Numerical and Experimental Evaluation of Brake Squeal

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ABSTRACT

It is widely known that a typical brake system works by mitigating vehicle kinetic energy and transforming it into thermal energy, ultimately leading to energy dissipation. The main concerns related to this kind of system are: 1) low frequency vibration energy propagating throughout the vehicle structure when the system begins its unblocking action; and 2) high frequency vibration energy propagation which induces undesirable noise levels. Modal analysis of the system can provide important information about its vibration characteristics. Provided that coupling between the dynamic behavior, the pre-stress caused by the applied load, and friction characteristics will certainly occur, it is required that analyses be performed on the entire assembly. As such, this paper presents evaluation of a brake disc system regarding the brake squeal using finite element method comparing with experimental assessment.

INTRODUCTION

Brake systems are known for occasionally generating undesirable vibrations and unpleasant noise in vehicles. One of the most commonly known problems with these systems is brake noise, which has become a significant and challenging issue for the automotive industry in the last few decades. Brake noise issues have led car manufacturers, brake and friction material suppliers, to investigate various ways of improving their processes in order to minimize this issue, thereby reducing vehicle noise and increasing passenger comfort.

Some of the more popular brake noise and related vibration phenomena are described in a review by Kinkaid et. al. [1], where terms such as moan, judder, hum, chatter, squeak and squeal can be found. To narrow down the most critical issues, brake noise may be sub-divided into three main categories: 1) groan and moan noise, which is also known as low frequency noise (100-1000 Hz); 2) low frequency squeal, due to coupling of out-of-plane modes of the rotor with bending modes of the pads (1000-7000 Hz); and 3) high frequency squeal, due to coupling of in-plane modes of the rotor with bending modes of the pads (8000-16000 Hz).

Brake squeal is regarded as the most troublesome issue and as described by Crolla and Lang [2], it is directly responsible for passenger discomfort and heightened expenses to brake and car manufacturers in terms of warranty costs. It is generally accepted that brake squeal occurs as a consequence of the friction-induced oscillations developed in the brake disc, whether it be induced vibration or self-excited vibration via a rotating disc. As stated by Lang [3], brake squeal generally occurs at frequency values higher than 1000 Hz and, according to Eriksson [4], it can be described as a sound pressure level above 78 dB.

Experimental, analytical and numerical techniques have been developed for over 70 years in order to better understand, predict and prevent the occurrence of brake squeal. Some of the main issues limiting a wider use of experimental techniques are: the inability to predict brake squeal at the early stages of the design process; and the cost associated with numerous design iterations [5]. Therefore, finite element analysis has emerged as a viable approach for performing rapid changes in the geometry of disc brake components, thereby reducing the trial-and-error approach inherent in experimental procedures and their added expense. In such light, this paper presents numerical modal analysis applied to the brake squeal issue using the finite element method and comparing its results with experimental one.
NUMERICAL MODAL ANALYSIS

Modal analysis is a numerical technique based on the finite element method which aims at determining which natural frequencies and their modes of vibration of a part or assembly based on the distribution of mass and stiffness of the structure analyzed. For most cases, the damping involved is so small that their effects can be disregarded. Thus, the solution of a numerical modal analysis is obtained by determining the eigenvectors and eigenvalues that satisfy the following relationship:

\[ [K] \{\phi_i\} = \lambda_i [M] \{\phi_i\} \]  
(1)

Where:

- \([K]\) is the stiffness matrix;
- \([M]\) is the mass matrix;
- \(\lambda_i\) is the eigenvalue of the eigenvalue \(i\), and;
- \(\{\phi_i\}\) is the eigenvector of the eigenvalue \(i\).

The natural frequency is calculated from the eigenvalue by:

\[ \lambda_i = \omega^2 \]  
(2)

In this case, as the stiffness and mass matrices are symmetric, the solution can be determined by Lanczos Block algorithm. That is employed in the initial stage of the analysis where experimental tests are carried on the pads and the brake disc itself, calibrating the numerical models of each single component to provide a good dynamic behavior as compared to the experimental tests.

However, the dynamic behavior of each isolated component does not provide significant information, since the phenomenon of Brake Squeal occurs when there is a coupling between the modes out of the plane of the disc and the pads, because in this condition the system gets a self-excited vibration [6].

Therefore the numerical model must consider the same physical conditions found out by the system at the instant at the phenomenon of Brake Squeal occurs where there is a coupling between the modes out of the plane of the disc and the pads, because in this condition the system gets a self-excited vibration [6].

In finite element models, the tangent stiffness matrix is obtained by performing a non-linear static analysis in two steps. The first one applies pressure on the contact pads to close it and then a rotational velocity is induced providing the relative displacement between the parties with dissipative effects due to the friction sliding between theirs surfaces.

After that, the linear perturbation method is applied to get the tangent stiffness matrix from the last step of the static analysis and to carry it to the modal analysis, where damping must be considered, and then the natural frequencies is solved by the following equation:

\[ [K^T] \{\phi_i\} + \lambda_i [C] \{\phi_i\} = -\lambda_i^2 [M] \{\phi_i\} \]  
(4)

Where \([C]\) is the damping matrix.

The eigenvalue resultant from the equation above is now a complex one:

\[ \lambda_i = \delta_i \pm j \omega_i \]  
(5)

Where imaginary part is the natural frequency of the vibration mode and the real part provides the mode stability. If the real part is positive, the mode is stable and will not induce vibration in the system, while a negative value indicates an unstable model, thus generating noise.

In this condition, the matrices become non-symmetrical and complex, so the algorithm block Lanczos can no longer handle
it, but by combining a damped reduced Lanczos method with complex Hessenberg one the eigenvalues can be determined.

EXPERIMENTAL ANALYSIS

A commercial full-scale brake dynamometer (Link 3900 – Figures 2 and 3) was used to evaluate the brake performance of a Brazilian passenger car front brake on a NVH test procedure (GMW 14591). The relevant test parameters are summarized on Table 1.

Figure 2. Full scale brake dynamometer used in this study.

Table 1. GMW14591 test parameters.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Rolling Radius</td>
<td>275.0 mm</td>
</tr>
<tr>
<td>Wheel Load</td>
<td>595 kg</td>
</tr>
<tr>
<td>Inertia</td>
<td>45 kg·m²</td>
</tr>
<tr>
<td>Piston Diameter</td>
<td>48.00 mm</td>
</tr>
<tr>
<td>Effective Radius</td>
<td>97.50 mm</td>
</tr>
<tr>
<td>Number of Pistons</td>
<td>1</td>
</tr>
</tbody>
</table>

Figure 3. Brake assembly of a test on the dynamometer.

EXPERIMENTAL X NUMERICAL

Initially the numerical analyzes were performed on each component separately, by calibrating up the dynamic mechanical properties of the materials involved, getting very close numerical and experimental results, as can be seen in the figures below.

Figure 4. Numeral and experimental response of the free-free disc.

The disc used in this work was made from commercial gray cast iron, and their hardness was in the ranges of 210–230 HB. As can be seen in Figure 4, the disc has no ventilation channels. The brake pad material is a commercial OEM low-steel formulation.
After the calibration step of the individual components, the assembly model was developed. Figure 6 shows the mesh used in the finite element model.

As boundary condition, considering that the rotation speed is not high enough for the gyroscopic effects chances significantly the dynamic behavior of the assembly, so to facilitate the convergence of the static analysis of the application of pressure on the pads, a cylindrical support with all components restricted was applied on the inner cylindrical faces of the disk (Figure 7). To obtain an easier convergence, the displacement in the plane of the flat face of the rivets pads was fixed (shown in red in the Figure 7).

The interaction between the pads and the disk is established by defining a pair of contact for each pad, using nonlinear contact with friction coefficient of 0.4 and Augmented Lagrange formulation. To join the various parts of the pad were used bonded contacts with MPC formulation.

Static analysis was performed into two load steps, in the first one, a force is applied on the faces of the pads to close the contact status, and in the second one, a rotation of the contact elements of the faces of the pads is applied to provide a frictional sliding motion between the contact elements the faces of the pads and the faces of the disk (Figure 8).

From the pre-stressed condition at the end of the static analysis, the tangent stiffness matrix is got by linear perturbation method and carried out to the undamped modal analysis.

In modal analysis, all natural frequencies up to 14kHz were evaluated by a reduced damped eigenvalue algorithm. From those, six vibration modes had negative value of the real part,
so indicating an unstable mode, and therefore being able to induce Brake Squeal.

The brake dynamometer test results are summarized on Figure 10. It can be seen that the higher noise amplitudes are in frequencies around 11 and 13 kHz and could be predicted with the numerical analysis.

Table 2 collects the data comparison between experimental and numerical results. In this frequency range, the error is around 4%. The lower frequencies occurrences were not predicted by the numerical analysis, but this error can be attributed to the simplified assumptions on the modeling. Further work to include the fastening conditions, spider and caliper in the modeling will be addressed in the future.

<table>
<thead>
<tr>
<th>Predicted Unstable Mode</th>
<th>Predicted Frequency</th>
<th>Squealing Frequency (Experimental)</th>
<th>% Error</th>
</tr>
</thead>
<tbody>
<tr>
<td>53</td>
<td>10795 Hz</td>
<td>10350 Hz</td>
<td>4.29%</td>
</tr>
<tr>
<td>69</td>
<td>13165 Hz</td>
<td>12875 Hz</td>
<td>2.25%</td>
</tr>
</tbody>
</table>

**SUMMARY/CONCLUSIONS**

The numerical modeling of a brake system concerning squeal propensity was conducted in this work. In this study, the unstable modes of a brake system were calculated on the range up to 17 kHz using ANSYS.

Also, a noise dynamometer test was made in order to validate the simulation. The numerical analysis was able to predict successfully the two unstable modes with higher noise amplitude on the dynamometer test procedure. This analysis can lead to measures involving pad/disk shape or materials optimization in order to reduce or eliminate squeal noise.

Further work can be done in order to predict the other unstable modes, including refining the fastening conditions and adding other important parts who can contribute to mode coupling.

**REFERENCES**

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