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DYNAMIC BEHAVIOR AND SQUEAL PREDICTION OF A DRUM BRAKE SYSTEM USING ANSYS MECHANICAL

Mrs. K.Tulasi[1]
D.Aadarsh[2]**, T.Mukesh Sathwik**[3]**, K.Mohan Sai**[4]**, T.Naga Durga Eswara Suresh**[5]

[1]Assistant Professor
Department of Mechanical Engineering
Pragati Engineering College (Autonomous)
(Affiliated to JNTUK)
Kakinada District, Surampalem -533437

[2,3,4,5]Students, Department of Mechanical Engineering
Pragati Engineering College (Autonomous)
(Affiliated to JNTUK)
Kakinada District, Surampalem -533437
Mail.Id: tulasi.k@pragati.ac.in[1], Mail.Id: dasariaadarsh1@gmail.com[2]
Mail.Id: mukeshsathwiknaidu17@gmail.com[3], Mail.Id: Kariimohansai5259@gmail.com[4]
Mail.Id: sureshtarun7@gmail.com[5]

Abstract

Brake squeal is a friction-induced vibration phenomenon commonly observed in automotive braking systems and represents a major noise, vibration, and harshness (NVH) concern. Although it does not directly affect braking performance, squeal significantly impacts passenger comfort and perceived vehicle quality. This study presents a finite element–based investigation of the dynamic behavior and squeal propensity of a brake disc–pad assembly using complex eigenvalue analysis. A three-dimensional model of the brake system is developed, incorporating frictional contact, prestress effects, and realistic boundary conditions. Linear non-prestressed modal analysis and full nonlinear perturbed modal analysis are performed and compared to evaluate the influence of friction-induced coupling on system stability. The results reveal the presence of unstable vibration modes in the high-frequency range, indicating potential squeal occurrence. Mode shape analysis confirms that squeal originates from coupling between disc and pad bending modes. The study demonstrates that finite element–based complex eigenvalue analysis is an effective and efficient tool for early-stage brake squeal prediction and NVH-oriented brake system design.

Keywords: Brake squeal, Friction-induced vibration, Complex eigenvalue analysis, Finite element method, NVH, Disc brake

1. Introduction

Brake squeal is a persistent noise problem encountered in automotive and railway braking systems, typically manifesting as a high-pitched tonal sound in the frequency range above 1 kHz. While squeal does not compromise braking effectiveness or safety, it has a strong influence on customer satisfaction and perceived product quality. Consequently, brake squeal has become a critical design concern in modern vehicle development, particularly as noise regulations and NVH expectations continue to tighten.

From a physical perspective, brake squeal is classified as a self-excited vibration caused by frictional interaction between the brake disc and pads. During braking, tangential friction forces generated at the sliding interface may inject energy into certain structural vibration modes rather than dissipating it. When this energy input exceeds the system's inherent damping, dynamic instability occurs, resulting in sustained oscillations and audible noise. This behavior distinguishes squeal from forced vibration phenomena and makes it highly sensitive to small variations in system parameters such as friction coefficient, contact stiffness, material properties, and boundary conditions.

The brake assembly consists of multiple flexible components, including the disc, pads, caliper, and supporting structures, each possessing closely spaced natural frequencies and complex mode shapes. Under frictional contact, these modes may interact and couple, leading to instability. The presence of contact nonlinearity, sliding friction, and prestress further complicates analytical treatment of the problem. As a result, traditional analytical models and simplified lumped-parameter approaches are often insufficient to accurately predict brake squeal in practical systems.

Historically, brake squeal evaluation has relied on experimental testing methods such as dynamometer tests and vehicle road tests. Although experiments provide valuable validation data, they are expensive, time-consuming, and typically performed late in the design cycle, when design modifications are costly. To overcome these limitations, numerical simulation techniques based on the finite element method (FEM) have gained widespread acceptance for brake squeal prediction during early design stages.**Significance of Numerical Brake Squeal Prediction**With increasing regulatory pressure on vehicle noise emissions and rising customer expectations for refinement, the ability to predict and mitigate brake squeal at the design stage has become critically important. Noise-related complaints constitute a significant portion of warranty claims in the automotive industry, and brake squeal is one of the most difficult NVH issues to resolve once a system reaches production.**Challenges in Brake Squeal Modeling**Despite significant advancements in computational methods, accurate brake squeal prediction remains a challenging task due to the inherent complexity of the problem. Brake systems exhibit strong sensitivity to operating conditions such as braking pressure, sliding speed, temperature, and wear state. Small variations in friction coefficient or contact stiffness can lead to substantial changes in system stability. Furthermore, friction behavior itself is uncertain and often depends on temperature, surface roughness, humidity, and material aging, which are difficult to represent precisely in numerical models.

Objectives of the Present Study

The specific objectives of this study are as follows:

- To explain the fundamental physical mechanisms responsible for brake squeal, with emphasis on friction-induced mode coupling.
- To present the theoretical background of complex eigenvalue analysis as applied to brake systems.
- To demonstrate the implementation of brake squeal prediction using ANSYS Mechanical, following the Technology Showcase example.
- To compare linear non-prestressed modal analysis and full nonlinear perturbed modal analysis in terms of accuracy, physical realism, and computational effort.
- To identify unstable vibration modes and interpret their contribution to squeal propensity.
- To highlight the practical relevance of FEM-based squeal analysis for brake system design optimization and NVH improvement.
- **Scope and Limitations of the Study** The scope of the present work is limited to frequency-domain stability analysis using complex eigenvalue methods. Time-domain transient simulations of squeal, which can capture nonlinear limit-cycle oscillations and amplitude growth, are not considered due to their high computational cost. This report is based on an ANSYS Mechanical Technology Showcase example problem on brake squeal analysis from Release 2025 R2, which demonstrates how commercial FE software can be employed to predict and analyze squeal in a realistic disc–pad system. The example highlights the use of advanced contact elements, frictional interfaces, and complex eigenvalue extraction techniques to identify unstable vibration modes. Two numerical approaches are demonstrated:

2. Literature Review

Brake squeal has been extensively investigated due to its complex physical nature and strong impact on noise, vibration, and harshness (NVH) performance. Early research focused on experimental observations, identifying squeal as a high-frequency tonal noise occurring during steady sliding between brake pads and discs. These studies established that squeal is not caused by external excitation but arises from internal dynamic instability within the brake system.

Initial theoretical explanations were based on stick–slip mechanisms, where differences between static and kinetic friction coefficients were assumed to generate oscillatory motion. While this theory successfully explains low-frequency friction noise such as chatter and groan, it has been shown to be insufficient for predicting high-frequency brake squeal observed in modern braking systems.

Subsequent research introduced the mode-coupling theory, which is now widely accepted as the dominant squeal mechanism. According to this theory, frictional contact couples two or more structural vibration modes with closely spaced natural frequencies, leading to instability. Experimental and numerical studies have demonstrated that disc out-of-plane bending modes often couple with pad bending or torsional modes, resulting in squeal frequencies typically in the range of 5–15 kHz.

With the advancement of computational power, finite element–based methods have become central to brake squeal prediction. Complex eigenvalue analysis (CEA) has been widely adopted to identify unstable modes by solving the unsymmetric eigenvalue problem arising from friction-induced stiffness coupling. Numerous studies report good correlation between CEA predictions and experimental measurements, particularly in identifying squeal-prone frequency ranges and dominant mode shapes.

Recent research has focused on improving prediction accuracy through enhanced contact modeling, inclusion of prestress effects, realistic boundary conditions, and damping characterization. Parametric studies have shown that small variations in friction coefficient, pad stiffness, and disc geometry can significantly alter squeal propensity, explaining the intermittent nature of squeal in real-world braking systems. Overall, the literature confirms that FEM-based CEA is an effective and practical tool for early-stage brake squeal prediction and design optimization.

3. Theoretical Background

3.1 Friction-Induced Vibration Mechanism

Brake squeal is a self-excited vibration phenomenon caused by frictional interaction at the pad–disc interface. Unlike forced vibrations, which require external excitation, self-excited vibrations arise when internal forces inject energy into the system. In braking systems, friction forces may act as a negative damping source under certain conditions, causing vibration amplitudes to grow with time.

When the brake pad slides against the rotating disc, small perturbations in relative motion generate frictional forces proportional to the normal contact load. These forces interact with the elastic deformation of the disc and pad, potentially feeding energy into specific structural modes. If the energy input exceeds the inherent material and structural damping, dynamic instability occurs, resulting in squeal.

3.2 Mode-Coupling Theory

Mode-coupling theory explains brake squeal as the interaction between two or more vibration modes with similar natural frequencies. Under frictional contact, tangential friction forces couple normal and tangential motions at the interface, introducing unsymmetric terms into the system stiffness matrix.

As friction increases, two real eigenvalues associated with distinct modes may coalesce and split into a complex conjugate pair. When the real part of an eigenvalue becomes positive, the system becomes dynamically unstable. The imaginary part of the eigenvalue corresponds to the squeal frequency, while the positive real part represents the growth rate of the vibration.

This theory successfully explains key characteristics of brake squeal, including:

- High-frequency tonal noise
- Occurrence under steady sliding conditions
- Sensitivity to small design and operating parameter changes

4. Complex Eigenvalue Analysis

The dynamic behavior of a brake system with frictional contact can be described by the linearized equation of motion:

[M]u¨+[C]u˙+([K]+[Kf])u=0[M]\ddot{u} + [C]\dot{u} + ([K] + [K_f])u = 0[M]u¨+[C]u˙+([K]+[Kf])u=0

where
[M][M][M] is the mass matrix,

[C][C][C] is the damping matrix,

[K][K][K] is the structural stiffness matrix, and

[Kf][K_f][Kf] represents friction-induced stiffness terms.

Due to frictional coupling, the global stiffness matrix becomes unsymmetric, leading to a complex eigenvalue problem. The solution yields eigenvalues of the form:

λ=α±iω\lambda = \alpha \pm i\omegaλ=α±iω

Here, ω\omegaω represents the vibration frequency, and α\alphaα represents the stability characteristic. A positive value of α\alphaα indicates an unstable mode and potential squeal occurrence.

Complex eigenvalue analysis provides a direct and computationally efficient method to identify squeal-prone modes and to evaluate the influence of design parameters such as friction coefficient, contact stiffness, and component geometry.

5. Finite Element Modeling

5.1 Geometry and Material Properties

A three-dimensional finite element model of a brake disc–pad assembly is developed to capture the essential dynamics responsible for brake squeal. The brake disc is modeled as an annular solid rotor with an outer diameter of 350 mm, an inner diameter of 250 mm, and a thickness of 10 mm. Two brake pads, each with a thickness of 15 mm, are positioned symmetrically on either side of the disc to represent a simplified disc brake configuration.

All components are assumed to be linear elastic, homogeneous, and isotropic. Steel-like material properties are used, with a Young's modulus of 200 GPa, Poisson's ratio of 0.3, and density of 7800 kg/m^3. These properties are sufficient to accurately represent stiffness and mass distribution governing modal behavior.

5.2 Contact Modeling and Boundary Conditions

Surface-to-surface frictional contact is defined between the disc and pad interfaces with a constant coefficient of friction. Normal contact pressure is applied to the pad surfaces to simulate braking force. The disc hub region is fully constrained to represent attachment to the wheel hub, while rotational sliding of the disc is imposed to activate frictional coupling.

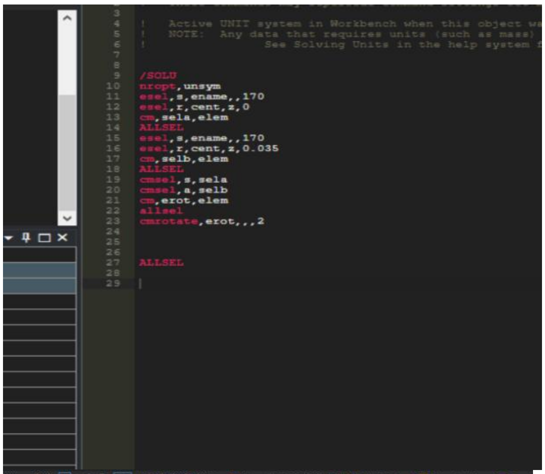
5.3 Meshing Strategy

Quadratic three-dimensional solid elements are used for all components. Mesh refinement is applied in the pad–disc contact region to accurately capture contact pressure distribution and friction-induced stiffness effects. The final model achieves a balance between computational efficiency and accuracy required for high-frequency modal analysis.

5.4 Solution Approach

Two solution strategies are adopted:

1. **Linear non-prestressed modal analysis**, neglecting contact prestress.
2. **Full nonlinear perturbed modal analysis**, including contact pressure and frictional prestress prior to eigenvalue extraction.



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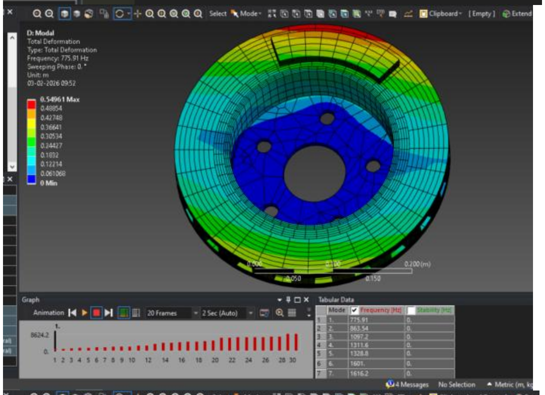

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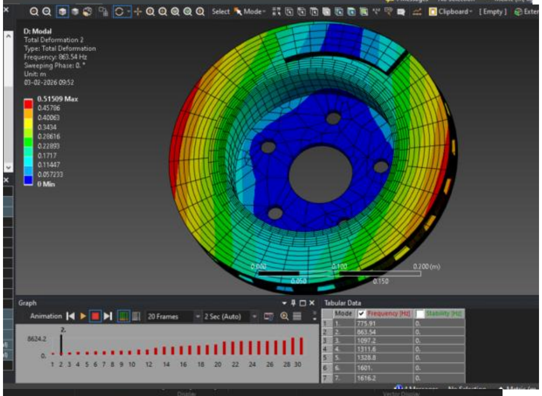
1	1.	775.91	0.
2	2.	863.54	0.
3	3.	1097.2	0.
4	4.	1311.6	0.
5	5.	1328.8	0.
6	6.	1601.	0.
7	7.	1616.2	0.



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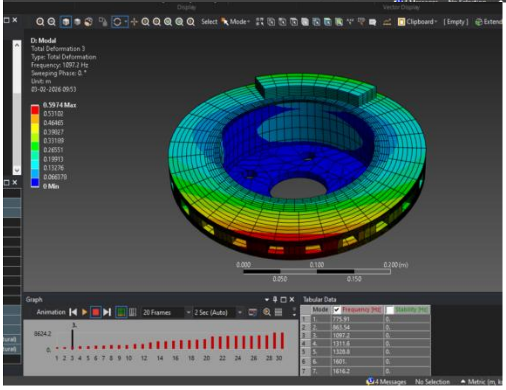
2	2.	863.54	0.
3	3.	1097.2	0.
4	4.	1311.6	0.
5	5.	1328.8	0.
6	6.	1601.	0.
7	7.	1616.2	0.



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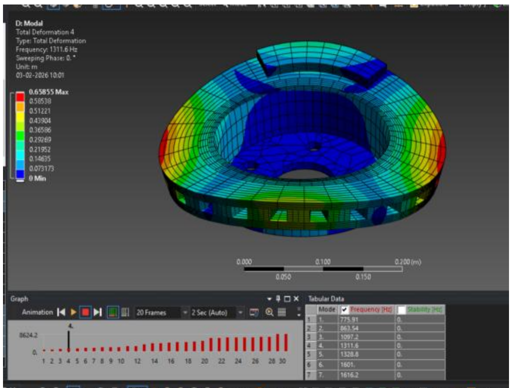
	Mode	☑ Frequency [Hz]	☐ Stability [Hz]
1	1.	775.91	0.
2	2.	863.54	0.
3	3.	1097.2	0.
4	4.	1311.6	0.
5	5.	1328.8	0.
6	6.	1601.	0.
7	7.	1616.2	0.



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Type: Total Deformation
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Tabular Data

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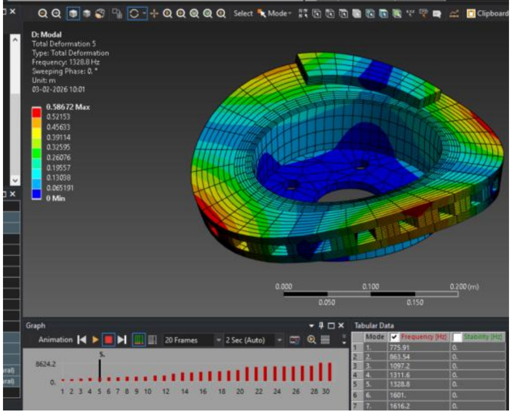
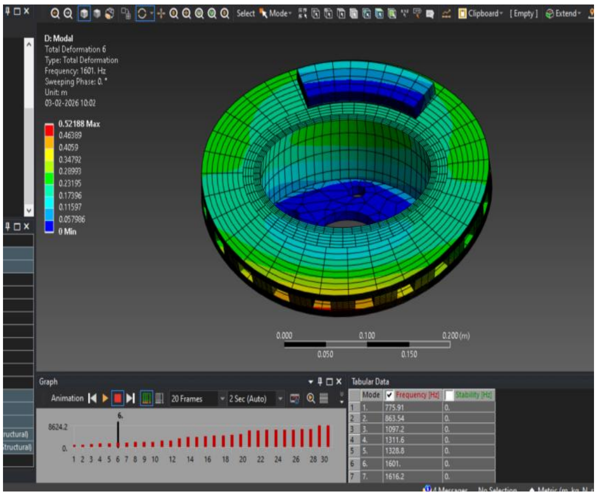
2	2.	863.54	0.
3	3.	1097.2	0.
4	4.	1311.6	0.
5	5.	1328.8	0.
6	6.	1601.	0.
7	7.	1616.2	0.



D: Modal
Total Deformation 5
Type: Total Deformation
Frequency: 1328.8 Hz
Sweeping Phase: 0. °
Unit: m
03-02-2026 10:01

Tabular Data

	Mode	☑ Frequency [Hz]	☐ Stability [Hz]
1	1.	775.91	0.
2	2.	863.54	0.
3	3.	1097.2	0.
4	4.	1311.6	0.
5	5.	1328.8	0.
6	6.	1601.	0.
7	7.	1616.2	0.



D: Modal
Total Deformation 6
Type: Total Deformation
Frequency: 1601. Hz
Sweeping Phase: 0. °
Unit: m
03-02-2026 10:02

Tabular Data

	Mode	☑ Frequency [Hz]	☐ Stability [Hz]
1	1.	775.91	0.
2	2.	863.54	0.
3	3.	1097.2	0.
4	4.	1311.6	0.
5	5.	1328.8	0.
6	6.	1601.	0.
7	7.	1616.2	0.

Complex eigenvalue analysis is performed in **ANSYS Mechanical** to identify unstable vibration modes.

6. Results and Discussion

The complex eigenvalue analysis reveals unstable modes in the high-frequency range, indicating the potential onset of brake squeal.

6.1 Linear Non-Prestressed Analysis

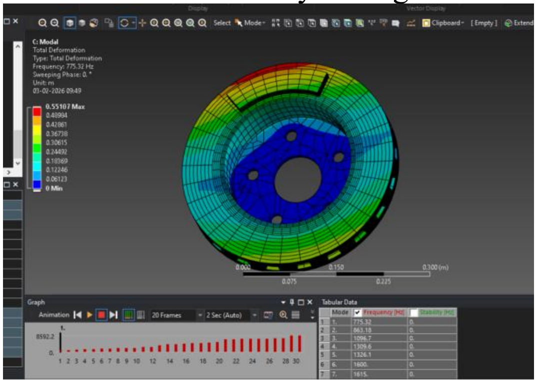
The linear analysis predicts unstable modes near 6.47 kHz, characterized by positive real parts of the eigenvalues. This approach provides rapid identification of squeal-prone frequencies but neglects contact-induced prestress effects.

6.2 Nonlinear Perturbed Modal Analysis

The nonlinear analysis predicts unstable modes at approximately 6.46 kHz. The slight reduction in frequency compared to the linear case is attributed to contact prestress and frictional stiffness effects. The presence of positive real eigenvalues confirms friction-induced dynamic instability.

6.3 Mode Shape Interpretation

The unstable modes exhibit strong coupling between disc out-of-plane bending modes and pad deformation modes. Large deformation amplitudes are concentrated near the pad–disc contact region, consistent with mode-coupling theory. These findings explain the generation of high-frequency tonal noise under steady sliding conditions.



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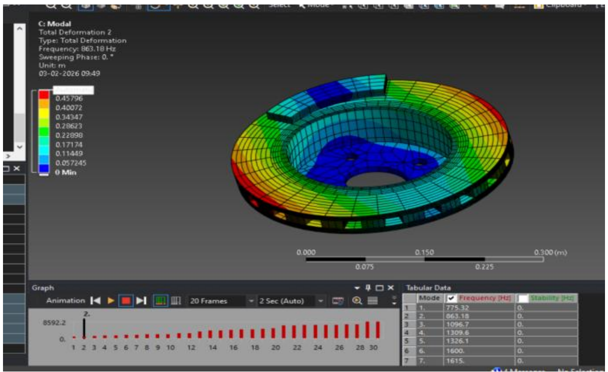
Tabular Data

	Mode	☑ Frequency [Hz]	☐ Stability [Hz]
1	1.	775.32	0.
2	2.	863.18	0.
3	3.	1096.7	0.
4	4.	1309.6	0.
5	5.	1326.1	0.
6	6.	1600.	0.
7	7.	1615.	0.



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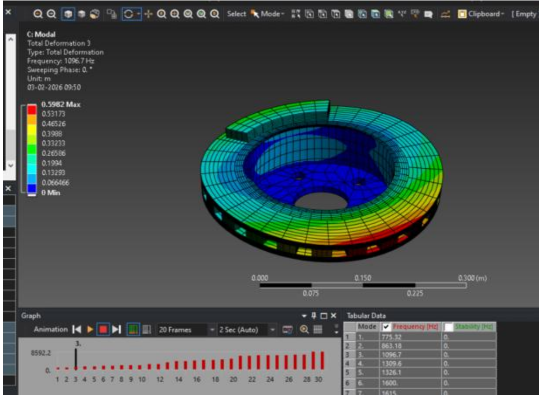
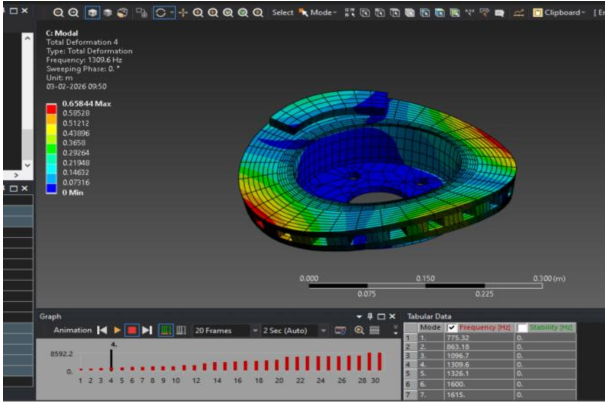
Type: Total Deformation
Frequency: 863.18 Hz
Sweeping Phase: 0. °
Unit: m
03-02-2026 09:49

	Mode	Frequency [Hz]	Stability [Hz]
1	1.	775.32	0.
2	2.	863.18	0.
3	3.	1096.7	0.
4	4.	1309.6	0.
5	5.	1326.1	0.
6	6.	1600.	0.
7	7.	1615.	0.



C: Modal
Total Deformation 3
Type: Total Deformation
Frequency: 1096.7 Hz
Sweeping Phase: 0. °
Unit: m
03-02-2026 09:50

	Mode	Frequency [Hz]	Stability [Hz]
1	1.	775.32	0.
2	2.	863.18	0.
3	3.	1096.7	0.
4	4.	1309.6	0.
5	5.	1326.1	0.
6	6.	1600.	0.
7	7.	1615.	0.

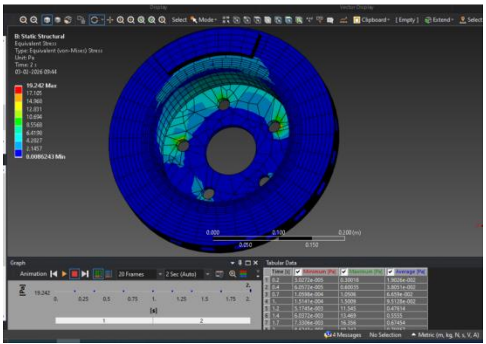


C: Modal
Total Deformation 4
Type: Total Deformation
Frequency: 1309.6 Hz
Sweeping Phase: 0. °
Unit: m
03-02-2026 09:50

	Mode	Frequency [Hz]	Stability [Hz]
1	1.	775.32	0.
2	2.	863.18	0.
3	3.	1096.7	0.
4	4.	1309.6	0.
5	5.	1326.1	0.
6	6.	1600.	0.
7	7.	1615.	0.

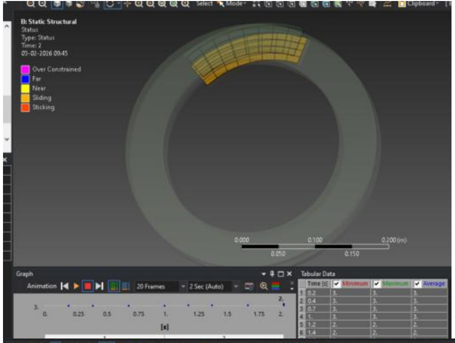
6.4 Comparison of Approaches

Both analysis methods predict similar squeal frequencies, with less than 1% deviation. This indicates that, for the present configuration, structural stiffness dominates system behavior and linear analysis can be effective for preliminary squeal screening, while nonlinear analysis provides improved physical realism.



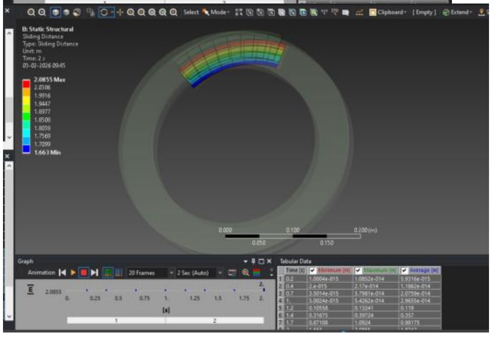
B: Static Structural
Equivalent Stress
Type: Equivalent (von-Mises) Stress
Unit: Pa
Time: 2 s
03-02-2026 09:44

	Time [s]	Minimum [Pa]	Maximum [Pa]	Average [Pa]
1	0.2	3.0272e-005	0.30018	1.9026e-002
2	0.4	6.0572e-005	0.60035	3.8051e-002
3	0.7	1.0598e-004	1.0508	6.659e-002
4	1.	1.5141e-004	1.5009	9.5128e-002
5	1.2	5.1745e-003	11.545	0.47614
6	1.4	6.0372e-003	13.469	0.5555
7	1.7	7.3306e-003	16.356	0.67454



B: Static Structural
Status
Type: Status
Time: 2
03-02-2026 09:45

	Time [s]	Minimum	Maximum	Average
1	0.2	3.	3.	3.
2	0.4	3.	3.	3.
3	0.7	3.	3.	3.
4	1.	3.	3.	3.
5	1.2	2.	2.	2.
6	1.4	2.	2.	2.



B: Static Structural
Sliding Distance
Type: Sliding Distance
Unit: m
Time: 2 s
03-02-2026 09:45

	Time [s]	Minimum [m]	Maximum [m]	Average [m]
1	0.2	1.0004e-015	1.0852e-014	5.9316e-015
2	0.4	2.e-015	2.17e-014	1.1862e-014
3	0.7	3.5014e-015	3.7981e-014	2.0759e-014
4	1.	5.0024e-015	5.4262e-014	2.9655e-014
5	1.2	0.10558	0.13241	0.119
6	1.4	0.31675	0.39724	0.357
7	1.7	0.87108	1.0924	0.98175

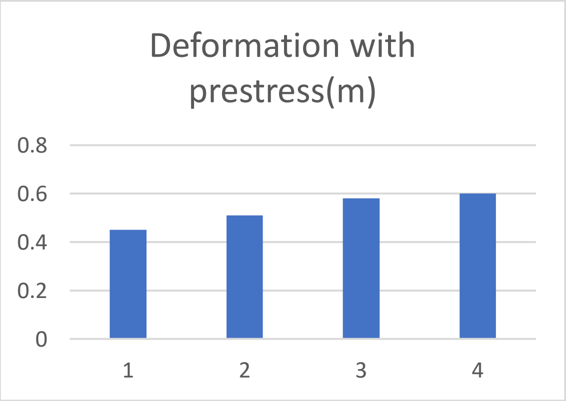
7. Conclusion

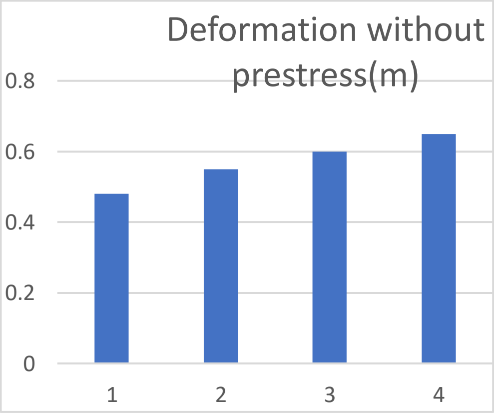
This study presents a finite element–based investigation of brake squeal using complex eigenvalue analysis. A simplified brake disc–pad assembly was modeled to evaluate friction-induced dynamic instability under sliding conditions. Both linear non-prestressed and nonlinear perturbed modal analyses successfully predicted unstable vibration modes in the high-frequency range.

The results confirm that brake squeal arises from friction-induced mode coupling between disc and pad vibration modes. The close agreement between linear and nonlinear analyses suggests that linear CEA can be efficiently used during early design stages, while nonlinear analysis is better suited for detailed validation. The methodology demonstrated in this work provides a robust and computationally efficient framework for brake squeal prediction and NVH-oriented brake system design.

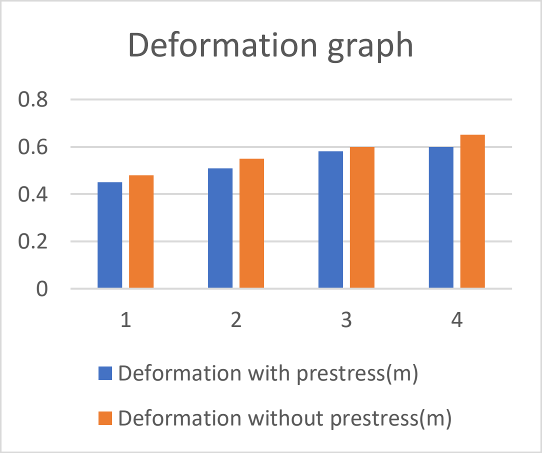
S.No	Deformation with prestress(m)
1	0.45
2	0.51
3	0.58
4	0.6

S.No	Deformation without prestress(m)
1	0.48
2	0.55





3	0.6
4	0.65



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