

## THE COMPLEX EIGENVALUE ANALYSIS OF BRAKE SQUEAL USING FINITE ELEMENT METHOD

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**Abstract:** *This paper presents the study of brake squeal, its investigation and suppression, using FEM and complex eigenvalue analysis. This paper supplies with simple FEM model which is used for the main calculation. Complex eigenvalue analysis is applied in order to discover the instability represented by two unstable modes at different values of frequency. Further in this paper the analyses of the influence of two parameters, friction and stiffness, to the system are executed.*

**Keywords:** Brake squeal, Dynamic instability, Complex eigenvalue analysis, FEM, Mode coupling

### 1. Introduction

Brakes represent one of the most important safety and performance components of a vehicle. They are important in terms of reliability, braking power as well as in a fluent operation. The unwanted side effect of braking operation is its occasional squeal. Brake squeal is a noise problem caused by friction induced vibrations initiated by the disc rotation (Nouby, 2009). Usually brake squeal occurs in the frequency range between 1 and 20 kHz or even more.

Squeal is considered to be a difficult subject due to complexity of the brake mechanism, its mechanical interactions, significant dependence on many parameters and high sensitivity on operating conditions. Many investigators have been trying to solve this problem since the 1930s, using experimental, analytical and numerical methods in order to understand, prevent and predict squeal occurrence (Culla, 2009) but this problem has not been solved yet.

In studies of North (Ouyang, 2003) and other researchers we can find that the dynamics of brake systems puts in evidence a coincidence between the squeal frequencies and the natural frequencies of the system. Particularly, they associated the squeal phenomenon to the coalescence of two eigen frequencies of the system. Their studies suggested the complex modal analysis of brake systems as a tool for squeal investigation (Massi, 2005). To study the stability of the system by the complex eigenvalues analysis, the friction effects between the pad and the disc are accounted by introducing linear elements in an asymmetric stiffness matrix. Finite element method (FEM) has become the preferred tool in studying disc brake squeal, whereas it offers much faster and more cost efficient solutions than experimental methods and it can predict squeal noise performance at early stage of design development. It is feasible to apply transient analysis or complex eigenvalue analysis. Complex eigenvalue analysis is more convenient since transient is more time consuming.

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## 2. Basic model, theories and mechanisms of brake squeal

The first experimental investigation discovered that squeal could have been caused by the variation of friction coefficient  $\mu$  at the contact interface. In 1938 Mills claimed that squeal was associated with the negative gradient characteristics of dynamic friction coefficient against the sliding velocity  $v$ . Later on, the stick-slip phenomenon had been described as unstable due to the static coefficient of friction  $\mu_s$  (stick), higher than the dynamic  $\mu_d$  (sliding) and this led to low sliding speed friction induced vibration instability. Both mechanisms led to unstable oscillation and had risen to self-excited vibration in the system (Duffour, 2002).

Later Fosberry and Holubecki (Frosberry, 1961) suggested that the disc brake tended to squeal when either a static coefficient of friction was higher than the dynamic coefficient or a dynamic coefficient decreased with increase of speed presented in the contact interface. But subsequent experiments proved that brake squeal could occur even though the friction coefficient was constant or even with a slight increase of relative velocity in the contact surface (Kinkaïd, 2003). In 1962 Spurr came up with the sprag-slip model, which claimed that unstable friction induced oscillations occur even with a constant friction coefficient.

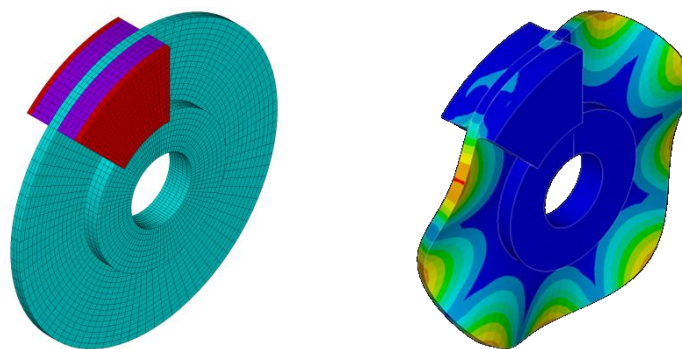
During squeal the friction material is deflected elastically along the disc surface by the frictional force (Ouyang, 2009). This deflection causes a second deflection with a component normal to the surface of the pad which reduces the friction and the stored elastic energy returns the system to the first configuration, and the cycle is repeated. Therefore, as a response to many tests and analysis, next studies were concerned to describe brake squeal as a friction induced self-excited mode coupling vibration with constant friction coefficient  $\mu$  (Hoffmann, 2002).

Using complex eigenvalue analysis, it is possible to obtain the dependency of eigenvalues, which indicates the instability of such system as a function of various parameters, for instance force, relative velocity, particular stiffnesses as well as the friction coefficient.

## 3. Development of simple FE model

For investigation of squeal a simple FE model had been created (Fig. 1) using FEM. This model composes of a disc, brake pads and pads' back plates. It was created for nonlinear analysis and complex eigenvalue extraction in purpose of stability examination. The model was used for calculations when investigating the influence of two system parameters: the change of the friction coefficient and the stiffness of the disc. This model does not reflect dependency on disc angular velocity and normal force because we assume constant friction coefficient. All the mechanical, inertial, stiffness and geometrical properties including the contact properties, between the pad and the disc, were done in ANSYS environment. After creation of this model static and modal analysis were applied.

Particular geometry of the brake assembly parts were distinguished according to the real brake assemblies and material parameters gained from experiments (Suchal, 2013).



*Fig. 1: Simple FE model and typical stable vibration mode*

The first step was to create a FE model of a disc brake by defining all the stiffness and inertial properties of the system. The second step was the application of pre-stressed static analyses for definition of contact and contact elements. The third step was to apply dynamical analysis for the rotation of the disc what gave the asymmetric stiffness matrix. The final step was application of modal analysis for non-symmetric systems matrices, in order to calculate eigenvalues of the system.

#### 4. Influence of different parameters on brake squeal

This section deals with the investigation of the effects of system parameters on squeal occurrence. It starts with changing the value of friction coefficient, followed by the analysis the influence of stiffness of the disc.

##### 4.1. Investigation of instability due to variable value of friction coefficient

In the following the instability will be called mode-coupling type (among the other names in literature are non-conservative displacement dependent forces and binary flutter). This instability is characterized by the process of the creation of the unstable frequencies. The oscillation frequencies of two stable structural modes of an undamped system come, as a function of a control parameter (friction value), closer and closer until they merge together and a pair of an unstable and stable mode is created (Hoffmann, 2002).

The results include the frequencies at which the squeal phenomenon occurs. The evolution of two frequencies is investigated by changing the friction coefficient  $\mu$  in range  $\{0 - 1\}$ .

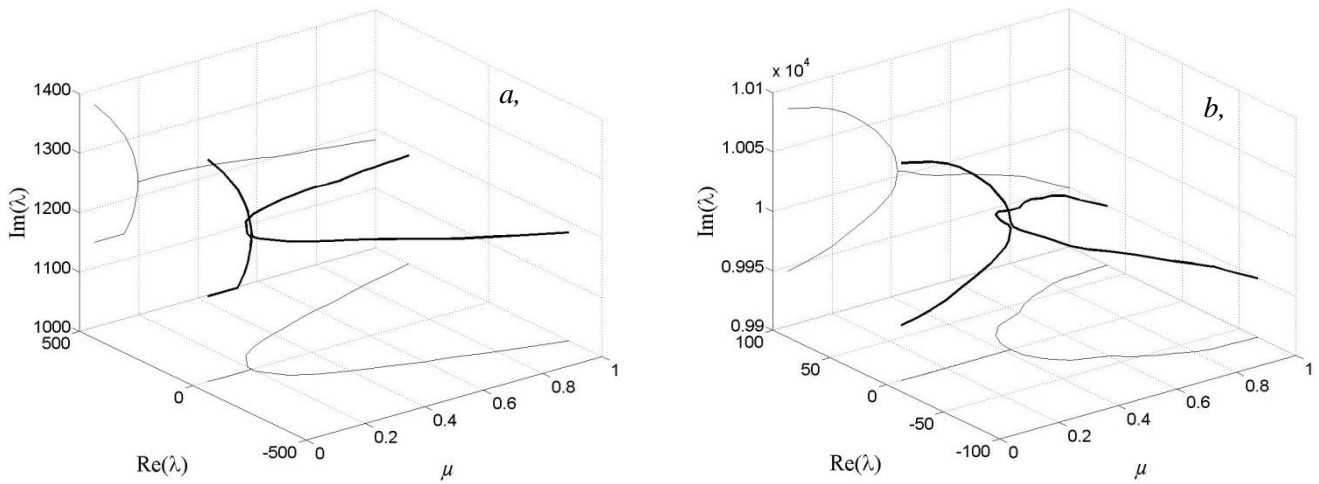


Fig. 2: Mode coupling gained from calculated data: a, first mode; b, second mode

For each value of the friction coefficient the complex eigenvalue analysis was executed. The results of the analysis are a list with complex eigenvalues and modes. Reviewing the list of complex eigenvalues the attention was paid to positive real part. Fig. 2 displays the evolution of real parts as a function of the friction coefficient. Firstly, the system features two separate modes in the frequency range; these two modes are stable since their real parts are zero. As the friction coefficient increases, the figure shows that the real parts remain zero and that the frequencies tend to get closer. The two modes reach the same frequency at a point known as the “bifurcation point” (Fig. 3). After that point, the system behavior is deeply altered (Fritz, 2002). Indeed, the frequencies remain equal but the real parts become non-zero and opposite.

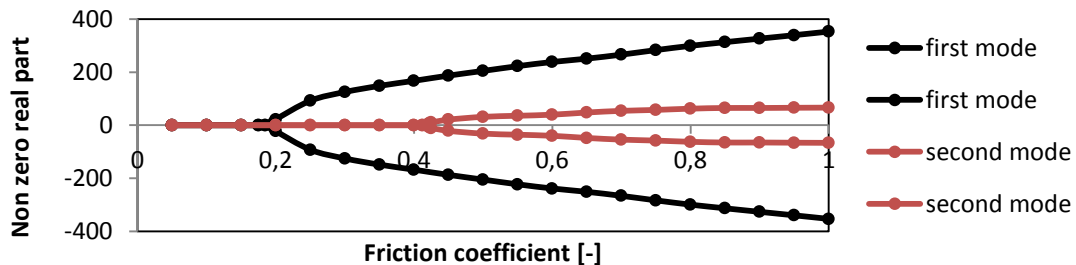


Fig. 3: The first two unstable Eigen modes of the disc brake assembly

##### 4.2. Effect of changing stiffness of the disc and back plate on brake squeal

Investigation of the effect of the disc stiffness on brake squeal was studied by changing the Young modulus of elasticity in range  $\{85 - 115 \%$  according to reference value of  $E_D = 1.975e11$  Pa. The analysis was performed on the first unstable mode because of the higher instability level (Fig. 3). As the Young modulus increased, the

stiffness of the disc increased as well and the value of instability dropped significantly. The graphical results show the dependency between different values of the disc stiffness and the instability (Fig. 4). It is assumed that by increasing stiffness, it is possible to reduce squeal propensity and disc vibration magnitude.

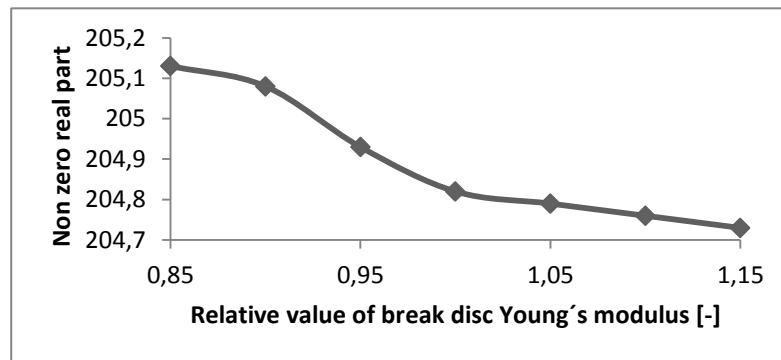


Fig. 4: Disc stiffness vs. instability dependency.

## 5. Conclusion

This work deals with the problem of brake squeal, its investigation and suppression. The instability causing brake squeal by gaining two unstable modes at different values of frequency is discovered by using the complex eigenvalue analysis on simple finite element model. It was found out that incensement of the friction coefficient results in higher squeal propensity.

The next investigation was the stiffness vs. instability dependence. Squeal can be reduced by increasing the Young's modulus of the brake disc.

The brake squeal problem is still under investigation. There has not been found any proper solution how to suppress this phenomenon completely. It is due to complexity of its mechanism and the influence of numerous parameters and side effects including thermal influence.

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