## Dominant damping effects in friction brake NVH: The relevance of joints

<table>
<thead>
<tr>
<th>Journal:</th>
<th>Part D: Journal of Automobile Engineering</th>
</tr>
</thead>
<tbody>
<tr>
<td>Manuscript ID:</td>
<td>Draft</td>
</tr>
<tr>
<td>Manuscript Type:</td>
<td>Special Issue</td>
</tr>
<tr>
<td>Date Submitted by the Author:</td>
<td>n/a</td>
</tr>
<tr>
<td>Complete List of Authors:</td>
<td>Tiedemann, Merten; Hamburg University of Technology, Dynamics Group Kruse, Sebastian; Audi AG, Hoffmann, Norbert; Imperial College London,</td>
</tr>
<tr>
<td>Keywords:</td>
<td>braking dynamics, damping, brake squeal, joints, NVH</td>
</tr>
<tr>
<td>Abstract:</td>
<td>The experimental analysis of a single component of a brake system and an assembly consisting of three components is used to clarify the relevance of joints in terms of damping and nonlinearity in state-of-the-art brake systems. For this purpose a series of experimental modal analyses is conducted. A comparison of the results obtained from the single component and of the assembly strongly indicate that the joints which necessarily exist in an assembled structure have an impact on the dynamic behaviour of the structure. The modal damping values of the jointed structure are up to factor sixty higher than those of the single component values. Also, a significant amplitude dependency of the frequency response functions is visible. These observations demonstrate that joints are a major source of energy dissipation in friction brake systems and, in addition, that they introduce nonlinear behaviour to the system which has the potential to limit squeal amplitudes. Therefore, mechanical joints in brake systems should be considered as decisive design elements for noise-vibration-harshness (NVH) issues in brakes.</td>
</tr>
</tbody>
</table>
Dominant damping effects in friction brake NVH: The relevance of joints

1 Tiedemann, Merten; 2 Kruse, Sebastian; 1,3 Hoffmann, Norbert

1: Dynamics Group, Hamburg University of Technology, Hamburg, Germany
2: Audi AG, Ingolstadt, Germany
3: Imperial College London, London, UK

KEYWORDS – damping, brake squeal, modelling, joints, NVH

ABSTRACT - The experimental analysis of a single component of a brake system and an assembly consisting of three components is used to clarify the relevance of joints in terms of damping and nonlinearity in state-of-the-art brake systems. For this purpose a series of experimental modal analyses is conducted. A comparison of the results obtained from the single component and of the assembly strongly indicate that the joints which necessarily exist in an assembled structure have an impact on the dynamic behaviour of the structure. The modal damping values of the jointed structure are up to factor sixty higher than those of the single component values. Also, a significant amplitude dependency of the frequency response functions is visible. These observations demonstrate that joints are a major source of energy dissipation in friction
brake systems and, in addition, that they introduce nonlinear behaviour to the system which has the potential to limit squeal amplitudes. Therefore, mechanical joints in brake systems should be considered as decisive design elements for noise-vibration-harshness (NVH) issues in brakes.

ACKNOWLEDGEMENT

Part of the funding for the presented study is provided by Audi AG and AiF in the context of Cooperative Industrial Research (IGF) grant no. 16799N.

The AiF project of the Research Association GFaI has been supported by the Federal Ministry of Economics and Technology on the basis of a decision by the German Bundestag.

1. INTRODUCTION

Friction induced vibrations in automotive disc brakes are of interest for academic research as well as for industry [1]. The numerous customer complaints due to brake noise cause high warranty costs [2]. To enable the development of silent brakes, NVH engineers analyze these phenomena using computational and experimental simulations as well as car tests. In automotive industry computational simulation becomes more and more important due to shorter product development processes as well as cost reduction
issues. This is also true for the brake system for which complex eigenvalue analysis has now been established as the standard simulation approach in industry [3]. However, simulation results are far from being as reliable as results from brake dynamometers and car tests. Among the variety of unsolved problems concerning the simulation of automotive disc brake squeal the role of damping and its modelling is crucial [4]. Up to now, damping is either not modelled or inadequately modelled (e.g. Rayleigh damping), even though damping effects mainly determine the stability of the brake system. This leads to the well-known problem that computational simulation of the brake system identifies more instabilities than can be found in experimental simulations on a brake dynamometer [5].

Like each complex mechanical structure an automotive disc brake contains a plethora of joints. Figure 1.1 highlights a selection of them. Compared to the effects of material damping, energy dissipation due to mechanical joints in automotive disc brakes has been overlooked for a long time although in most assembled structures joint damping is the dominant damping effect [6]. Moreover, joints introduce nonlinear behaviour to the brake system which causes amplitude dependency and can limit squeal amplitudes. In other engineering disciplines like defence or aviation the use of joints as design variables is already state-of-the-art [7,8] and they are effectively used to optimize the dynamic behaviour of the respective mechanical structures. A survey on the use of dry
friction for system enhancement is given in [9]. However, the analysis of the dynamics of joints, from an experimental as well as from a simulation point of view, is known to be challenging [8]. Ibrahim [10] gives an overview about the mechanics of contact as well as friction and highlights the main characteristics. Recent investigations [11,12,13,14] indicate a strong impact of joint damping on the dynamic behaviour of disc brakes. However, further investigations are necessary to determine qualitatively and quantitatively the contribution of joint damping to global damping.

[Figure_1_1.png]

Figure 1.1: Joints in automotive disc brake (1: calliper-carrier; 2: calliper-pad; 3: pad-carrier; 4: carrier-knuckle)

A first step towards the integration of adequate damping models in the simulation is the identification of dominant damping effects in the brake system. The objective of this paper is to show the close interrelation of the modelling of damping and the characteristics of mechanical joints as well as to demonstrate that joint damping is the dominant damping mechanism in brake systems.

The paper is structured in the following way. In the next section the experimental setups and the conducted experiments are introduced. This includes a detailed description of the analyzed assembly. The obtained results are analyzed and discussed in the third
The final section provides conclusions and an outlook to future research activities.

2. EXPERIMENTAL SETUP

To identify dominant damping effects in automotive disc brakes two sets of experiments have been defined: an experimental modal analysis of single components of the brake system under free boundary conditions and an experimental modal analysis of a component assembly. The latter setup is designed in such a way that additional damping effects imposed by joints in a squealing state can be investigated. By comparing the results of both experiments the magnitude of joint damping can be derived.

Among the components in the brake system the brake carrier is a crucial component. It has several contact areas with other components of the brake system and is regularly employed as a NVH countermeasure. In addition, it is close to the contact interface between brake pad and disc which is the origin of brake squeal. Due to these characteristics the brake carrier is used as the guiding component in this paper and is taken as a representative single component of the brake system. However, the statements made in the following about modal damping and modal density can be
transferred to most other components in the brake system. In the following two
subsections the setups and the conducted experiments are described in detail.

Brake Carrier

An experimental modal analysis of a brake carrier under free boundary conditions has
been conducted using two triaxial acceleration sensors (PCB 256A01) and a modal
hammer (PCB 084A14). The structure has been excited at nine points and for each
excitation six acceleration signals have been measured.

Assembly

For the experimental analysis of joint damping a trade-off is required: the setup should
be as simple as possible to enable the assignment of observed damping effects to a
specific joint, but it should also be as close as possible to a real brake system. Figure 2.1
shows the selected minimal assembly used for additional investigations. It consists of a
floating calliper, a brake carrier and an outboard brake pad. The assembled structure has
three mechanical joints: the contact surface between brake pad and carrier arm, the
contact surface between the fingers of the brake calliper and the brake pad, and the
guidance of the brake calliper in the brake carrier. In order to be able to observe the
effect of one specific joint, the other two have been deactivated. The rubber bushing
normally used between brake calliper and brake carrier has been removed. Instead a steel bushing is used. To prevent relative movement in the joint the components are connected by a weld spot. The deactivation of the joint between the fingers of the brake calliper and the brake pad is achieved by using bolts (as depicted in Figure 2.1). The two bolts fix the position of the brake pad relative to the brake calliper. Despite these modifications the general conditions are still comparable to a real brake system concerning position of the components and the force transmission between them.

![Figure_2_1.png]

Figure 2.1: Assembly consisting of brake carrier, brake calliper and brake pad

The only remaining free joint is the contact surface between brake pad and brake carrier (see Figure 2.2). This joint is of special interest since it is often used for NVH countermeasures (e.g. shim, grease). For an experimental modal analysis of a structure with nonlinearities (e.g. joints) the choice of the excitation signal is crucial and determines whether the nonlinearity can be studied or not. In this context, the excitation with non-harmonic signals causes a variety of difficulties (e.g. linearization, uniform excitation of all frequencies) and is not suitable. To identify the effect of the nonlinearities the structure is excited with a force-controlled stepped sine signal with a shaker (type V406). This approach corresponds to the harmonic balance method and enables the analysis of nonlinear structures: the excitation energy is bundled at one
frequency and the harmonic response of the structure is measured at steady-state [15,16].

The response of the structure is measured via five triaxial acceleration sensors (PCB 256A01) placed on the brake carrier. The position of the force sensor (PCB 208C02) can be seen in Figure 2.1. The shaker is coupled to the force sensor via a stinger. To minimize the effect of the stinger and the shaker on the measured frequency response functions several pre-tests with different stinger lengths have been conducted. Since the ratio of effective shaker mass to the mass of the assembled structure is below five percent, the effect of the shaker mass on the frequency response of the structure is negligible.

[Figure 2.2.png]

Figure 2.2: Operative joint (with dry friction)

3. ANALYSIS AND RESULTS

For analysis of results the complex mode indicator functions (CMIF) of the measured data have been used since they provide a compact representation of the characteristics of the measurement. Peaks in this function indicate an eigenfrequency of the system [17]. The complex mode indicator functions of a set of frequency response functions can be obtained by a repeated singular value decomposition of the form
\[ H(\omega) = U(\omega) \cdot \Sigma(\omega) \cdot V(\omega), \] \hspace{2cm} (2.1)

where \( H \) is the matrix of frequency responses, \( \Sigma \) is the matrix containing the singular values, \( U \) and \( V \) are the left and right singular eigenvectors, respectively. The indicator functions can then be calculated by

\[ \text{CMIF}(\omega) = \Sigma(\omega)^T \cdot \Sigma(\omega). \] \hspace{2cm} (2.2)

These functions and the modal parameters for the two setups are presented in the following two subsections.

**Brake Carrier**

Figure 3.1 shows the first CMIF of the brake carrier. Up to 2.4 kHz eight eigenfrequencies can be clearly identified indicating a high modal density. The narrow, high peaks emphasize that modal damping for each of these modes is low.

[Figure_3_1.eps]

Figure 3.1: Complex mode indicator function (CMIF) of brake carrier
Table 3.1 shows the modal parameters extracted from the measured frequency response functions up to 2.4 kHz. The above emphasized idea of low modal damping is correct: the obtained values prove that the brake carrier is lightly damped. This is not surprising since in the case of a single component, modal damping represents mainly material damping of a structure which is small for all metals.

<table>
<thead>
<tr>
<th>Mode</th>
<th>Eigenfrequency [Hz]</th>
<th>Modal Damping [%]</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>586</td>
<td>0,21</td>
</tr>
<tr>
<td>2</td>
<td>772</td>
<td>0,25</td>
</tr>
<tr>
<td>3</td>
<td>1134</td>
<td>0,18</td>
</tr>
<tr>
<td>4</td>
<td>1273</td>
<td>0,10</td>
</tr>
<tr>
<td>5</td>
<td>1547</td>
<td>0,08</td>
</tr>
<tr>
<td>6</td>
<td>2103</td>
<td>0,07</td>
</tr>
<tr>
<td>7</td>
<td>2255</td>
<td>0,21</td>
</tr>
<tr>
<td>8</td>
<td>2324</td>
<td>0,15</td>
</tr>
</tbody>
</table>

Table 3.1: Modal parameters of the brake carrier up to 2.4 kHz

Assembly

Figure 3.2 shows the CMIF of the assembly for the frequency range between 1000 Hz and 1500 Hz for an excitation level of 0.5 N. The shape of the peaks indicate that the modal damping for each of the modes in this frequency range is significant.

[Figure_3_2.eps]

Figure 3.2: Complex mode indicator function (CMIF) of assembly
Table 3.2 shows the modal parameters of the assembled structure up to 2.4 kHz extracted from the measured frequency response functions for an excitation level of 0.5 N. This low frequency range is of special interest since with increasing frequency the displacement amplitudes decrease leading to a reduced impact of joints on global damping.

<table>
<thead>
<tr>
<th>Mode</th>
<th>Resonance Frequency [Hz]</th>
<th>Modal Damping [%]</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>597</td>
<td>0,66</td>
</tr>
<tr>
<td>2</td>
<td>708</td>
<td>0,88</td>
</tr>
<tr>
<td>3</td>
<td>1013</td>
<td>1,27</td>
</tr>
<tr>
<td>4</td>
<td>1117</td>
<td>1,8</td>
</tr>
<tr>
<td>5</td>
<td>1235</td>
<td>6,02</td>
</tr>
<tr>
<td>6</td>
<td>1354</td>
<td>0,65</td>
</tr>
<tr>
<td>7</td>
<td>1444</td>
<td>0,83</td>
</tr>
<tr>
<td>8</td>
<td>1583</td>
<td>2,43</td>
</tr>
<tr>
<td>9</td>
<td>1846</td>
<td>1,77</td>
</tr>
<tr>
<td>10</td>
<td>2019</td>
<td>1,09</td>
</tr>
<tr>
<td>11</td>
<td>2128</td>
<td>0,39</td>
</tr>
<tr>
<td>12</td>
<td>2184</td>
<td>0,67</td>
</tr>
<tr>
<td>13</td>
<td>2227</td>
<td>0,9</td>
</tr>
<tr>
<td>14</td>
<td>2264</td>
<td>1,08</td>
</tr>
<tr>
<td>15</td>
<td>2315</td>
<td>1,03</td>
</tr>
</tbody>
</table>

Table 3.2: Modal parameters of assembly for 0.5 N excitation level up to 2.4 kHz

The number of resonance frequencies in the analyzed frequency range changed from eight to fifteen in comparison to the brake carrier as a single component. The origin of these additional modes is twofold. First, new mode shapes occur due to the changed boundary conditions. Second, it is a well-known fact that deflection shapes of a brake system in squealing state are often dominated by single component mode shapes which
have been observed under free boundary conditions [1]. In this case, the brake calliper and the brake pad impose a single component mode shape on the structure mode shape. Keeping this in mind, at least the mode shapes of the assembled structure at 597 Hz, 708 Hz, 1117 Hz, 1235 Hz and 1583 Hz can be assumed to be similar to the mode shapes of the brake carrier found under free boundary conditions (see table 3.1).

Not only the resonance frequencies and their quantity change but also modal damping. Comparing the modal damping values listed in table 3.1 and table 3.2, it can be observed that the modal damping increased significantly. Taking the modes of the assembled structure which have been identified as dominated by brake carrier mode shapes and comparing their modal damping values to the respective values of the brake carrier, delivers additional insight into the characteristics of joint damping: the increment in damping is not uniform. Among the different modes it varies between a factor of three and a factor of sixty. The influence of joints on global damping is mode-dependent.

To explain this observation, the mode shapes of the brake carrier under free boundary conditions are addressed again. Figure 3.3 shows the first and the fifth mode shape of the brake carrier which relate to the first and eighth mode of the assembled structure.

[Figure_3_3.png]
Figure 3.3: First mode shape (left) and fifth mode shape (right) of the brake carrier (top view)

The first mode shape is characterized by a movement of the entire carrier arms in x-direction, whereas the fifth mode shape shows a huge movement of the end part of the arms in y-direction. The joint between the brake pad and the brake carrier allows relative displacement in the y-direction and limits the relative displacement in x- and z-direction. Since dry friction effects in the contact surface cause the energy dissipation in the joint, large amplitudes of relative displacement increase the quantity of dissipated energy.

Having a look at table 3.1 and 3.2 again and comparing the increment in modal damping of the first and fifth mode of the brake carrier with their respective counterparts of the assembly, it can be seen that the modal damping of the former mode is increased by a factor of three, whereas the modal damping of the latter one is increased by the factor of about thirty. These results clearly indicate the relation between displacements in the joint and their impact on modal damping.

Due to nonlinear characteristics of the joint different excitation amplitudes lead to a different response of the system. In order to analyze this amplitude dependency of the dynamics of the assembled structure tests with different force amplitudes have been
conducted. Figure 3.4 and Figure 3.5 show the complex mode indicator function for different excitation levels in three different frequency ranges.

**Figure 3.4:** Complex mode indicator function (CMIF) for different excitation levels

The plots clearly demonstrate that the modal parameters, resonance frequency and modal damping, are both amplitude dependent for all analyzed modes. The complex mode indicator function of the first mode, depicted in Figure 3.4, shows a clear shift of the resonance frequency to lower frequencies. In addition, a decreasing amplitude of the mode indicator function for higher force amplitudes can be observed. This indicates an increment in modal damping. Figure 3.5 shows the complex mode indicator function in the frequency range where the fourth and the fifth mode can be found. Whereas the resonance frequency decrease for both with increasing excitation level, the modal damping behaves in antithetical manner. It increases for the fourth mode and decreases for the fifth one.

**Figure 3.5:** Complex mode indicator function (CMIF) for different excitation levels

4. **CONCLUSION AND OUTLOOK**
It has been shown that the modal damping of the assembly consisting of brake carrier, brake calliper and brake pad is significantly higher than the modal damping of the brake carrier. The additional modal damping is introduced by the studied joint. Hence, joint damping is one of the dominant damping effects in automotive disc brakes. As a consequence, it seems as if material damping could rather be neglected in modelling but not joint damping. In addition, the results indicate that joint damping is highly mode-dependent. Taking into consideration joint damping in the stability calculations of the equilibrium solutions would definitely increase their validity.

A further result of this study is that joints introduce nonlinearities to the brake system. From a stability point of view this fact can be even more valuable than the strong energy dissipation in mechanical joints. In fact, nonlinearities determine the bifurcation behaviour and the evolution of non-equilibrium solutions like brake squeal.

The implications for application and future work are twofold. First, modelling of damping should not focus on material parameters alone, but also on adequate modelling of joint damping. This is either possible by using measured or estimated modal damping parameters of the brake system (or parts of it) or by using proper joint models. Second, a systematic use of joints for the reduction of brake noise problems could be a target. Although the brake community is aware of the potential of joints, until now, this tuning
knob is not used consequently and the industry instead strongly concentrates on structural and material modifications alone. Further work will focus on these two aspects in order to improve the quality of disc brake simulations. To achieve this, joints in disc brakes will be analyzed by further experiments and characteristic parameters will be identified. These can then hopefully be used for proper modelling of joints and will work as a novel basis for NVH countermeasures in brake squeal.

REFERENCES


14 Kruse, Tiedemann et al., The influence of joints on friction induced vibrations in brake squeal, submitted to JSV.


Figure 1.1: Joints in automotive disc brake (1: calliper-carrier; 2: calliper-pad; 3: pad-carrier; 4: carrier-knuckle)
Figure 2.1: Assembly consisting of brake carrier, brake calliper and brake pad
Figure 2.2: Operative joint (with dry friction)
Figure 3.1: Complex mode indicator function (CMIF) of brake carrier
232x159mm (300 x 300 DPI)
Figure 3.2: Complex mode indicator function (CMIF) of assembly
232x159mm (300 x 300 DPI)
Figure 3.3: First mode shape (left) and fifth mode shape (right) of the brake carrier (top view)
Figure 3.4: Complex mode indicator function (CMIF) for different excitation levels
238x168mm (300 x 300 DPI)
Figure 3.5: Complex mode indicator function (CMIF) for different excitation levels
232x159mm (300 x 300 DPI)