Computer aided engineering prediction of brake noise: modeling of brake shims

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CAE Prediction of Brake Noise: Modelling of Brake Shims

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Abstract

Brake shims, applied to brake pads, are used for suppressing high frequency noise in disc brake units. Also called brake insulators, they do so mainly by adding more damping to the system in the brake pad area. This reduces the likelihood of the energy transfer between the components which would cause modal coupling.

Finite Element Analysis (FEA), as a simulation and analysis technique, is widely used in the industry to perform squeal analysis as a part of the virtual development of new brake units. However, in most Computer Aided Engineering (CAE) simulations of brake noise, shims are modelled as thin sheets of steel or are not modelled at all. This introduces some inaccuracy due to ignoring the damping effect and flexibility of the rubber and adhesive material. Such inaccuracy in predicting system behaviour, in the virtual design stage, means the analyst may not be able to locate the right frequencies of any occurring instability in order to decide on a noise fix. Also, the over-prediction of instabilities by Complex Eigenvalue Analysis (CEA) adds to the inaccuracy of the process.

This paper introduces a simplified CAE model for the brake shim, which when implemented in brake system modelling helps in highlighting the actual frequencies at which instability occurs in the system by taking into account the correct level of damping in the system in the virtual design stage. The method is confirmed by correlating the analysis predictions with the noise performance of a brake unit in dynamometer tests.

Keywords

Brake noise, Brake insulator, Computer Aided Engineering, Finite Element Analysis, Complex Eigenvalue Analysis, Damping

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1. Introduction

Disc brake noise has always been a major problem for the automotive industry. Many years of research and development have resulted in substantial improvements. However, the customer expectation has also increased in the recent years. (Papinniemi et al., 2002)

Recent developments in the Computer Aided Engineering (CAE) specially the Finite Element Analysis (FEA) method have resulted in increased capability to design brakes which perform better from the noise point of view. However, not all aspects of the phenomenon are easy to model, damping characteristics being an example. Modal coupling is a known cause of brake noise, and accurate estimation and simulation of the system damping can significantly reduce the design and development process by predicting the modal couplings at the virtual design and analysis stage.

Estimation and simulation of damping characteristics of the major components of the brake unit was investigated in the previous study by the authors (Esgandari et al., 2013a). The current study focuses on the CAE modelling of brake shims.

2. Review of the literature

2.1. Brake noise problem in the advance CAE era

Brake noise has been studied since late 1930's (Lamarque and Williams, 1938) and there has been considerable improvement in terms of decreasing the noise level (Papinniemi et al., 2002). However, there has always been more attention given to other aspects of the braking technologies which is understandable considering that brakes are a major safety mechanism of the vehicle (Papinniemi et al., 2002).

Significant improvements in producing quieter brakes were achieved a few decades later, once the CAE tools were employed to simulate the dynamics of the system and analysing the possible instabilities (Papinniemi et al., 2002). The "fugitive nature" of the brake squeal problem is now more comprehensible, by implementation of CAE tools and analysis techniques such as CAE (Complex Eignevalue Analysis (CEA) or transient time domain analysis (Cantone and Massi, 2011; Massi et al., 2010). Also, application of other modern technologies such as ultrasonic methods (Yuhas et al., 2013) has helped in procurement of a deeper understanding of different aspects of the system such as material stiffness. New CAE tools have also facilitated invention of new technologies such as the Partitioned Brake Pad introduced by the author (Esgandari et al., 2013b).

New CAE tools have enabled investigation of the brake squeal in early stages of design process by performing most of the analyses using virtual models. Also, application of other smart numerical/analytical tools such as Artificial Intelligence can help in performing sensitivity studies by post-processing the CAE analysis results (Parra et al., 2010; Solaymani Roody, 2011).

2.2. Significance of damping

Numerous studies assume modal coupling to be a major cause of brake noise (Mottershead, 1998; Ouyang et al., 2005; Ouyang, 2006). It is possible to decouple the modal interaction and eliminate the dynamic instability by identifying and shifting resonant frequencies of the major components (Kung et al., 2000). By accurately modelling the components of the system and performing the CEA the resonant frequencies of unstable modes can be identified. Precision of the simulation mainly depends on the material properties of the system, as well as the interacting boundary conditions. In the CAE model, system damping plays a dominant role in simulating the potential modal couplings by replicating the realistic level of vibration amplitudes for the interacting components. Therefore it is significantly important to tune the system damping prior to performing the CEA. This also limits the instability over-prediction which is a known disadvantage of the CEA (Massi et al., 2007).

2.3. Shim modelling

Shim provides a flexible connection between the piston and the brake pad and is therefore capable of changing the modal response. Modelling of brake shim requires a comprehensive knowledge of material properties of different layers of the shim, some of which are not easy to measure. Also, precisely modelling the boundary conditions of the layers requires an in depth understanding of how these layers interact with each other once in action (Flint et al., 2004; Nishizawa et al., 2007; McDaniel et al., 2005).

Modelling shims along with other damping tuning techniques improves CEA analysis accuracy by eliminating instability over predictions. Accurate modelling of the shim is significant in assessing NVH performance of the brake unit, as it helps to identify instabilities which are caused by modal coupling where the shim is unable to provide enough damping to facilitate modal decoupling. This enables application of required structural noise fixing techniques in the CAE phase, before dynamometer or vehicle tests.

3. Methodology

The main aim of this study is to further develop a routine for damping tuning of the brake system CAE model in order to analyse instabilities of the system more accurately. This can result in significant improvement in clearer estimation of the potential of modal couplings to cause brake noise. Consequently, modal decoupling can be done by performing geometrical alteration or changes in the material properties to shift the resonances. Various modal decoupling techniques are practiced in designing brakes, which are selected based on the specific frequency range of the modal coupling and the components involved (Saw Chun Lin, 2009).

The method of estimating system damping and performing the squeal analysis using CEA is the same as in the previous study by the authors (Esgandari et al., 2013a). This section reviews the damping estimation in three levels namely material damping, contact damping and shim damping.

3.1.Material damping

Based on the Rayleigh damping tuning techniques (Esgandari et al., 2013a), material damping for major components of the brake unit has been identified. This includes brake disc, pad back-plates, caliper, and knuckle. Figure 1 to Figure 4 visualise the Rayleigh damping estimation compared to the material damping obtained from the modal extraction experiment. This comparison demonstrates that the estimated Rayleigh damping is within an acceptable range of variation from the experimental data. However, level of damping at different frequencies depend on the formulation of Rayleigh damping and the current Rayleigh damping graph is the best curve fitting based on the experimental data.

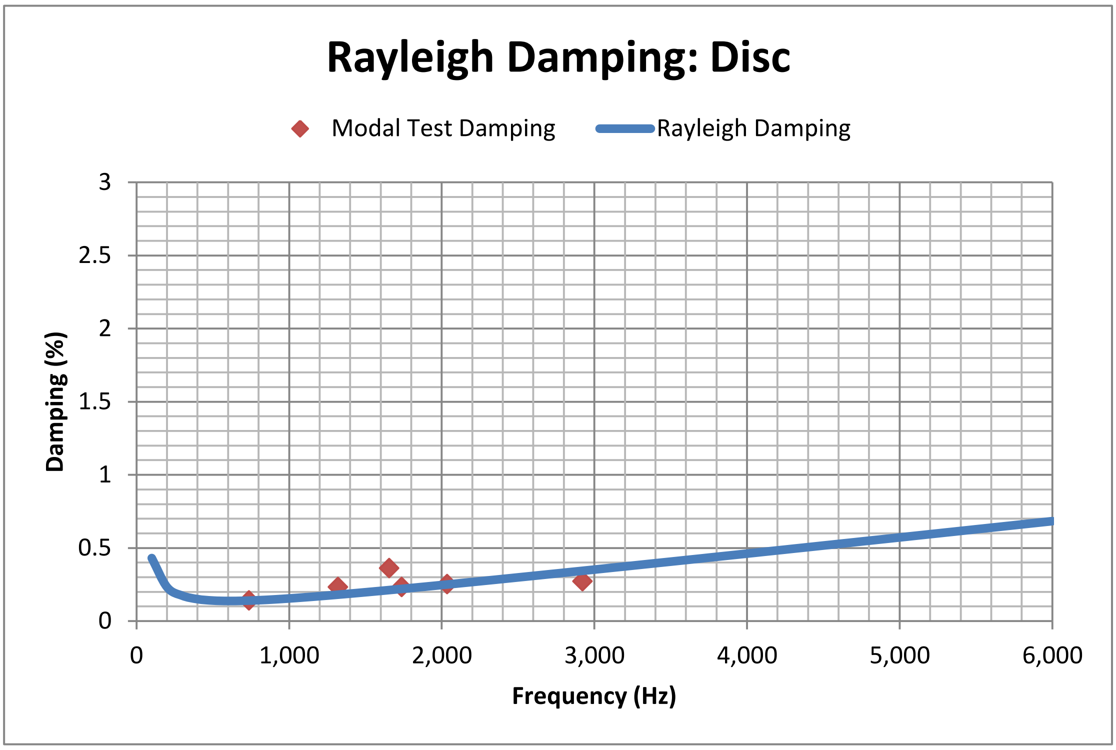

Figure 1, Rayleigh damping curve vs. actual test data - disc

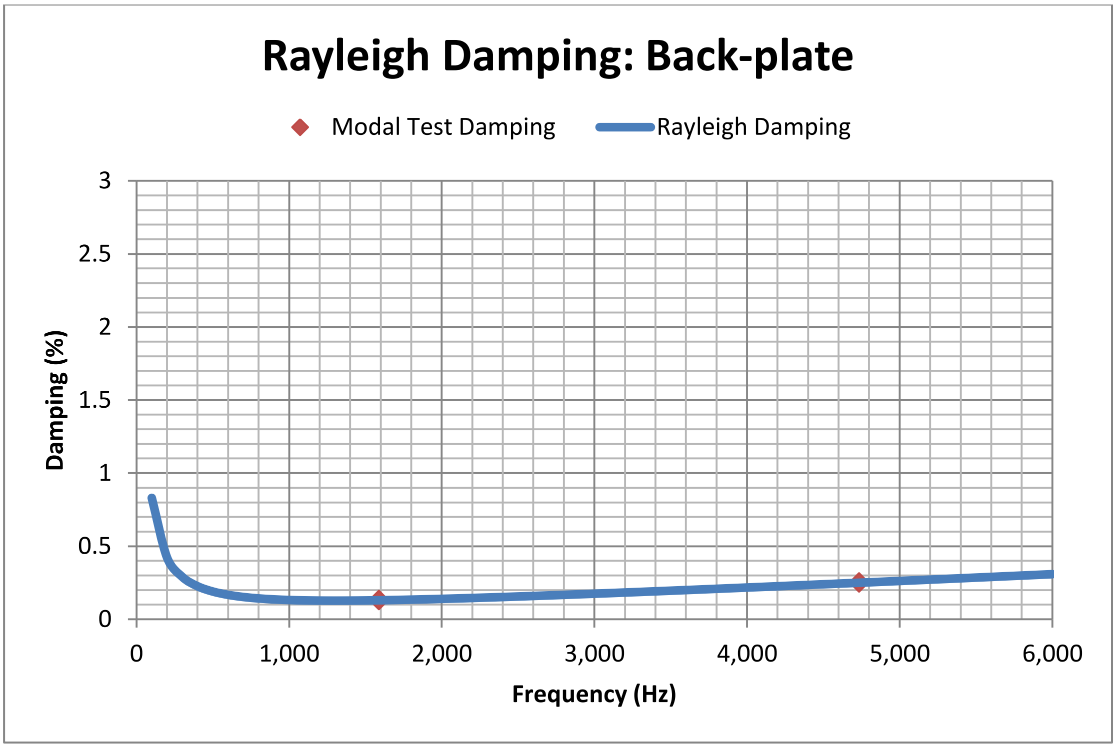

Figure 2, Rayleigh damping curve vs. actual test data - back-plate

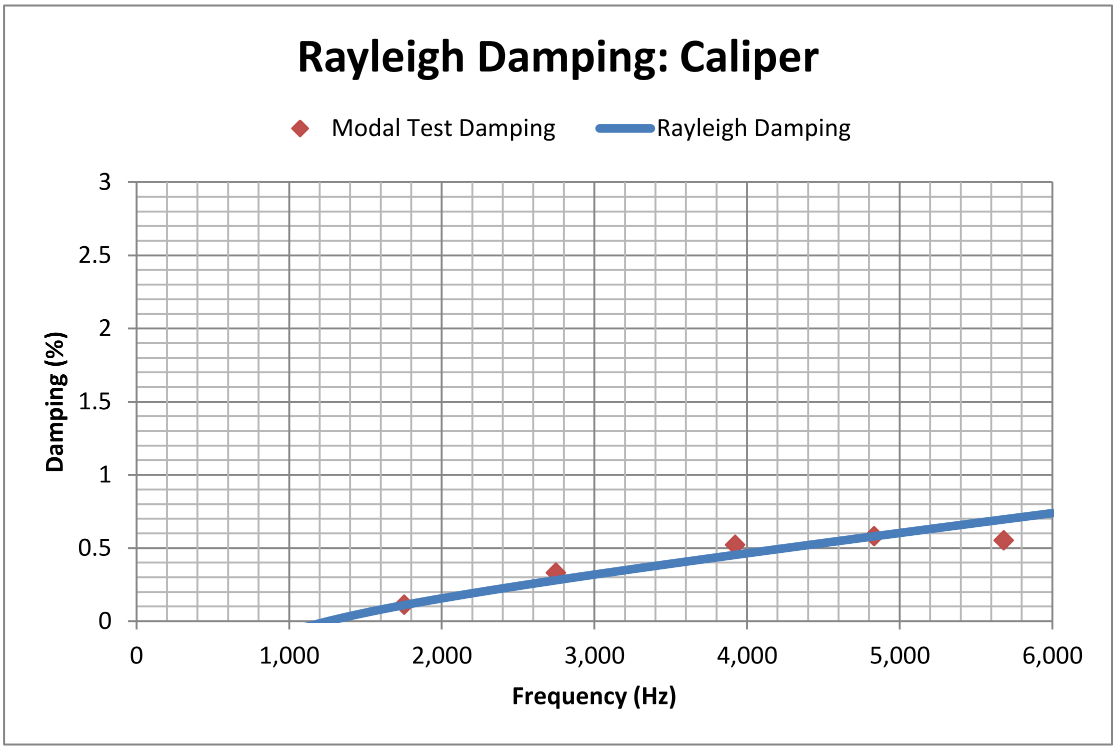

Figure 3, Rayleigh damping curve vs. actual test data - caliper

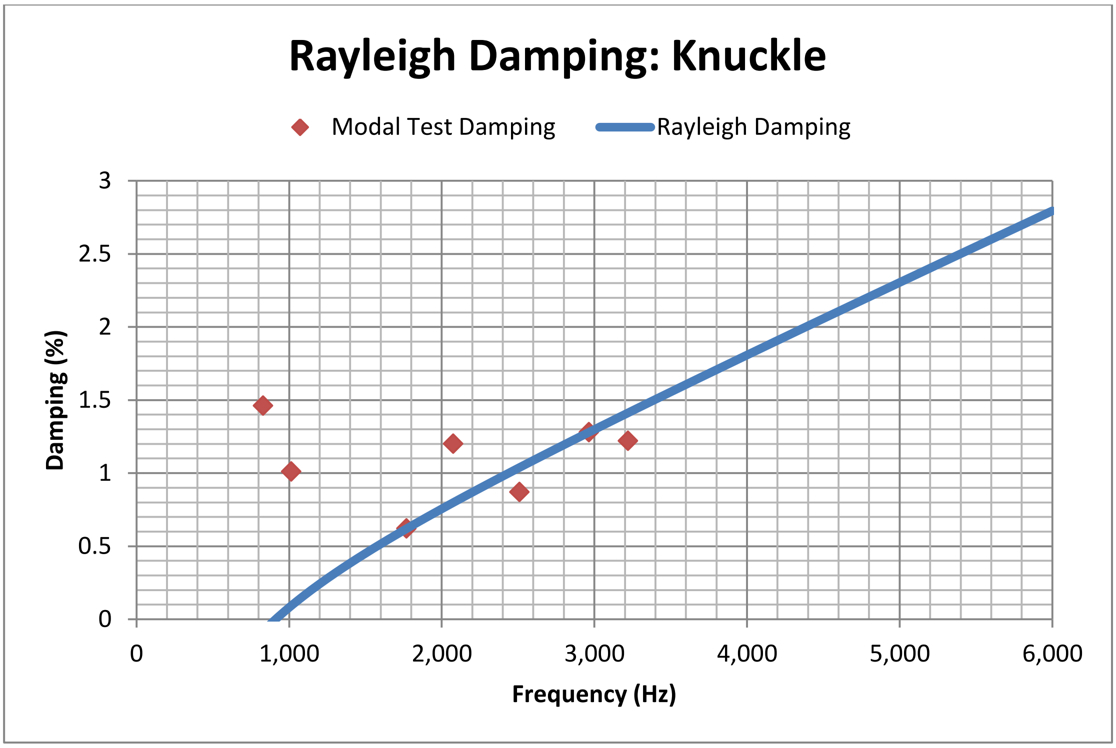

Figure 4, Rayleigh damping curve vs. actual test data - knuckle

3.2.Contact damping

As a part of the system damping estimation, a case study was performed to compare the significance of the contact damping with material damping. Quantifying contact damping is difficult to achieve in terms of designing an experiment to measure it, therefore it is estimated indirectly by measuring damping in an assembly. Assembly damping is made up of the material damping of the components plus the contact damping, and therefore can be used to estimate the significance of the contact damping.

An investigation was carried out to compare the assembly damping of caliper and knuckle with the material damping of the individual components. Figure 5 presents a comparison of the assembly damping and material damping, which confirms that the contact damping is less significant compared to the material damping. However, it does not quantify it.

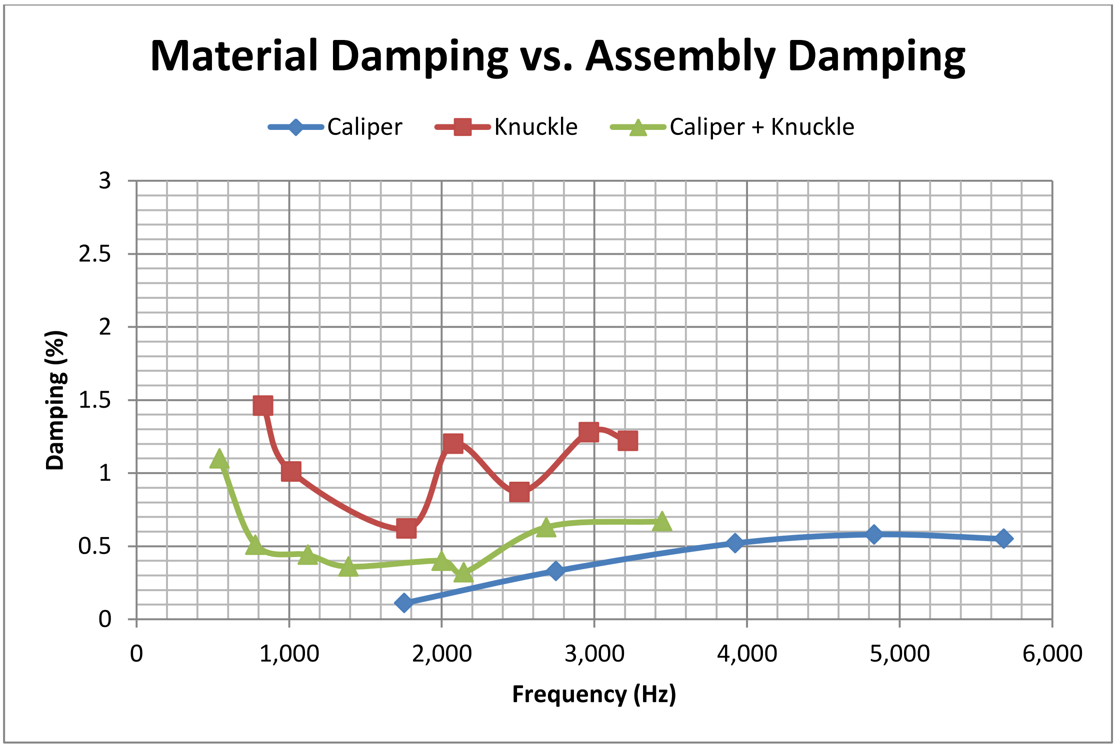

Figure 5, Material damping vs. assembly damping - Study of contact damping

Although an estimation of significance of the contact damping is given, this study is only assuming the material damping for performing the CAE damping tuning of the brake FEA model. Although the contact damping might be substantial in accuracy of the instability prediction, authors believe material damping is considerably more significant in the accuracy of instability prediction. Therefore, damping tuning is performed only at the component level.

3.3.Shim damping

Brake shims, due to their multi-layer structure and the characteristics of rubber and adhesive material, are difficult to model. Furthermore, the complexity involved in the material damping and boundary conditions of the adhesive material can significantly increase the analysis time beyond what is acceptable for the industry from a cost point of view. This study aims to introduce a representative model for the brake shim which can replicate the dynamics of brake shims while keeping the system complexity at an acceptable level.

Shims are formed of rubber material, adhesive, and usually a sheet of steel in the middle (McDaniel et al., 2005). The simplified model developed in this study is formed of three layers; two layers of rubber on either side and a sheet of steel in the middle. Figure 6 represents a schematic of the shim model developed.

Figure 6, Schematic of three layer shim design

Adhesive material is not simulated in this simplified model. Rubber parts are modelled as hyper-elastic materials in order to replicate the nonlinear elastic behaviour. Hyper-elastic material property does not account for the material damping; however, it does account for the flexibility of the material. This can help replicating the modal characteristics of the assembly more accurately by shifting resonant mode frequencies. The shim is initially modelled without any material damping, and then the hyper-elastic rubber material is damped in order to achieve a more accurate estimation of strength of instability from the simulation.

Rubber material damping can be obtained by performing stress relaxation experiments. With reference to the study performed by Flint *et al.* (Flint et al., 2004), this study uses Rayleigh damping technique and assumes damping level of $\alpha = 0$ and $\beta = 3.75 * 10^5$. They have measured the loss factor and shear modulus using an experimental setup and have correlated that with the Rayleigh damping simulation with FEA, which shows an acceptable correlation.

3.4. Instability prediction and ranking

In terms of the CAE model of the brake corner unit, the main components included in the brake corner unit are disc, caliper housing including pads and shims and pressurising pistons, the hub and knuckle. Also two mass blocks are added to the caliper as a brake noise fix by shifting the modes of the caliper to avoid modal coupling. Figure 7 presents a schematic of the brake corner unit used in this study.

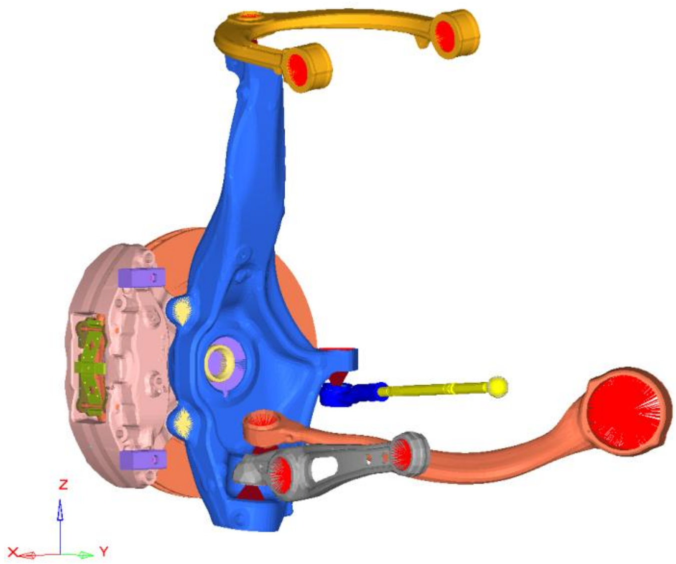

Figure 7, Brake corner unit CAE model, isometric view

Among different analytical approaches to investigating brake noise, CEA is chosen in this study. However, there is a debate on the best analytical approach (Ouyang, 2006; Nouby and Srinivasan, 2009) which was reviewed in the previous study (Esgandari et al., 2013a).

In terms of instability ranking or identification of strength of instability, CEA results can be represented either in terms of the eigenvalue real part or the negative damping ratio. Reviewing how these are different from each other (Ouyang, 2006; Ouyang et al., 2005; Nouby and Srinivasan, 2009) and their proportionality (Abubakar, 2005; Wallner et al., 2010) in the previous study, negative damping ratio was chosen as the measure of the strength of instability.

A full set of squeal analysis is performed on the tuned model using the CEA. This set of analyses includes friction values of 0.3, 0.4, 0.5, 0.6 and 0.7. Pressures analysed are 2, 5 and 10 bar. Each combination of friction and pressure are analysed in both forward and reverse directions. All different cases mentioned, summing up to 30 CEA analyses, are assumed to form one full set of squeal analysis (Nouby and Srinivasan, 2009; Massi et al., 2007). Figure 8 presents the analysis results of the baseline model which excludes application of material damping and shim.

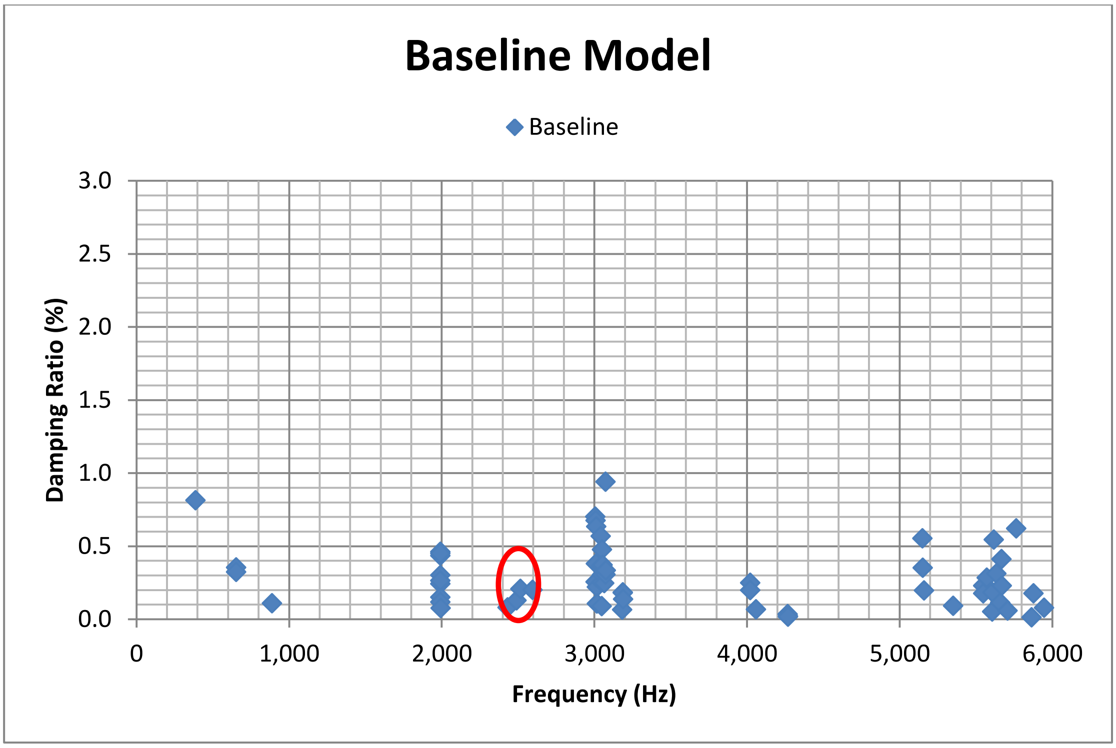

Figure 8, Squeal analysis: baseline (un-damped) model without shim

Analytical results need to be correlated with the real-world noise performance of the brake system. This also enables evaluation of the instability prediction improvements. The brake corner unit was tested in the brake dynamometer undergoing SAE J2521 (International, 2006) noise search procedure. Figure 9 presents the dynamometer test results which indicates above-acceptable noise levels recorded at frequencies around 2.45-2.55 kHz. Since this is the only noisy frequency, the tuned CAE model is expected to clearly show this and not report instabilities at other frequencies.

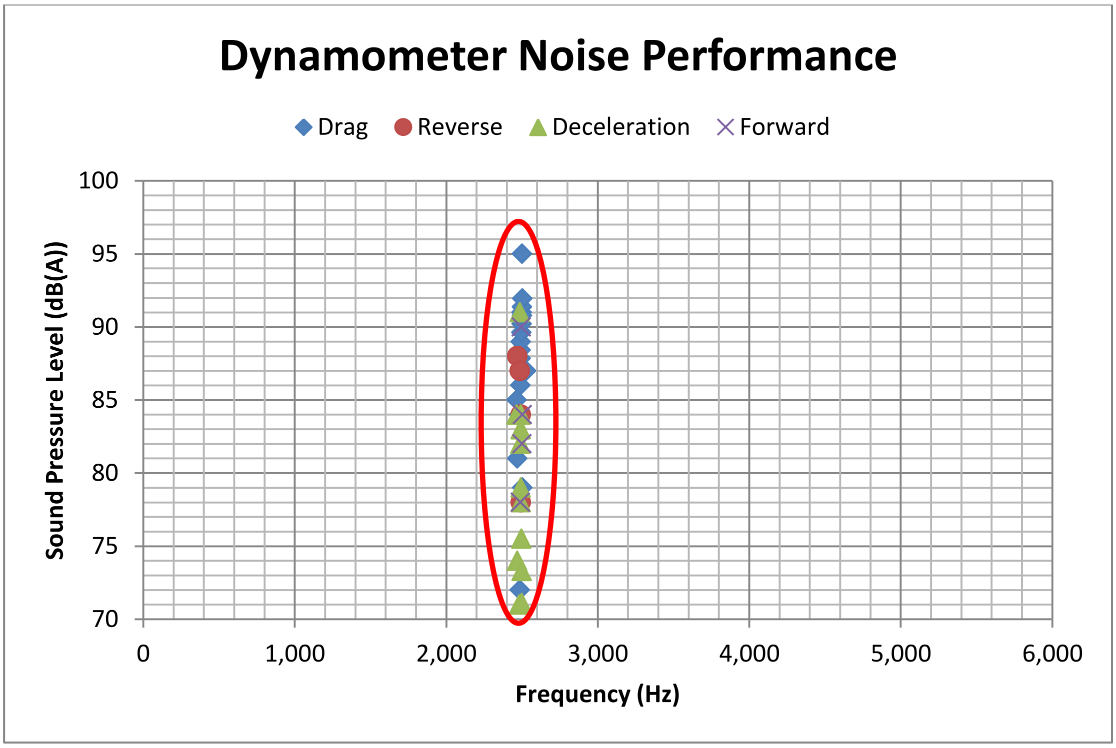

Figure 9, Noise performance on dynamometer undergoing SAE J2521 standard test

4. Results and discussion

4.1. Squeal analysis: Rayleigh damping of major components

Based on the methodology mentioned, the brake CAE model was tuned with the right level of damping for each component. A full squeal analysis was then performed in order to study the effects of damping in eliminating the instability over-predictions. Analysis results are compared with the baseline model. Comparison of the results presented in Figure 10 with the dynamometer results (Figure 9) shows that most of the over-predicted instabilities are now eliminated. Results from the baseline model show instabilities in frequencies of 0.38, 0.88, 1.4, 2.5, and 3.05 kHz.

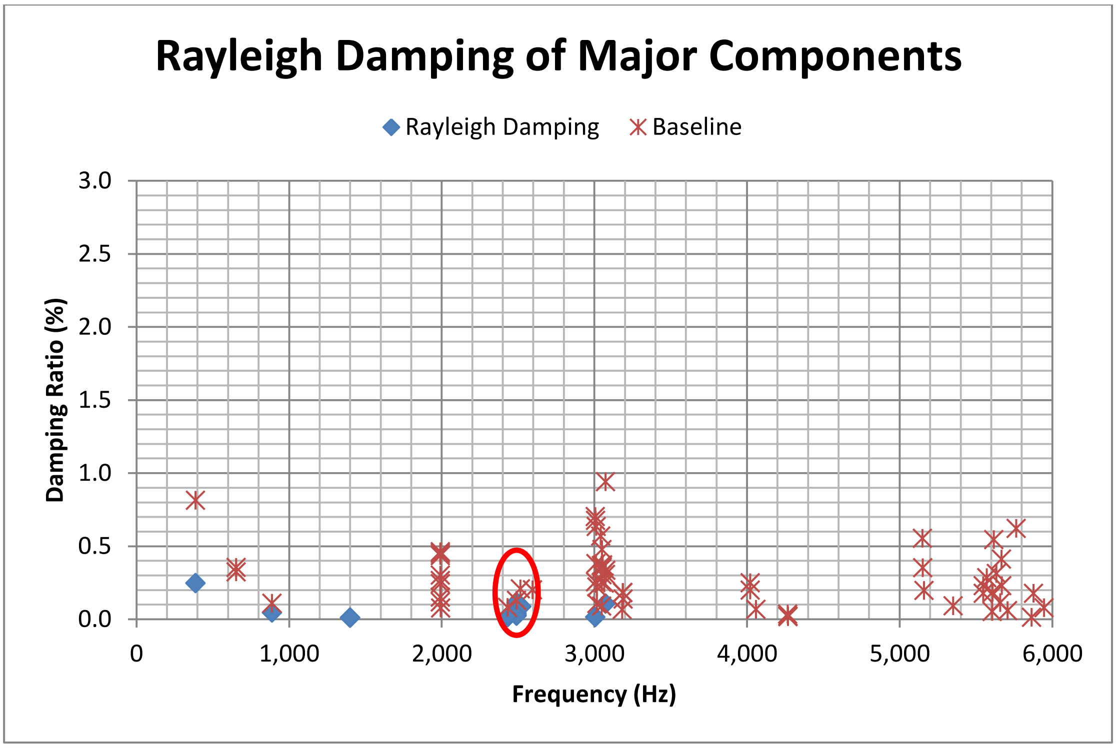

Figure 10, Squeal analysis: damped model vs. un-damped model

Although more repetition of instabilities predicted at 2.5 kHz can be counted as an indication of this frequency being the major instability, the instability prediction needs to be clearer in terms of differentiating the actual instabilities from the over-predictions. One solution could be evaluating the damped and un-damped models with different target lines (Esgandari et al., 2013a).

4.2.Squeal analysis: addition of three layer shim

In the next set of analysis, the 3 layer shim was added to the CAE model which was tuned using the Rayleigh damping technique. Figure 11 compares the squeal analysis results of the new model with the baseline model.

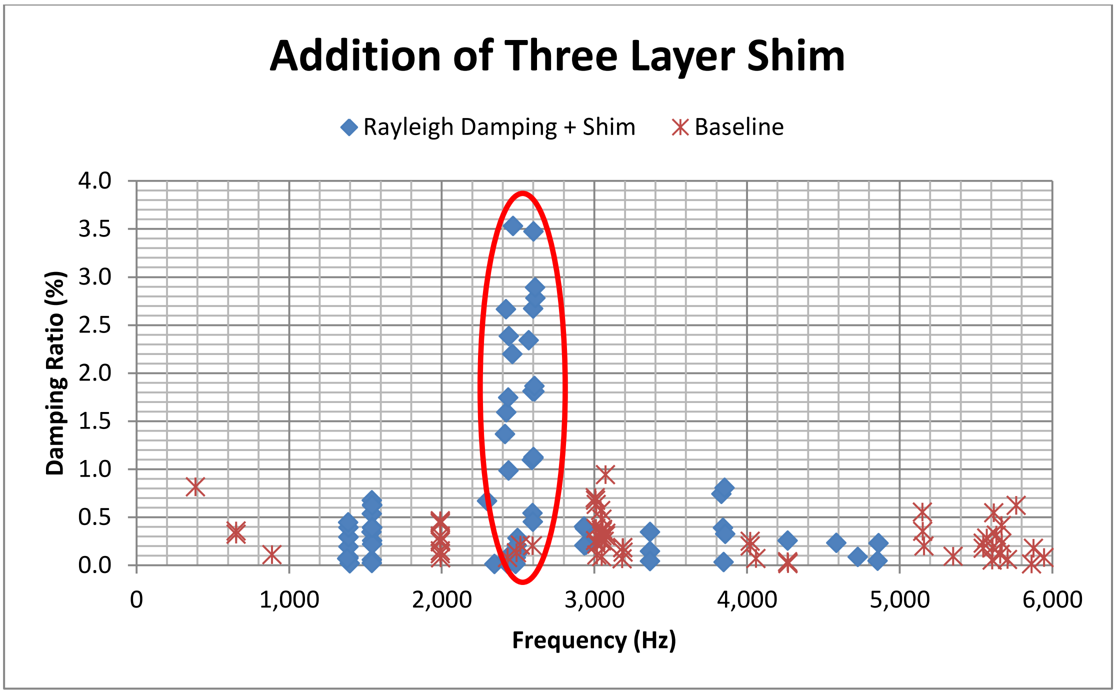

Figure 11, Three layer shim without damping tuning on the damped model vs. un-damped model.

As seen in Figure 11, the new modelling technique tangibly highlights the actual unstable frequency range, with a significant difference in both strength of reported instabilities and their repetition. The new model highlights instabilities in frequencies of 2.4-2.6 kHz with a damping ratio of up to 3.5%. However, there are still instabilities predicted at other frequencies with relatively significant repetitions. An example of this is 1.4-1.5 kHz frequency range, where new over-predicted instabilities are added. This is where damping tuning of the shim becomes critical. Application of damping on the shim can reduce the strength or number of repetition of over-predictions.

4.3.Squeal analysis: damping tuning of the shim

As a major source of damping in the system, the brake shim also needs to be tuned with the right level of damping. The steel section has a less significant damping compared to the rubber part. Therefore, in order to minimise the required computing, the steel section is not damped.

The rubber material is assigned Rayleigh damping of $\alpha = 0$ and $\beta = 3.75 * 10\text{^}5$, based on the stress relaxation experiments performed in reference (Flint et al., 2004). Figure 12 illustrates comparison of analysis results of this model with the baseline.

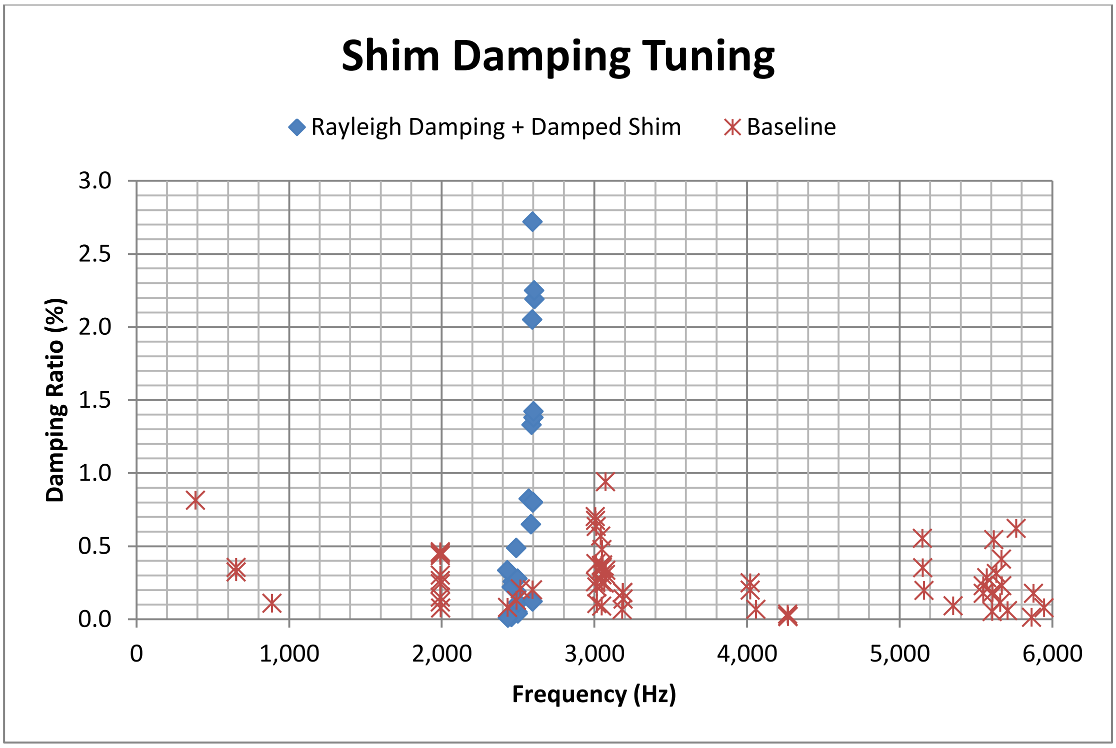

Figure 12, Three layer shim with damping tuning on the damped model vs. un-damped model

As seen in Figure 12, application of damping on the rubber section has significantly improved the instability prediction, and the CAE results are now thoroughly correlating with the dynamometer results. The over-predicted instabilities are completely eliminated, specially those at 1.4-1.5 kHz and 3.8 kHz which seemed to be mainly due to lack of damping in the shim.

5. Summary and conclusions

5.1. Study conclusions

Damping tuning of the brake corner unit CAE model is investigated using the Rayleigh damping method introduced in the previous study (Esgandari et al., 2013a). The damping tuning seemed to eliminate the majority of over-predicted instabilities.

A simplified yet representative modelling technique for the brake shim is introduced. The new modelling technique demonstrates the damping characteristic of the shim without adding unnecessary complexities to the system.

The new shim model is analysed and results are correlated with the dynamometer test results of the same brake unit. Results indicate very accurate prediction of the unstable frequency.

5.2. Future work

Damping characteristic and hyper-elastic properties of the shim is expressively dependant on the operating temperature, due to the rubber and adhesive material’s sensitivity to temperature. The back-plate temperature seems to be an influential factor in modelling shims, as they are in direct contact. Back-plate temperature could be used as a guide to read in the damping values for shim from the shim map. This requires addition of thermal data to the model.

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