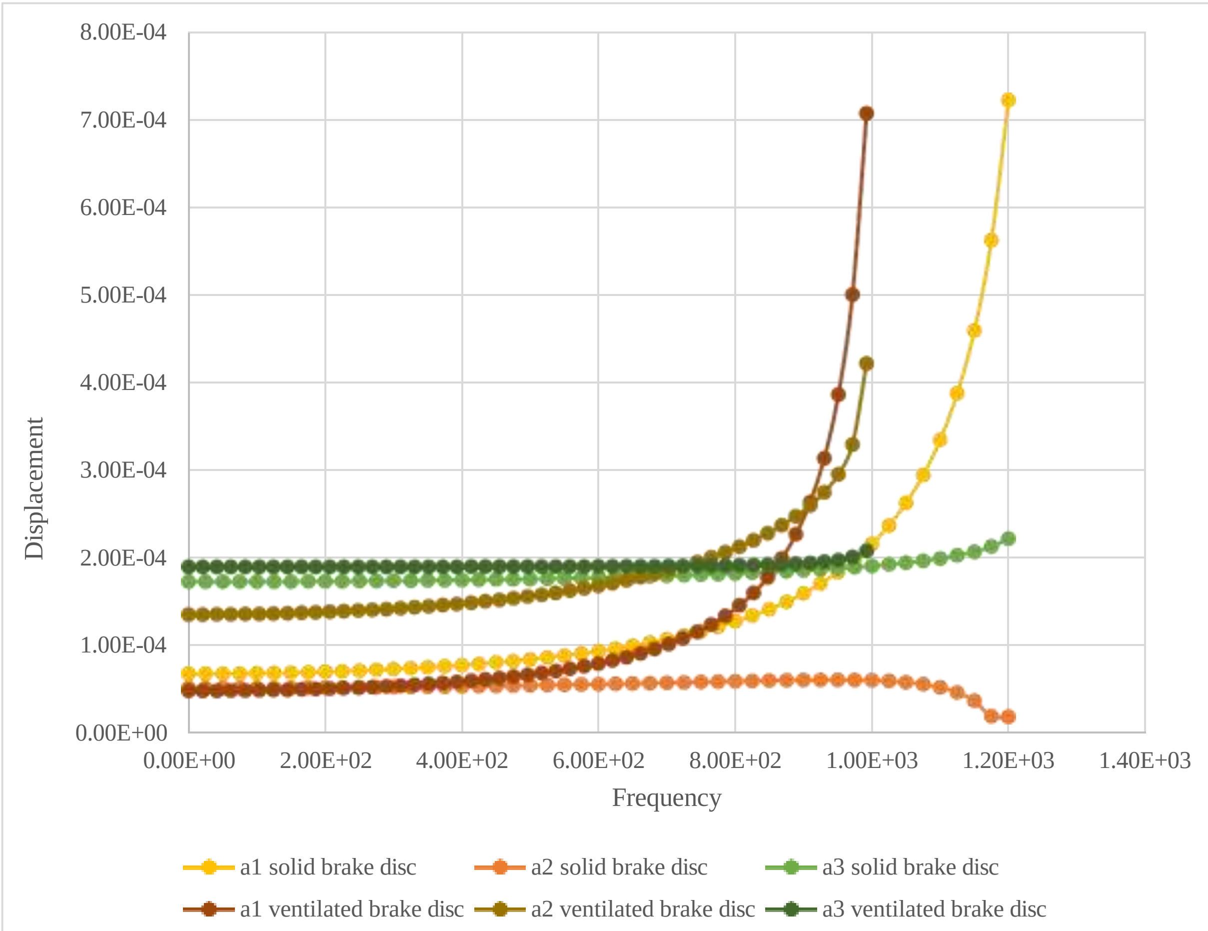


Figure 3.19. Comparison of displacement results for A1, A2 and A3 using CastIron material for Original Brake Disc and Ventilated Brake Disc in 70 km/h case.

b. ALSiC20%.

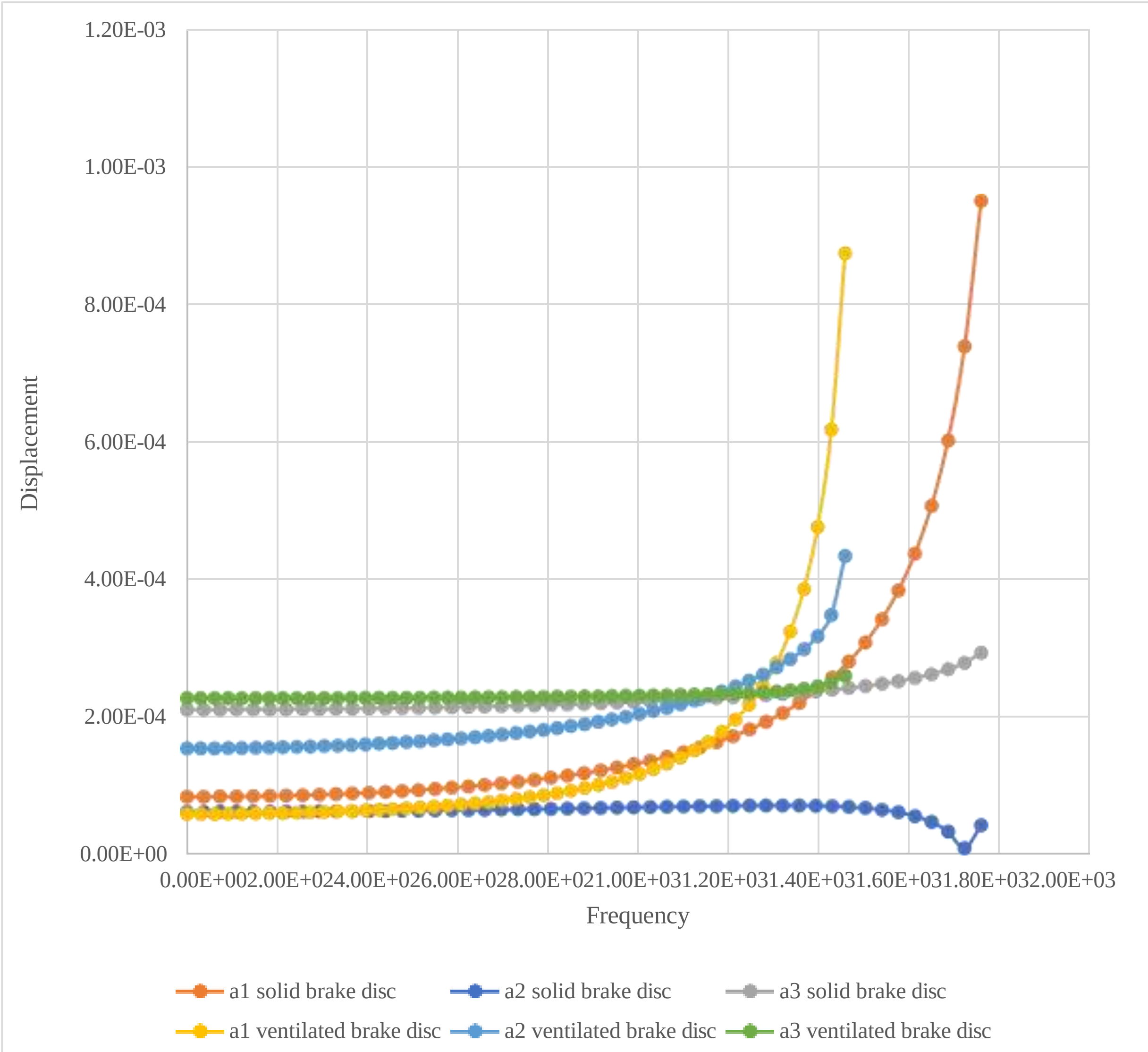


Figure 3.20. Comparison of displacement results for A1, A2 and A3 using ALSiC20% material for Original Brake Disc and Ventilated Brake Disc in 70 km/h case.

c. AISI1650.

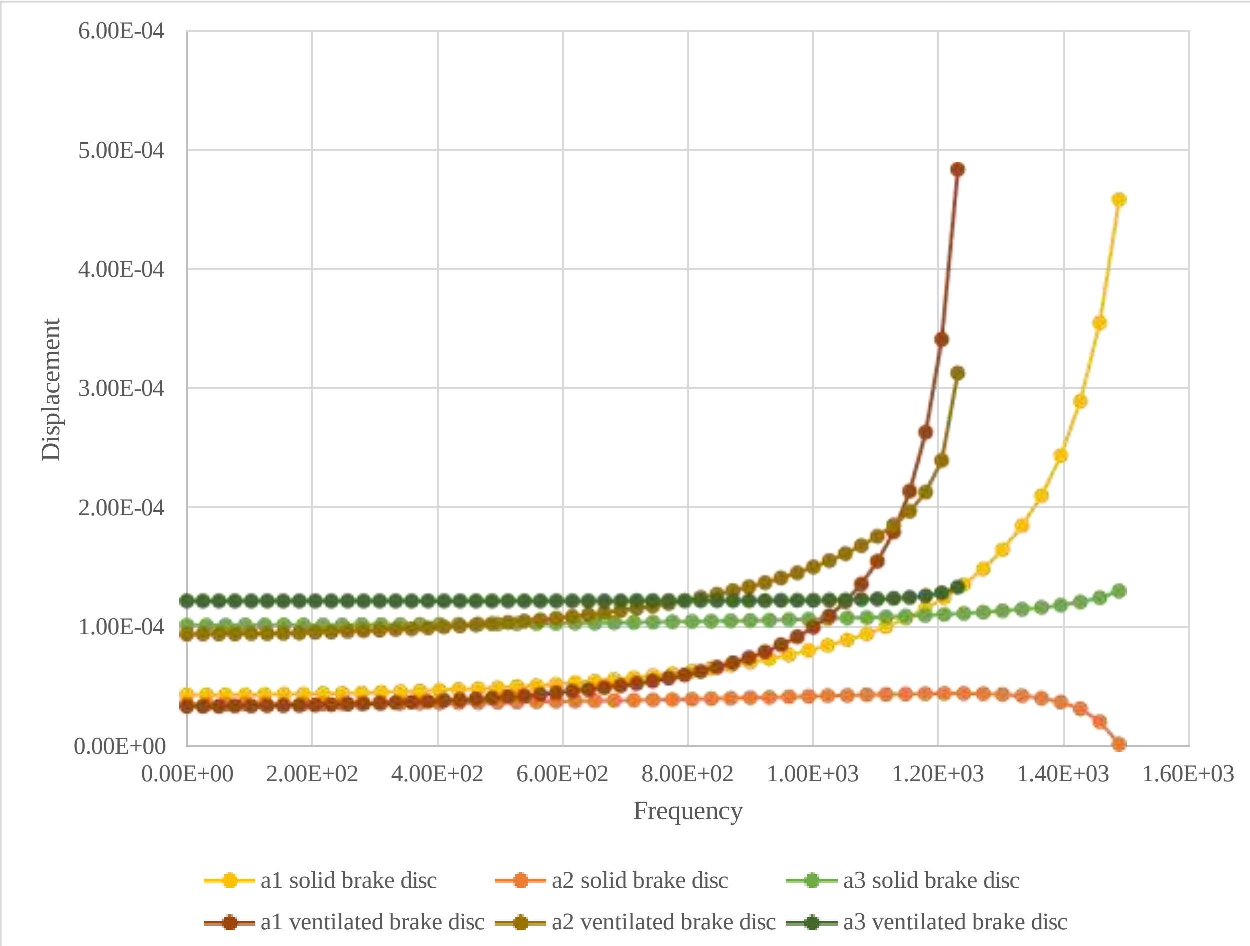


Figure 3.21. Comparison of displacement results for A1, A2 and A3 using AISI1650 material for Original Brake Disc and Ventilated Brake Disc in 70 km/h case.

Based on the frequency response analysis results shown in Figs. 3.19- 3.21, it is clearly observed that the ventilated brake disc exhibits resonance at lower frequencies compared to the solid disc, along with higher vibration amplitudes. This phenomenon is fully consistent with the previously presented theory: when the system's mass decreases, its resistance to vibration diminishes, leading to increased vibration amplitudes and earlier occurrence of resonance. This further reinforces the hypothesis that mass is a critical factor influencing the vibrational behavior of the system.

However, when comparing the displacement variation trends at the three characteristic measurement points (A1, A2, A3) between the solid and ventilated brake discs, a notable observation arises. Points A1 and A3 show similar trends across both disc types, whereas point A2 exhibits a distinctly different vibration trend. To explain this phenomenon, reference should be made to Figure 3.6, which illustrates the distribution of measurement points and the geometric features of the discs.

From observations in Figure 3.6, point A2 lies along the axis passing through the cooling holes. This leads to differences in how vibrational energy propagates and distributes in this area. The cooling holes not only reduce local stiffness around the region but also introduce discontinuities in the disc's continuous structure, altering the stress field and the vibration transmission paths. Therefore, points located on or near the axis through these holes display vibration characteristics markedly different from those of the solid disc, which has a uniform and uninterrupted structure. It can thus be concluded that the geometry of the brake disc, particularly the position and number of cooling holes, plays a significant role in defining the vibration mode shapes, this observation is consistent with the findings reported in [6], [10].

3.5. Proposed Technical Solution and Development Directions

3.5.1. Proposed Technical Solutions

Based on the results presented above, it can be concluded that there is no perfect choice that simultaneously optimizes fuel efficiency, material processing costs, and ensures effective vibration and noise reduction quality. Therefore, trade-offs must be made to prioritize factors suitable for different production and design objectives.

To select the optimal brake disc design for minimizing vibration and noise, the recommendation is to choose a solid disc made from materials with high stiffness and large mass, while limiting its use under harsh operating conditions to ensure durability and effective noise reduction during braking.

Specifically, if the priority is to reduce vibration and limit noise generation, the ideal design solution is to use a solid brake disc combined with high-stiffness, high-mass materials. This configuration enhances the vibration stability of the disc, reduces the risk of resonance, and limits displacement amplitudes under braking forces, thereby significantly lowering noise during operation. However, this advantage comes with the drawback of increased system weight, which can adversely affect fuel efficiency and driving dynamics, especially in vehicles demanding lightweight and high performance.

Conversely, ventilated discs are commonly employed to improve heat dissipation, reduce mass, and cut material costs. Nonetheless, these designs are more prone to vibrations due to overall reduced stiffness, especially in higher vibration modes. This requires careful consideration of the vehicle's specific operating conditions; for example, high-intensity, continuous braking systems in sports or heavy-duty vehicles prioritize heat dissipation, while passenger cars typically value comfort and noise reduction as primary criteria.

Therefore, the optimal solution should be determined based on specific design goals: if the main objective is vibration reduction and noise limitation to enhance user comfort, the appropriate choice is a solid disc made of stiff, heavy materials, with restricted operation in harsh conditions to maintain durability and long-term vibration mitigation effectiveness. Meanwhile, if balancing performance, weight, and cooling is necessary, advanced alternatives such as lightweight but stiff composite materials or ventilated discs with optimized cooling hole geometry can be considered to minimize adverse impacts on vibration characteristics.

In summary, selecting a brake disc design must be considered within the overall context of the vehicle system, usage characteristics, and specific technical requirements. There is no one-size-fits-all solution; rather, flexible design approaches and multi-criteria evaluations are essential to achieving the highest technical and economic effectiveness.

3.5.2. Development Directions

Based on the current analysis results, it is evident that the vibration and noise characteristics of the brake disc are influenced by multiple design factors and operating conditions. To enhance the simulation quality and move towards studies that better reflect real-world conditions, several future development directions are proposed as follows:

- Analysis of temperature effects on the vibration characteristics of the brake disc: Under actual operating conditions, the brake disc is significantly affected by temperature due to frictional heat generated during braking. The increase in temperature alters the mechanical properties of the material, such as stiffness, thermal expansion coefficient, and may induce thermal stresses. Therefore, integrating thermal analysis to evaluate the influence of temperature on vibration response will make the model more realistic and accurate.
- Investigation of the brake pad material influence: The brake pad is the direct component in contact with the disc and transmits the braking force. Material properties such as friction coefficient, stiffness, thermal conductivity, and thermal durability of the brake pad can impact the vibration characteristics of the system.
- Analysis of brake pad geometry effects: The contact shape between the brake pad and the disc plays an important role in the distribution of contact pressure and vibration transmission. Various pad geometries—such as curved surfaces, transverse grooves, or longitudinal grooves—can lead to different vibration patterns. Studying and optimizing the brake pad geometry can help improve noise reduction capabilities and enhance braking performance.

CHAPTER 4. CONCLUTIONS

Mass of the Brake Disc has a significant influence on the vibration characteristics. Specifically, when the mass of the disc increases, the system's inertia also increases, which results in a reduction of vibration amplitude as the system becomes more "resistant" to excitation. Conversely, lighter discs exhibit higher vibration amplitudes due to being more easily excited under the same loading conditions. This indicates that optimizing the mass not only affects fuel efficiency but also impacts the vibration stability of the braking system.

Material Hardness of the Brake Disc is a critical factor determining the resistance against resonance phenomena. Materials with higher stiffness tend to reduce vibration amplitude since they deform less under external forces, and their resonance frequencies shift to higher values, making them less likely to be excited within the normal operating frequency range. In contrast, materials with lower stiffness exhibit resonance at lower frequencies, increasing the risk of excessive vibrations that can affect braking performance and system durability.

Operating Speed (Applied Braking Force) primarily affects the vibration amplitude rather than the inherent vibration characteristics. When the braking force (torque) increases, the energy input into the system also rises, causing larger vibration amplitudes. Conversely, lower braking forces result in smaller amplitudes. However, the mode shapes and natural frequencies remain unchanged, as these are determined by the system's geometry and material properties.

Displacement Distribution of Points on the Disc shows consistency across different materials for the same mode shape, assuming geometry remains constant. This demonstrates the dominant influence of geometry on mode shapes, while the material properties mainly affect vibration amplitude and frequency

Mode Shape is directly influenced by the system's resonance modes. Each mode is characterized by its natural frequency and associated deformation pattern, depending on geometry, boundary conditions, and material distribution. Resonance occurs when excitation frequencies match or approach these natural frequencies, causing vibration amplitudes to spike and potentially compromising the brake disc's stability and lifespan.

Geometric Shape of the Brake Disc, particularly designs with ventilation holes, also impacts vibration behavior. Compared to solid discs, ventilated discs generally have reduced overall stiffness due to material removal, leading to higher vibration amplitudes near the holes. Moreover, points located along axes intersecting the ventilation holes tend to exhibit different vibration characteristics, creating localized non-uniform vibration patterns. Nevertheless, aside from these localized effects, the overall mode shapes remain largely similar to those of solid discs in other regions.

In Conclusion, the three main factors—mass, material, and geometric design—each play crucial roles in influencing vibration characteristics and the risk of resonance in brake systems. Optimizing these factors simultaneously is essential to ensure effective vibration control, durability, and operational stability of brake discs under real-world conditions.

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