Parametric Analysis of Break Squeal using Finite Element Method

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***Abstract*:- Brake Squeal noise has been under investigation by automotive manufacturers for decades due to consistent customer complaints and high warranty costs. Furthermore, the development of methods to predict noise occurrence during the design of a brake system has been the target of many engineers in recent years. The brake squeal generally occurs in the range of 1-20 kHz. Squeal is the most prevalent, annoying noise to overcome this, a feasible study is made to predict brake squeal frequencies and suggesting design modifications for minimizing occurrences of brake squeal frequencies. Brake squeal frequencies and instabilities present in the brake system are to be predicted by using Complex Eigen Value Analysis. Complex eigen values are extracted by finite element modelling and modal analysis of disc brake system using high end software tool (ABAQUS). After identifying unstable frequencies and the behavior of the brake system under different conditions, the performance of some control methods is to be tested. This annoying noise canbe reduced by variations in geometry parameters such as disc like No of Vanes inside disc, Geometry of disc and material properties of brake system.**

Keywords: Brake Squeal, Frequency, Modes, Complex Eigen values.

I. INTRODUCTION:

Nowadays, the refinement of vehicle Noise, Vibration and Harshness (NVH) has considerably increased the contribution of brake noise in vehicle design and development process. As a general practice, brake noise is an irritating sound for consumers who may believe that it is symptomatic of a defective brake system and fill a warranty claim, even though the brake is functioning properly. Thus, understanding, prediction and prevention of brake noise and vibration has become an important aspect in brake design and development related to quality processes. A wide variety of brake noise and vibration phenomena are described by various terminologies such as brake squeal, creep groan, chatter, brake judder, brake moan, muh, squeak, etc. Among them, one general term, squeal, is probably the most prevalent, disturbing to both vehicle passengers and environment, and expensive for brake and automotive manufacturers in terms of warranty costs. However, there is no precise definition of brake squeal that has gained complete acceptance, but it is generally agreed that squeal is a sustained, high frequency vibration (above 1 kHz and below 10 kHz) of brake system components during a braking action leads to audible noise to vehicle occupants or passengers. Many drivers are familiar with the high-pitched squealing sound that comes from their vehicles or those of fellow motorists. That squeal, while certainly not music to the ears, might be a good thing. That is because Pep Boys notes that some pads are down, these devices produce a squealing sound to let drivers know it's time to get new pads. But squealing can be indicative of other things as well. Squealing noises are sometimes heard immediately after brake pads or rotors have been replaced. In such instances, the noise typically subsides within a day or two once the pads have been broken in.

II. LITERATURE SURVEY:

Review of the research work reveals that much work has been done on various aspects of brake squeal analysis. Prof. V. Kumar has been experimentally investigated that brake squeal propensity is higher at the frequency where there is modal coupling between in-plane and out of plane modes of brake disc. And it is very difficult to eliminate this high frequency squeal noise by modifications in brake pad and shim. With changes in brake disc structure especially the vane pattern it is possible to decrease the squeal frequency. Mukesh Kumar, Jayashree Bijwe conducted studies on two series of friction composites viz. P series (Cu powder based) and F series (Cu fiber based) and concluded that NAO friction material was responsible for enhancing most of the properties. Inclusion of copper content led to an appreciable increase in magnitude of _, decrease in _ sensitivity towards pressure–speed, temperature, and wear resistance also. H Ouyang noticed that disc brake can be made more stable when the pressure distribution at the disc/pads interface is more. He found that symmetrical, or even by shifting the center of the piston line pressure to the trailing side of the pad is more consistent. Aniket B. Ghatwai1 observed that the Process of Eigen Value analysis Provide the Accurate cause of Noise generation in Disc and Drum Brakes. Zhigang Chu conducted a study on parameter determination of minimal model for brake squeal. He also performed Transient Analysis for the dynamic equations of the 2-DoF model. The results show that nonlinear variation characteristics of the friction coefficient with braking speed can affect modal coupling of the system, and therefore lead to squeal instability.

III. METHODOLOGY

Creating different brake components like brake disc, pads, caliper etc., by using 3D modelling software's with different parameters. Disc Modelling: Several aspects proved to be particularly important in disc modelling, which required much attention to obtain reliable prediction of natural modes and frequencies. They included: Disc material (cast iron) characteristics, FE mesh/meshing type (free, mapped), FE element size and element type. Numerous analyses have been conducted and the influence of above characteristics on FE results investigated in detail. The results are compared to the experimental values (Experimental Modal Analyses – EMA). The experimental results were obtained by conducting own measurements, with some provided by the Universities of Liverpool and Bradford (roving hammer method). The disc was excited by means of a hammer from several points and different arrangements were selected. Both accelerometers and laser vibrometer were used. 3.1.2 Pad and Backplate Modelling. Friction pads are having by far the simplest geometry of all major brake components, proportion of the pad. Having nonlinear, non-isotropic, viscoelastic properties which considerably change with temperature, experiencing reversible and non-reversible changes and exposed to almost complete wear, the friction material is most difficult to accurately model. Similarly, to disc modelling, selection of suitable element size and type when meshing pads, plays an important role in obtaining reliable results. Furthermore, pad mesh must enable accurate introduction of contacts with pistons, calipers and disc, with thermal loads and wear setting even more stringent requirements. Modelling of the backplate, made of mild steel, is relatively straightforward, however modelling of the vibration absorbing shims attached to the pad backplate is very challenging. Backplate holds the pad, and it receives the pressure from piston and applies on Disc by pad.

3.1.3 Caliper Modelling: Geometrically, the caliper is most complex component of the brake assembly. It is not only difficult to model complex geometry, but also interaction with other parts -pistons and pads. 3.1.4 Knuckle Modelling: Knuckle is the most important component in the brake assembly, because it is the connection between Frame Chassis and wheels, it holds the Caliper, upper arms, Lower Arms, Steering Arm, etc.,

IV MODAL ANALYSIS:

Modal analysis is used to determine vibration characteristics such as natural frequencies and mode shapes of a structure or a machine component. The frequencies obtained from the modal solution have real and imaginary parts due to the presence of an unsymmetric stiffness matrix. The imaginary frequency reflects the damped frequency, and the real frequency indicates whether the mode is stable or not. There are mainly two types of Modal Analysis. They are,

- Complex Eigen Value Analysis (CEA)
- Transient Dynamic Analysis (TDA)

Damping is an influence within or upon an oscillatory system that has the effect of reducing or preventing its oscillation. In physical systems, damping is produced by processes that dissipate the energy stored in the oscillation. Examples include viscous drag (a liquid's viscosity can hinder an oscillatory system, causing it to slow down) in mechanical systems, resistance in electronic oscillators, and absorption and scattering of light in optical oscillators. Damping not based on energy loss can be important in other oscillating systems such as those that occur in biological systems and bikes. Not to be confused with friction, which is a dissipative force acting on a system. Friction can cause or be a factor of damping. The damping ratio is a dimensionless measure describing how a system decay after a disturbance. Many systems exhibit oscillatory behavior when they are disturbed from their position of static equilibrium. A mass suspended from a spring, for example, might, if pulled and released, bounce up and down. On each bounce, the system tends to return to its equilibrium position, but overshoots it. Sometimes losses (e.g., frictional) damp the system and can cause the oscillations to gradually decay in amplitude towards zero or attenuate. The damping ratio is a measure describing how rapidly the oscillations decay from one bounce to the next.

$$\zeta = \frac{c}{c_c} = \frac{\text{actual damping}}{\text{critical damping}},$$

where the system's equation of motion is

$$m\frac{d^2x}{dt^2} + c\frac{dx}{dt} + kx = 0$$

and the corresponding critical damping coefficient is

$$c_c = 2\sqrt{km}$$

or

$$c_c = 2m\sqrt{\frac{k}{m}} = 2m\omega_n$$

Complex Eigenvalue Analysis (CEA):

The classical method for instability prediction in squeal analysis can be done by calculation of unstable eigen frequencies of linearized equation of motion around the equilibrium point. Thus, the first step is to find the equilibrium point ($u0$) which can be done using Eq. (1). $K.$ $u0 = Fext + Fnl$ (1) Where $Fext$ is the external force such as the pressure on back plates and Fnl is the nonlinear force at the contact interface of disc and pad. Having the equilibrium point, the perturbation around the equilibrium point can be defined. Where M, C and K are the mass, damping and stiffness matrix, respectively. Regarding that Fnl $= Knl.\bar{u}$, the equation of motion can be written as $M \cdot$ u $C \cdot \bar{u} + (K - Knl) \cdot \bar{u} = Fext$ (4) In Eq. (4), Knl represents the nonlinear stiffness generated by friction force between pad and disc. Considering $u\ (\bar{}t) = A.e\lambda t$ and $Fext$ = 0, the closed form of Eq. (4) can be written as $(M \cdot \lambda\ 2 + C \cdot \lambda + K - Knl) \cdot A = 0$ (5) Where λ is the complex eigenvalue and A is the corresponding eigenvector. In fact, complex eigenvalue analysis is solving Eq. (5). The eigenvalues, can be expressed as $\lambda = \alpha \pm j\omega$, where $\sqrt{j} = -1$, in which α is the real part of complex eigenvalue, nominate as Re (λ), represents the stability of the system, and ω is the imaginary part of complex eigenvalue, nominated as I'm(λ), represents the mode frequency. The Eq. (6) expresses the generalized displacement of the system, $u\bar{}\ u(\bar{}t) = exp(\alpha t)$. (A1$cos\ \omega t \pm$ A2 $sin\ \omega t$) (6)

Where A1 and A2 are constants calculated by initial conditions (e.g., $\bar{u}(0) = u0$ and $\bar{u}(0) = v0$). According to Eq. (6), if α is negative, $\bar{u}$ decreases exponentially with time and the system will be stable. Otherwise, when α is positive, $\bar{u}$ increases with time and the system is then unstable. It is essential to have all eigenvalues on the left-hand side, to have a stable system. Although, all the eigenvalues with positive real part represent the instability but the squeal may not happen in all of them.

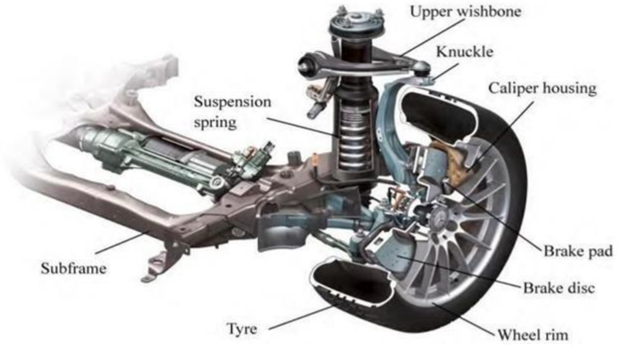
Transient Dynamic Analysis (TDA):

Transient means, something that fades with time. The dynamic analysis in time domain is called Transient Dynamic Analysis. You give the time history of loading and then you will get the time history of response. That means, load vs. time will be the input,Response(displacement, stresses, velocities, acceleration) vs. time will be the result. In dynamic analysis, there is frequency response analysis and transient response analysis. One is where you give loading as a function of frequency and the other as a function of time. In this project, the transient analysis is performed using 'Dynamic, Explicit' in Abaqus/Explicit which uses central difference method as time integration. The differential equation for structural dynamics can be considered as $M \cdot \ddot{u}(t) + C \cdot \dot{u}(t) + K \cdot u(t) = P(t)$ Such second order differential equation needs two initial conditions in order to be solved in a non-generic state. Let denote $u(0) = u0$ and $\dot{u}(0) = v0$ as initial conditions. Considering $u(tn) = un$, the foundation of central difference algorithm can be written as $\dot{u}n = (un+1 - un-1)/2h$ In which h is the time step. The above equation can be interpreted as the derivative at time t is approximated as the slope of the line passing through the function at $tn-1$ and $tn+1$. Using the Tylor series for the $un+1$ and $un-1$, we have $un+1 = un + h.\dot{u}n + h22.\ddot{u}n$, $un-1 = un - h.u + h22.\ddot{u}n$ By adding above two equations, u can be derived as $\ddot{u}n = (un+1 - 2un + un-1)/h2$ By substituting the first and second derivative, the discrete governing equation can be written as $(1/h2.M + 1/2h.C).un+1 = pn - (K - 2/h2.M).un - (1/h2.M - 1/2h.C)un-1$ Central difference method is conditionally stable provided that the step size h is smaller than the critical step size that can be calculated as below: L min=minimum element length $hcr \leq lmin/c$, $c = \sqrt{E/\rho}$. As a rule, in explicit transient analyses, the small value for h leads to the very time-consuming calculation that limit generating detailed FE models in terms of time, file volume and software capability for handling very large result files. Therefore, there is a trade-off between the time step (h) and total simulation duration that should be managed in a correct way. Step 0: Input mass (M), damping (C), stiffness (K) Calculate LU factorization of M such that M=LU Note that LU factorization of matrix M means to define it as M=LU in which L: lower triangular matrix U: upper triangular matrix Input initial conditions $u0$, $v0$ and step size, Calculate the initial acceleration from $\ddot{u}0 = M-1.[p(0) - C \cdot \dot{u}0 - K \cdot u0]$. Calculate LU factorization of $1h2M + 12hC$ such that $1h2M + 12h, C = LU$ in which L: lower triangular matrix U: upper triangular matrix. Calculate the starting displacement from Taylor series $u-1 = u0 - h.\dot{u}0 + h22\ddot{u}0$. Step 1: Loop for each time step, n=1……n. Step 2 Calculate the right-hand side of the iteration. $RHSn = pn - (K - 2h2M)un - (1h2M - 12hC)un-1$. Step 3 Solve for displacement at the next time step. $(1h2M + 12hC).un+1 = RHSn$. Step 4 Evaluate the set of velocity and acceleration. Step 5 Set $n \rightarrow n+1$ and continue to the next step.

ADVANTAGES & DISADVANTAGES:

The advantage of CEA is that it gives mode shape of unstable eigenvalues. In other words, it gives unstable modes that can be used for vibration refinement in terms of geometrical revisions in brake system. CEA uses the linearized equation of motion around the equilibrium point, thus it can be considered as a limitation of this method because it is valid only at the vicinity of the equilibrium point. Furthermore, it is a steady state method that is not able to model the transient behavior and nonlinearities of squeal phenomena. In addition, it uses some assumptions such as a constant contact area between disc and pads and linear friction law. In fact, these assumptions are not compatible with reality. The transient analysis gives only the frequencies that may contribute to brake squeal together with the "dominating" displacement field. It does not give any information about the unstable mode shapes at a given frequency. Therefore, having knowledge about the brake squeal mode shape, is essential to do structural modification and prevent brake squeal as well. On the other hand, due to the time dependency of moving load on disc, it is better to use TDA for modelling the transient variables of model such as displacement, velocity and acceleration. The main drawback of the transient analysis is mainly long computational time and having convergence for each iteration. Time-domain simulations require huge data storage in case of using small time step. In this case, it is hard to handle it by commercial software such as ABAQUS. The computational time and data storage will be a serious problem, particularly, when the FRF up to high frequencies is desirable so that the time steps should be very small in the central difference integration to give the Discrete Fourier Transform (DFT) in high frequencies with reasonable accuracy.

V. COMPONENTS DESCRIPTION:

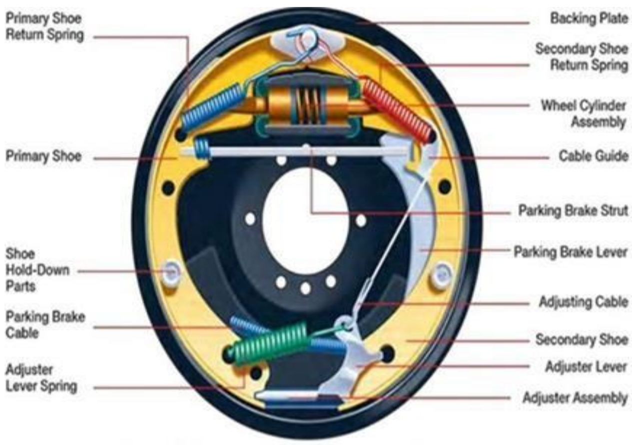


Brake system-functions:

The fundamental functions of a brake system can be summarized as follows. Reducing the speed of vehicle and if necessary to a requested stationary position (normal braking) Prevent unwanted acceleration while travelling downhill. Maintain the vehicle at stationary condition by parking brake. Conduct the vehicle to full stop with high deceleration braking (emergency braking). Ensure vehicle stability (under and over-steering and maintain tire friction)The principal type of brake used to generate braking force at the wheels is called friction brake. Friction braking converts the potential and kinetic energy into heat. Friction brakes can be categorized into two types: disc brakes and drum brakes. Figure depicts a disc brake assembly with suspension and part of the subframe. The brake assembly consists mainly of a brake disc, a calliper and a pair of pads with shims and under layer designed to generate a brake torque. During braking, the calliper pushes the pads onto disc by hydraulic piston. The friction force generated at the frictional surfaces between the brake pads and rotating brake disc generates the required braking torque transferred to the wheel/tire to stop the vehicle. Disc brakes are used in all front axle of passenger cars and in some cases can also be found in the rear axle. Despite their higher cost as compared to drum brakes, disc brakes are more robust and havebetter cooling performance.

Drum brake:

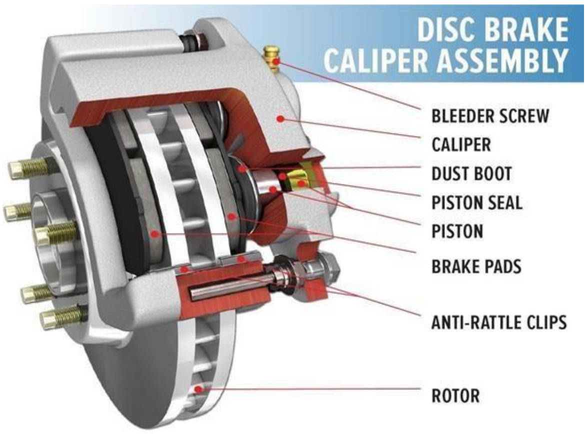
Drum brakes are radial brakes combining a brake shoe mounted on the stub axle and a rotating brake drum mounted on the axle. Drum brakes are composed of two brake shoes (seldom one only) that are pressed outward against the friction surface of the drum by the action of hydraulic piston during braking. When the braking operation is done, a spring pulls back the brake shoes to ensure a clearance between the surface of drum and brake pad. Drum brakes are less sensitive to external 3 dust and rain as it is a closed component and is cheaper than disc brakes. However, drum brakes suffer from poor cooling characteristics and early cracking.



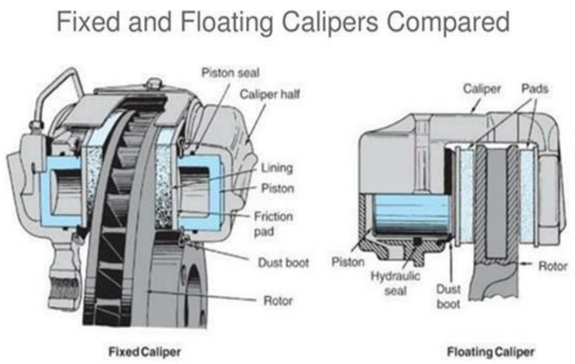
Automotive friction brakes are grouped according to their basic designs into two classes: Drum brakes use brake shoes that are pushed out in a radial direction against a brake drum. Disc brakes use pads that are pressed axially against a rotor or disco Advantages of disc brakes over drum brakes have led to their universal use on passenger-car and light- truck front axles, many rear axles, and medium-weight trucks on both axles.

Types of calipers:

There are mainly two types of calipers based on difference in their design they are Fixed calipers and Floating Calipers.

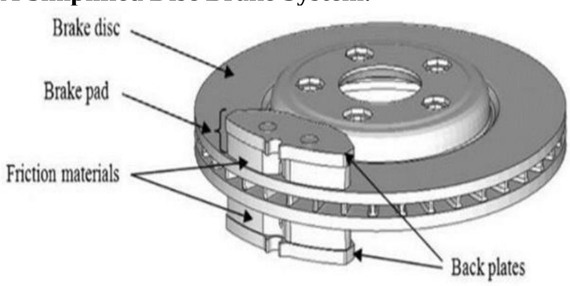


Fixed-caliper design, bolted solidly to the flange, has either two or four pistons which push the pads out. Fixed-caliper disc brakes have more balanced inner and outer pad wear with less pad taper than floating caliper designs. They require no anchor or integral knuckle for shoe support.



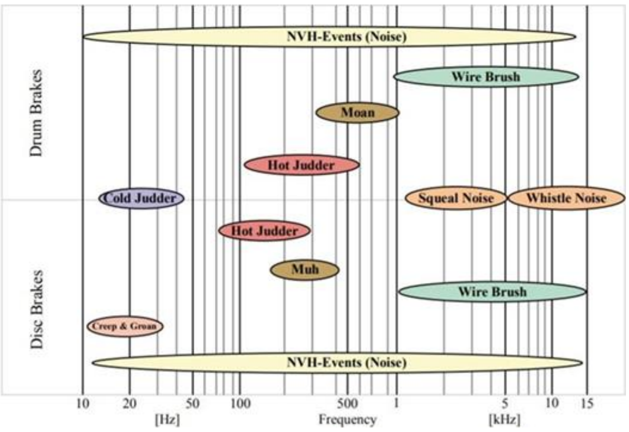
A typical floating caliper disc brake, one or two pistons are used on the inboard side only. The hydraulic pressure forcing the piston and pad toward the rotor also forces the piston housing (wheel cylinder) in the opposite direction to apply the outboard pad against the rotor. Floating caliper brakes offer a number of advantages over fixed caliper designs. They are easier 10 package in the wheel since they do not have a piston on the outboard or wheel side. They have a lower brake fluid operating temperature than the fixed caliper and, hence, lower brake fluid vaporization potential. They also have fewer leak points and are easier to bleed in service.

A Simplified Disc Brake System:



The “simplified FE model” of a brake system, see Figure 3, is composed of brake disc, a pair of brake pads. Here, the brake pads include the back plate but no shim neither friction material under layer is modelled. The back plates are made of steel, see Figure 4, to push the pad onto the brake disc and generate the requested friction torque. The normal force and frictional torque, I e tangential friction force, between the brake pads and brake disc may excite the brake disc to vibrate. The friction force makes the problem as nonlinear vibration so that the linear vibration theory is not able to model the squeal problem properly.

Brake Noises and Vibrations:



There are various kinds of brake noises and vibrations with numerous terminologies to nominate the phenomena in the literature that are probably inconsistent. One of the terminologies. Noises and vibrations can be classified that are based on frequency range. Some types of brake noise and vibration can be summarized.

VI Current Analysis Approach:

The complex eigenvalue analysis is the preferred method in the brake research community because it is more mature than the dynamic transient analysis. Furthermore, the complex eigenvalue analysis can provide much faster solutions than the dynamic transient analysis. To perform the complex eigenvalue analysis using ABAQUS, four main steps are required as follows. Nonlinear static analysis for applying brake-line pressure. Nonlinear static analysis to impose the rotational speed on the disc. Normal mode analysis to extract natural frequency of undamped system. Complex eigenvalue analysis that incorporates the effect of friction coupling. Since the equivalent stiffness matrix is asymmetrical because of friction, a complex eigenvalue analysis (CEA) is required. Equation can be written as a general eigenvalue problem. $A.X = \lambda.X$ Where λ is the eigenvalue and X the eigenvector. Both are complex valuated. Especially, the eigenvalue may be written. $\lambda = a + ib$.

Complex Eigenvalue Extraction Procedure:

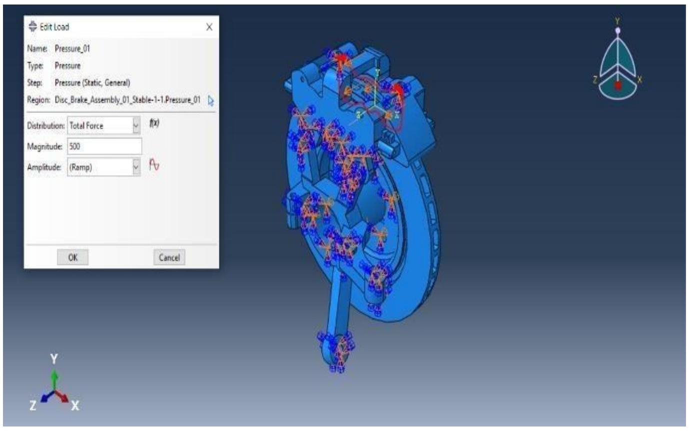
To extract complex eigen values in ABAQUS following procedure need to be followed: Step 1: Importing 3D CAD parts of brake system (Rotor, Pad, Caliper, Back plate, Fasteners and Knuckle) into the part module. Step 2: By using Property Module give the necessary properties like Elastic Modulus(E), Density(ƒ), and Poisson ratio(υ) etc. to the respective parts. The property values for the parts are listed below. List of materials and their properties for case 1:

PART NAME	MATERIAL	MECHANICAL PROPERTIES
Disc/Rotor	Cast Iron	ƒ=7.3 E-9 t/mm3, υ= 0.29, E=120 E3 Mpa
Caliper	Thermoplastics	ƒ=1.19 E-9 t/mm3, υ= 0.33, E=3 E3 Mpa
Pad	Non-Asbestos Organics	ƒ=2.44 E-9 t/mm3, υ= 0.3, E=219 E3 Mpa
Backplate	Steel	ƒ=8.05 E-9 t/mm3, υ= 0.27, E=210 E3 Mpa
Knuckle	Ductile-Iron	ƒ=1.129 E-9 t/mm3, υ= 0.27, E=165 E3 Mpa
Fasteners	Carbon Steel	ƒ=8.65 E-9 t/mm3, υ= 0.295, E=200 E3 Mpa

List of materials and their properties for case 1

Step 3: Select Assembly Module and assemble all components. Step4: create a general static step from Step Module. In the Model Tree, double-click Steps to create a new step. In the Create Step dialog box that appears, enter PRESSURE as the name and select ‘Initial’ as the step after which the new step will be inserted. Choose Static, General from the list of available general procedures and click Continue. Accept all default setting for the step definition. Step5: create a general static step using Step Module. In the Model Tree, double-click Steps to create a new step. In the Create Step dialog box that appears, enter ROTOR MOTION as the name and select PRESSURE as the step after which the new step will be inserted. Choose Static, General from the list of available general procedures and click Continue. Step 6: Create Frequency step using Step Module. In the Model Tree, double-click Steps. In the Create Step dialog box that appears, enter FREQ as the name. Select frequency from the list of linear perturbation procedures and click Continue. Step 7: Now, create a complex eigenvalue extraction step to extract the eigenmodes after the natural frequency extraction step FREQ. In the Model Tree, double-click Steps. In the Create Step dialog box that appears, enter COMPLEX FREQ as the name. Select Complex frequency from the list of linear perturbation procedures and click Continue. In the step editor, request eigenvalues. In the Other tabbed page of the editor, note that the unsymmetric matrix solver is selected. Click OK. Step 8: Define an interaction property with a coefficient of friction of 0.4 and apply it to the predefined contact pairs in the ROTOR MOTION step. In the Model Tree, double-click Interaction Properties. In the Create Interaction Property dialog box that appears, enter PAD_ROTOR-FRICTION as the name and select Contact as the type. Click Continue. In the Create Interaction Property dialog box that appears, enter PAD_ROTOR-FRICTION as the name and select Contact as the type. Click Continue. In the Edit Contact Property dialog box that appears, select Mechanical → Tangential Behavior. Choose Penalty as the friction formulation and enter 0.4 as the coefficient of friction. Click OK. In the Model Tree, click mouse button 3 on Interactions and select Manager from the menu that appears. In the Interaction Manager, select the cell corresponding to the first interaction under the step named ROTOR MOTION and click Edit. In Edit Interaction dialog box that appears, change the contact interaction property to PAD_ROTOR-FRICTION and click OK. Repeat steps e and f for the other interactions. Step 9: Predefined motion fields are not currently supported in Abaqus/CAE; so, we will use the keyword editor to specify an angular velocity of 5.0 rad/sec about the Z axis. The predefined set ROTOR will be used for this purpose. From the main menu bar, select Model □ Edit Keywords □ brake. In the Edit keywords dialog box appears, scroll down to find the *STATIC option block associated with the ROTOR MOTION step and select it. Click Add After at the bottom of the dialog box and enter the following text. MOTION, ROTATION ROTOR, 5., 0,0,0, 0,0,1 Note: the first entry is the angular velocity and the next two are coordinates that define the axis of rotation. Click OK. Step 10: Applying Loads and boundary conditions to the given brake system using Load Module.

Pressure applied on both sides & Boundary condition

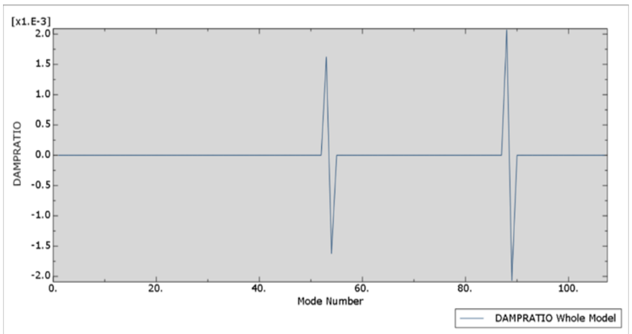


Step 11: Build Finite element model by meshing the components by using Mesh Module. Apply Tetrahedral mesh to all the components. Step 12: Now create job by using Job Module and run the simulation. **Output**: The real (EIGREAL) and imaginary (EIGIMAG) parts of the eigenvalues, (a and b) frequencies in cycles/time (EIGFREQ); and damping ratios (DAMPRATIO = -2a/|b|) are written automatically to the data (.dat) file and to the output database (.odb) file as history data. In addition, you can request that the generalized displacements (GU), which are the modes of the projected system, be written to the output database file. Output variables such as stress, strain, and displacement (which represent mode shapes) are also available for each eigenvalue; these quantities are perturbation values and represent mode shapes, not absolute values. Finally, two unstable modes are formed at 54th and 89th modes with squeal frequencies of 6096.5Hz and 8821.9Hz respectively as shown in figure.

VII. Results and Conclusion:

The focus of this study is on the reduction of low frequency squeals which ranges from 1.5 kHz to 10 kHz. To cover such a wide frequency range, 200 modes are requested in the complex eigenvalue calculation. In this case study, the basic parameters chosen for the parametric studies are friction coefficient and brake disc rotation speed. The friction coefficient μ = 0.4, and brake disc rotation speed is 5 rad/s. The results of complex eigenvalue analysis are in the form of λ= a +ib where a is the real part and b is the imaginary part of the eigenvalue which presents frequency of the unstable mode. All calculated eigenvalues are then plotted on a complex plane as shown in Figure. The squeal frequencies of 3 and 8 kHz from the test data are all predicted in the complex eigenvalue results as unstable modes. CASE-1: Finding component contributions through component contribution factor plug in at unstable modes.

Damping ratio Vs Mode number Plot (Case -1)



Mode No.	Real Part	Eigen Frequency	Damping Ratio
51	0	6000.329	0
52	0	6032.432	0
53	-31.0729	6069.014	0.001622
54	31.07292	6109.964	-0.001622
55	0	6224.416	0
56	0	6474.384	0
57	0	6522.656	0

Results around 1st Unstable mode (Case-1)

Mode No.	Real Part	Eigen Frequency	Damping Ratio
86	0	8669.965	0
87	0	8740.81	0
88	-57.3183	8815.311	0.002068
89	57.31826	8844.016	-0.002068
90	0	8937.792	0
91	0	9008.47	0

Results around 2nd unstable mode (Case-1)

CCF plots:

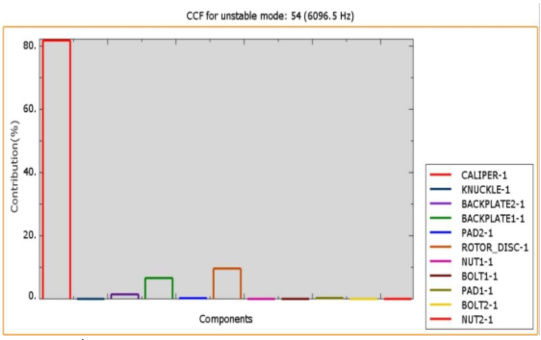
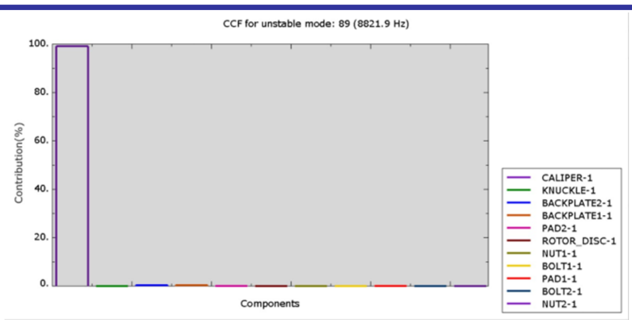


Fig.17 CCF plot at 54^{th} mode



CCF plot at 89th mode

Results from CCF plots found that Caliper and Rotor are contributing more for forming of instabilities in the brake system.

CASE-2:

Reduction of unstable modes by varying Material and Geometric parameters of the Caliper. From the literature review it is seen that instability of the most unstable modes are reduced by varying material properties and geometric properties of the Caliper. New material list for optimization of squeal.

PART NAME	MATERIAL	MECHANICAL PROPERTIES
Disc/Rotor	Cast Iron	ƒ=7.3 E-9 t/mm3, υ=0.29, E=120 E3 Mpa
Caliper	Aluminium alloys	ƒ=2.8 E-9 t/mm3, υ= 0.33, E=70 E3 Mpa
Pad	Non-Asbestos Organics	ƒ=2.44 E-9 t/mm3, υ=0.3, E=219 E3 Mpa
Backplate	Steel	ƒ=8.05 E-9 t/mm3, υ=0.27, E=210 E3 Mpa
Knuckle	Ductile-Iron	ƒ=1.129 E-9 t/mm3, υ=0.27, E=165 E3 Mpa
Fasteners	Carbon Steel	ƒ=8.65 E-9 t/mm3, υ=0.295, E=200 E3 Mpa

List of Materials and their properties for case 2

When Caliper is changed from "Thermoplastics" to "Aluminium alloys" and changes in geometry of the caliper, following results are obtained.

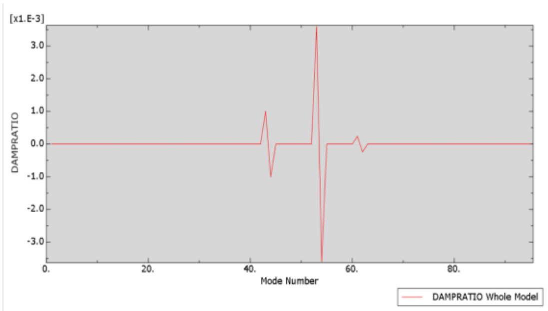


Fig.19 Damping ratio Vs Mode Number plot (case – 2)

Mode Number	Real Part	Frequency	Damping ratio
41	-4.9E-07	5645.229	0
42	-8.5E-07	5734.332	0
43	-18.3422	5808.744	0.001005
44	18.34225	5819.415	-0.001
45	-8.9E-07	5876.121	0
46	4.49E-07	5921.134	0

Table 5. Results around 1st unstable mode (Case-2)

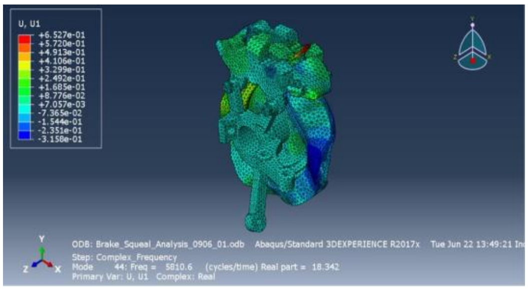


Fig.21 Mode shape at 44th Mode(case-2.1)

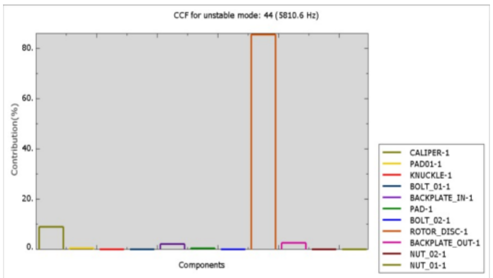


Fig.22 CCF for unstable mode: 44(5810.6 Hz)

Mode Number	Real Part	Frequency	Damping Ratio
51	8.43E-08	6340.875	0
52	0	6427.127	0
53	-73.0257	6449.677	0.003597
54	73.0257	6462.009	-0.0036
55	0	6486.661	0
56	9.31E-08	6518.755	0

Results around 2nd unstable mode (Case-2)

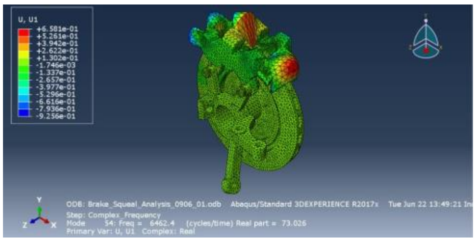


Fig.23 Mode shape at 54th Mode(case-2.2)

Mode Number	Real Part	Frequency	Damping Ratio
59	-1E-11	6755.876	0
60	4.93E-07	6775.311	0
61	-5.24906	6908.403	0.000241
62	5.24906	6920.376	-0.00024
63	1.74E-06	6923.108	0
64	-2.5E-06	6970.668	0

Table 7. Results around 3rd unstable mode (Case-2)

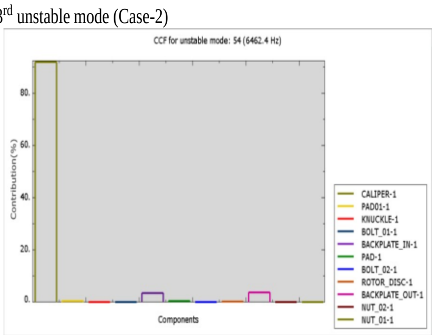


Fig.24 CCF for unstable mode: 54(6462.4 Hz

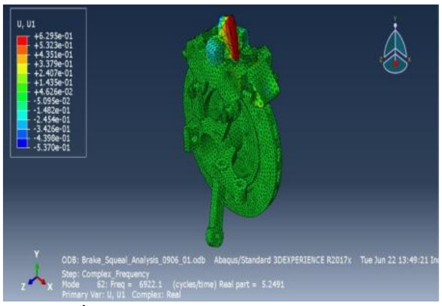


Fig.25 Mode shape at 62nd Mode(case-2.3)

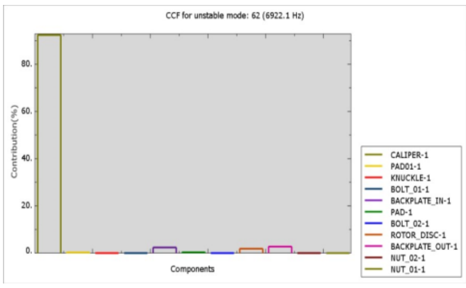


Fig.26 CCF for unstable mode: 62(6922.1 Hz)

Verifying Brake Squeal Simulations:

Verifying via correlation with test results or results from other programs. Checking the mass of the assembly of the brake system. Mass of the brake system from simulation results =12Kg. Approximate mass of the original brake system =12 to 15Kg. Assembly level correlation. Checked how the natural frequencies of the whole system correlate with test data or other available results.

Results and Conclusion:

This paper explores a proper way of predicting squeal frequencies using complex eigenvalue analysis. The complex eigenvalue analysis is performed with an implicit version of ABAQUS. The Process of Eigen Value analysis Provides the cause of Noise generation in Disc brake assembly. Unstable modes are predicted by using complex eigen value analysis. Instabilities present in the system are reduced by changing material of caliper from Thermoplastics to Aluminium alloys and outer geometry of the Caliper. With changes in brake disc structure especially the vane pattern and length of the flange it is possible to further reduce and eliminated the unstable modes at frequencies. Thus, instabilities present in the system are eliminated for low squeal frequency range.

Recommendations for Future Work:

The following suggestions are made for future work. Squeal analysis can be performed by varying parameters such as brake pressure, brake temperature, wear etc. The materials of the assembly can be optimized by composite materials. Dynamometer Experimental analysis may produce optimum results. Combining the data of modal analysis and thermal analysis would produce optimum results.

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