DAMPING TECHNOLOGIES FOR AUTOMOTIVE PANEL STRUCTURES

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Abstract

Nowadays, particular attention is devoted to the vibroacoustic comfort of vehicles in automotive industry. Various panels in the automobile, which are liable to vibration problems, are often treated with damping measures. A prevalent method is to attach viscoelastic materials (VEM) on the surfaces of the panels. Nevertheless, this approach has some drawbacks like marked weight and limited effective range in frequency domain.

This thesis targets developing novel damping approaches for these automotive panel structures with competitive damping performance, while regarding the strict demand on weight-cost efficiency. At the beginning, screening tests were performed from a wide approach spectrum based on various physical principles. The focus was then concentrated to several interesting approaches and finally to an unique approach. The project can be divided into four phases.

In the first phase of the thesis, 35 configurations based on 7 principal damping categories were investigated experimentally on an aluminum beam with the dimension of $200 \times 40 \times 1$ mm. A test rig was developed for the measurement in time domain. The results were compared considering the specific damping-weight performance in low frequency ranges. According to the results, the investigation range was diminished to three approaches for the second phase: Interface damping (ID) attains damping with the special arrangement of common VEM at the joint interface. Active constrained layer damping (ACLD) employing piezoelectric elements together with VEM provides a possibility of lightweight damping with high loss factors. Particle damping (PD) exhibits reliable damping with a simple and cost-efficient configuration.

In this second phase, the three approaches were further assessed with a more realistic configuration composed of panel and periphery. Another test rig enabling measurements in a wide frequency range was developed and utilized. The specimen was extended from a beam to an aluminum plate with the dimension of $400 \times 300 \times 1$ mm. The results were compared with respect to the damper type and specimen resonance origins. The sectional damper ACLD and PD show some similar characteristics. They are efficient for some specific plate modes by achieving composite loss factors up to 5%. Some local modal loss factors of the sectional dampers are higher than ID of 2.7-3.6% at these plate modes, while their average damping for all modes is inferior to ID. Conversely, ID accomplishes a better universal damping at every eigenmode in a wide frequency range with loss factors of 2.7-4.2% in comparison to ACLD and PD of 1.3-3.1%. Moreover, extraordinary benefits for the real application can be reaped by ID: The global modes at which substantial vibration is transmitted from periphery
to the plates can be well attenuated with ID.

Thus, ID was selected as the most interesting approach and further characterized in the third phase. Instead of experiments, the influence of the interface’s material and geometry properties on the system vibration behavior was systematically analyzed by means of simulation. It can be concluded that a soft and highly viscoelastic interface with great thickness-width ratio is superior in terms of attenuation. 16 different interface concepts were designed and evaluated on beams in the simulation. Remarkable potential of damping can be envisioned by saving considerable VEM weight concurrently.

Lastly, the feasibility of ID in real environment is exhibited with the simulation on a real vehicle panel: a spare wheel pan (SWP) made of glass mat reinforced thermoplastic (polypropylene) with the volume of ca. $623 \times 657 \times 73 \, \text{mm}$. Butyl rubber ID and four commercial adhesives were applied on the SWP with a periphery frame under same boundary conditions. The vibration behavior of the SWP doesn’t vary significantly with different soft and stiff adhesives, where no pronounced damping can be perceived. Only the butyl ID suppresses the resonances notably with the amplitude decay of 4-12 $\text{dB}$ at all modes. A compromised option, which provides acceptable performance for damping and mechanical requirements, is to combine both structural adhesives and butyl ID at the interface.
Zusammenfassung


In der ersten Phase des Projekts wurden 35 Konfigurationen aus sieben Hauptdämpfungsarten mit Experimenten an einem Aluminiumbalken mit den Abmessungen von $200 \times 40 \times 1\, mm$ aufgegriffen. Ein Prüfstand für die Messung in Zeitbereich wurde entwickelt. Die Ergebnisse wurden im niedrigen Frequenzbereich miteinander verglichen. Angesichts der spezifischen Dämpfung-Gewichtseffizienz, wurde der Untersuchungsumfang auf drei Verfahren reduziert: Schnittstellendämpfung (ID) verringert die Vibration durch die spezielle Anordnung von VEM an der Schnittstelle von Paneel zu Umgebung. Active constrained layer damping (ACLD) erreicht einen hervorragenden Verlustfaktor für die Leichtbaustruktur mithilfe kombinierter Anwendung von VEM und Piezoelementen. Partikeldämpfung (PD) bietet zuverlässige Dämpfung mit einfacher und kostengünstiger Bauform.

höher als ID (2.7-3.6%) bei diesen Plattenmodes, wobei der durchschnittliche Dämpfungsgrad für alle Modes kleiner ist. Demgegenüber leistet ID eine bessere gemittelte Dämpfung mit den Verlustfaktoren von 2.7-4.2% in einem breiten Frequenzspektrum im Vergleich zu ACLD und PD (1.3-3.1%). Zusätzliche Vorteile ergeben sich durch den Einsatz von ID, indem die globalen Modes, bei welchen die wesentlichen Schwingungen von der Peripherie herrühren und zum Paneel übertragen werden, mit ID gedämpft werden können.


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- Last but not least, my whole family - my wife, parents and parents-in-law who make me believe in that every difficulty can be overcome.
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<table>
<thead>
<tr>
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<th>Description</th>
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<tr>
<td>$\alpha(T)$</td>
<td>Arrhenius shift function, 35</td>
</tr>
<tr>
<td>$\beta$</td>
<td>Coefficient of the derivative fractional model, 35</td>
</tr>
<tr>
<td>$\delta$</td>
<td>Damping angle (phase lag in time domain), 12</td>
</tr>
<tr>
<td>$\Delta \omega$</td>
<td>Frequency range of decay in radian, 12</td>
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<tr>
<td>$\Delta \omega_{3\text{dB}}$</td>
<td>Frequency range of 3 dB decay in radian, 23</td>
</tr>
<tr>
<td>$\Delta T$</td>
<td>Decay time, 12</td>
</tr>
<tr>
<td>$\Delta U_h$</td>
<td>Dissipated energy in one cycle, 12</td>
</tr>
<tr>
<td>$\Delta x_{dB,f}$</td>
<td>Amplitude decay of $x$ dB in frequency domain, 12</td>
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<td>$\Delta x_{dB,t}$</td>
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<tr>
<td>$\eta$</td>
<td>Loss factor, 12</td>
</tr>
<tr>
<td>$\eta_c$</td>
<td>Composite / structural loss factor, 12</td>
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<td>$\eta_m$</td>
<td>Material loss factor, 12</td>
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<td>$\sigma_{max}$</td>
<td>Max. stress, 12</td>
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<tr>
<td>$\tau_{critical}$</td>
<td>Maximal shear stress, 149</td>
</tr>
<tr>
<td>$\zeta$</td>
<td>Damping ratio, 12</td>
</tr>
<tr>
<td>$A$</td>
<td>Area of bonding, 149</td>
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<tr>
<td>$a_1$</td>
<td>Coefficient of the derivative fractional model, 35</td>
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</tbody>
</table>
\begin{tabular}{ll}
\textbf{\(b_1\)} & Coefficient of the derivative fractional model, 35 \\
\textbf{\(b_2\)} & Coefficient of the derivative fractional model, 35 \\
\textbf{\(c\)} & Viscous damping coefficient, 14 \\
\textbf{\(c_1\)} & Coefficient of the derivative fractional model, 35 \\
\textbf{\(c_2\)} & Coefficient of the derivative fractional model, 35 \\
\textbf{\(F_0\)} & Max. magnitude of the force excitation, 13 \\
\textbf{\(f_{res}\)} & Resonance frequency, 12 \\
\textbf{\(g^*\)} & Shear parameter, 87 \\
\textbf{\(G_{hyst}\)} & FRF of hysteretic damping system, 16 \\
\textbf{\(G_{visc}\)} & FRF of viscoelastic damping system, 15 \\
\textbf{\(Im(E^*)\)} & Imaginary part of the complex modulus, loss modulus, 34 \\
\textbf{\(j\)} & Imaginary unit, 16 \\
\textbf{\(k\)} & Stiffness coefficient of the system, 14 \\
\textbf{\(m\)} & Mass of the system, 14 \\
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\textbf{\(p\)} & Total number of measured points, 20 \\
\textbf{\(Q\)} & Quality factor, 12 \\
\textbf{\(r\)} & Frequency ratio, 14 \\
\textbf{\(Re(E^*)\)} & Real part of the complex modulus, storage modulus, 34 \\
\textbf{\(S\)} & Total energy at a mode, 140 \\
\textbf{\(s\)} & Laplace variable, 15 \\
\textbf{\(S_{critical}\)} & Safety factor, 149 \\
\textbf{\(T\)} & Temperature, 35 \\
\textbf{\(T_0\)} & Reference temperature, 35 \\
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\textbf{\(T_R\)} & Reverberation time, 24 \\
\textbf{\(T_{1/2}\)} & Time of decay 1/2 amplitude (6 dB), 24 \\
\textbf{\(x_0\)} & Max. strain, 12 \\
\textbf{\(x_h\)} & Homogeneous solution, 14 \\
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\( x_p \)  Particular solution, 14  
\( Y_A \)  Accelerance, 18  
\( Y_D \)  Receptance, 18  
\( Y_V \)  Mobility, 18  
\( Z_A \)  Inertance, 18  
\( Z_D \)  Dynamic stiffness, 18  
\( Z_V \)  Impedance, 18
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ACLD  Active constrained layer damping, 29
BBD   Bean bag damper, 30
CLD   Constrained layer damping, 4
DEM   Discrete element method, 30
ECD   Eddy current damper, 31
EM    Electromagnet, 56
EMCLD Electro-magnetic constrained layer damping, 30
FE    Finite element, 89
FEM   Finite element method, 30
FLD   Free layer damping, 4
FPBW  Fractional power bandwidth method, 22
FRF   Frequency response function, 18
GMT   Glass fiber mat reinforced thermoplastic, 132
HPBW  Half-power bandwidth method, 23
MCLD  Magnetic constrained layer damping, 30
MDOF  Multiple-degree-of-freedom, 13
MR    Magnetorheological, 32
NOPD  Non-obstructive particle damper, 30
NVH   Noise, vibration and harshness, 1
PET   Polyethylene terephthalate, 60
PI    Polyimide, 61
PIX   Performance index, 140
PM    Permanent magnet, 56
<table>
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<tr>
<th>Acronym</th>
<th>Description</th>
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<tr>
<td>RMS</td>
<td>Root mean square</td>
<td>20</td>
</tr>
<tr>
<td>SDOF</td>
<td>Single-degree-of-freedom</td>
<td>13</td>
</tr>
<tr>
<td>VEM</td>
<td>Viscoelastic material</td>
<td>2</td>
</tr>
<tr>
<td>SWP</td>
<td>Spare wheel pan</td>
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Chapter 1

A Brief Overview

In this chapter, a general overview of the thesis is given. The background of the vehicle vibrations is stated. Prevalent damping materials and treatments are introduced. The scope of the research including the current problems, requirements and objectives are enumerated. The whole research phases over time and the structure of this thesis are elucidated at the end.

1.1 Introduction

In the automotive industry, customer demands in terms of acoustic comfort is continuously increasing in the past decades. Thus, NVH (noise, vibration and harshness) reduction is gaining more prominence in recent years. In order to improve the NVH behavior of the vehicle, concerning each component individually is insufficient. The comprehensive consideration with adjacent components and periphery is important. Individual excitations are transmitted through various paths to the auto body and generate unpleasant noise and vibration in the interior of the vehicle [1, 2].

In table 1.1, various oscillatory effects, which significantly deteriorate the ride quality and driving comfort of the passengers, are listed. The majority of them occur at low frequencies. Up to 50 Hz, the low frequency excitations with a wavelength of 90-150 mm are especially deleterious to the ride comfort [3].

These kinds of transmissions of vibrations into the vehicle panels are ubiquitous. E.g., body floors are typical structures that are prone to vibrations and should be treated to tackle multifarious vibration and acoustic problems. By using different design and materials (polymer, foam, fibre, carpet, etc.), acoustic refinements can be achieved on these structures like reduction of the perceiv-
Table 1.1: influential aspects with respect to the ride comfort [4, 3]

Chapter 1. A Brief Overview

The topic of this thesis is focused on the damping techniques of the automotive panels. Different aspects in this field were investigated, e.g., comparison of the existing and potential damping treatments, search for novel damping approaches and complete the insight of the whole vibration transmissions, in order to improve the vibration behavior of these panels.

1.2 State-of-the-art

Vibration suppressions can be achieved through various damping materials and mechanisms. As an economical solution, viscoelastic damping materials are applied in over 85% of the passive damping treatments [6]. In vehicles, viscoelastic material (VEM) is conventionally applied in surface damping treatments.
1.2. State-of-the-art

1.2.1 Damping materials

From the pure material point of view, following substances can be used for the damping purpose [7]:

- metals: shape-memory alloys, ferromagnetic alloys
- polymers: mainly rubbers
- ceramics: typically cements or concretes

The damping efficiency of ceramics is generally inferior to the other two sorts. Polymers have the highest efficiency among all of them. A concrete list of the loss factors of some typical materials can be also found in [8]. Furthermore, due to the cost, polymers have become the most popular damping materials for the automotive engineering. The most widely applied polymers are bitumen and butyl-based (see figure 1.1). In many cases, butyl rubber performs better than bituminous damping material. Both of them are prevalently available in the form of damping sheets by many suppliers [9, 10, 11, 12, 13, 14, 15]. The ”spray dampers”, which is applied by robotic equipments, is another variation of commercial damping products [16].

![Figure 1.1: typical damping materials: bitumen based material (left) and butyl rubber (right)](image)

1.2.2 Damping approaches

In order to mitigate the vibrations on the automotive panels, their surfaces are typically partially covered with the introduced viscoelastic material like butyl rubber. Substantially, two approaches are applied: free layer damping (FLD) and constrained layer damping (CLD) [17, 6]. As depicted in figure 1.2: In FLD, the panel is simply covered with one layer VEM. Attributable to the generated extensional / compressional stress in the VEM layer during the vibration, energy
is dissipated. In CLD, one additional layer (typically metallic), which has a much higher stiffness and normally lower thickness than the VEM layer, is glued onto the VEM layer. In this case, the bending deformation of the plate induces dominantly shear stresses in the VEM layer. The performance of the CLD is enhanced significantly compared to FLD. These two approaches are cost-efficient and provide remarkable damping.

![Figure 1.2: traditional damping techniques](image)

Many industrial products enhancing NVH behavior have been developed based on the configuration of CLD for the automotive application. Multilayer composites which have viscoelastic cores are provided by [18, 19]. Such sandwich composites can be typically applied on the structures displayed in figure 1.2.

In various literatures, comprehensive formulations and models of CLD treatment have been reported [21, 22, 23]. The damping effectiveness like composite loss factor was quantitatively predicted. Optimizations with respect to the ge-
1.3. Scope of the research

This thesis is highly oriented on vehicle engineering. Both interests of industrial applicability and academic novelty are regarded for the research scope.

1.3.1 Research objective

Although FLD and CLD have been rapidly developed and are successfully applied in the recent decades, like all other damping approaches, they have also their own drawback. Since only locations where the dampers are bonded can be damped, if a big surface must be mitigated, the whole surface including damping layer tend to have a heavy weight, which is a limiting factor for the potential of lightweight design of cars. The question is, if the weight of the surface damper can be reduced or if there are any other surface treatments which are lightweight and possess similar damping efficiency concurrently.

Another point caused by the host structure is that, in practical applications, the damping of the CLD surface treatment is decent only in high frequency range where the eigenmodes of the individual panel are excited. Otherwise, the damping is rather light in low frequencies until about 150 Hz where global modes of the vehicle structure prevail (typically the first bending or torsion mode of the vehicle body is in the range of 25-45 Hz) [28]. The reason is that at these global modes, primary deflections occur in the auto body, which transmits the vibration further to the receivers: the panel structures. These panels, which are bonded at all their edges to the auto body, vibrate with their peripheral components in quasi self-rigid behavior. But the eigen bending deformations on

<table>
<thead>
<tr>
<th>Engines and powertrains</th>
<th>Body structures</th>
<th>Brakes and accessories</th>
</tr>
</thead>
<tbody>
<tr>
<td>oil pans</td>
<td>dash panels</td>
<td>brake insulators</td>
</tr>
<tr>
<td>valve covers</td>
<td>door panels</td>
<td>backing plates</td>
</tr>
<tr>
<td>engine covers</td>
<td>floor panels</td>
<td>brake covers</td>
</tr>
<tr>
<td>push rod covers</td>
<td>wheelhouses</td>
<td>steering brackets</td>
</tr>
<tr>
<td>transmission covers</td>
<td>cargo bays</td>
<td>door latches</td>
</tr>
<tr>
<td>timing belt covers</td>
<td>roof panels</td>
<td>window motors</td>
</tr>
<tr>
<td>transfer case covers</td>
<td>upper cowl</td>
<td>exhaust shields</td>
</tr>
</tbody>
</table>

Table 1.2: application of CLD products in automotive structures [20]

ometry and material properties of the constraining and damping layer have been systematically performed [24, 25, 26, 27].
the panels are limited. Since there are barely other effective damping measures that can be disposed on the panels, one possibility that remains is to influence their vibration behavior through their interfaces with other components in the auto body. The question here is, if sufficient damping can be introduced into the panels by the joints and how far is it viable.

Thus, in order to overcome the aforementioned drawbacks of FLD and CLD with the emphases of high damping-weight performance and reliability in a wide frequency range, this thesis aims to tackle two main aspects: the survey of competitive innovative surface treatments and feasibility study of damping at the interface of a panel. Goals of the research can be concretely summarized as the following points:

- to have a better insight and overview of damping approaches which can be applied on vehicle panels
- to develop novel dampers which overcome the drawbacks of classical CLD dampers
- to assess the possibility of introducing damping via the interface
- to reduce the weight of the damper by reaching a high damping performance
- to improve the damper behavior at low frequencies, with the particular attention to the global modes
- to utilize the developed damping treatment on a real automotive panel and to demonstrate its performance

1.3.2 Research novelty

In the vibration and damping community, varieties of sophisticated theories, models and damping treatments have been developed and reported. Particularly, classical VEM and CLD have been studied profoundly. However, systematic analysis and review of damping technologies for vehicle panels are difficult to find. In most literatures, a damper is investigated under specific boundary conditions. Comprehensive experimental comparative study covering a large scope of damping materials or mechanisms under similar boundaries is scarce.

The studies presented in this thesis are firmly related to the practice of automotive applications. The original contribution of this thesis are made as following:

- A comprehensive review and experimental comparison of damping approaches regarding the attenuation-weight performance.
• Comparable investigations of some non-conventional damping treatments on a panel under realistic environment of vehicle panels.

• Some detailed observations of particle damping and several suggestions of its practical application on panels under different conditions.

• Systematic research on interface damping which is quantified with respect to both material and geometry properties.

• The investigation of a hybrid interface, which provides a solution satisfying both mechanical and damping requirements of the system.

1.3.3 Research approaches

In this thesis, both experiments and simulations were performed. For the experiments, all damping treatments were measured by two in-house developed test rigs. In order to get a better overview of different damping approaches, the first test rig was applied for the screening of various damping approaches on a relative simple structure at low frequencies. The second test rig facilitates the measurement of more complex structures at high frequencies. It was thus further used for assessing the performances of the novel damping approaches. For a better understanding of the interesting damping treatments, the most important specimens were correlated by simulation models, e.g., the panel model with interface damper. The dynamic behaviors of the investigation objects were analyzed in Hypermesh® and Ansys® with mainly modal analyses (acquisition of mode pattern) and harmonic analyses (acquisition of mode amplitude). These analyses were executed to study the influence of some important system parameters, e.g., the material properties and geometry of the damper. Also the applicability of the damper on a automotive panel was verified by simulation.

The frequency transfer function [8] is an important approach for both experiments and simulations. In this thesis, the mobility, which is a frequently used frequency transfer function, was applied. The results were substantially acquired with it. From the results, the damping values are extracted with fractional power bandwidth method, which is a standard approach documented in [29]. To enhance the robustness of the fractional power bandwidth method, a curve-fitting tool has been developed. This tool is a post-processing procedure, which was programmed by Matlab® codes. It is typically useful to restore a distorted mobility curve according to the classical viscous damping model and thus improve the quality of the damping evaluation.

In additional, diverse tools were developed for different purposes, e.g., optimization of sensor / damper location and visualization of certain results.
1.4 Outline of the thesis

This dissertation can be divided into four main phases over time. In each phase, several damping approaches or miscellaneous studies were conducted. They are described following in section 1.4.1. In order to have a better overview of these investigated damping approaches individually, the structure of this thesis is organized in another sequence sorted by the damping treatment types as described in section 1.4.2.

1.4.1 Project phases

The four phases of the PhD project over time are:

- screening of damping approaches
- experiments of three interesting approaches: particle damper (PD), active constrained layer damping (ACLD) and interface damping (ID)
- further investigation of the focus approach: ID
- application of ID on a real automotive panel

They were conducted from a relative wide range of damping treatments to a narrow spectrum, while the research was gradually deepened and focused on a single damping approach. In the first two phases, the damping approaches were investigated experimentally, whilst in the latter two phases, ID was investigated by simulation. In the experimental part, the specimens are expanded from a simple beam in the first phase to a complex plate structure in the second phase. In the simulation, the FE-models are extended from the plate model (the correlated model with the plate specimen in the second phase) in the third phase to the real automotive panel in the last phase.

1.4.2 Structure of the thesis

The thesis is ordered in the way that each investigated damping approach is introduced in a separate chapter. As the key approach, chapter 7 contains both results of ID from the second and third research phase, while in the other chapters, the results were obtained in only one research phase.

chapter 1 - A brief overview

A brief overview of the PhD thesis, including the background of vehicle vibrations, scope of the research, i.e., the problems, demands, challenges and aims of the investigation and also the structure of the thesis.
1.4. Outline of the thesis

chapter 2 - Theories and models of vibration and damping
The fundamental theories of vibration damping, procedures of determining damping, basic damping approaches and modeling of viscoelastic materials.

chapter 3 - Experimental
The experimental setups of damping measurements in the first and second project phase are described. The first test rig was used for the screening study of chapter 4. The second test rig was applied for the investigations of chapter 5, 6 and 7.

chapter 4 - Screening of damping approaches
In the first project phase, various damping approaches were compared on a simple beam specimen under similar boundary conditions.

chapter 5 - Particle damping treatment
This chapter represents part of the investigations in the second project phase. Experimental results of nine plate samples with particle dampers (PD) are presented.

chapter 6 - Active constrained layer damping treatment
This chapter represents part of the investigations in the second project phase. Experimental results of four plate samples with active constrained layer dampers (ACLD) are presented.

chapter 7 - Interface damping treatment
This chapter represents part of the investigations in the second and third project phase. Experimental results of two plate samples with interface dampers (ID) are presented. A FE-model of the ID specimen is developed and correlated to the experiment. The damping behavior of a simple rectangular interface has been systematically determined by simulation.

chapter 8 - Novel concepts of interface damper
This chapter is a supplementary study of ID with the emphasis on the damper design. Instead of quantifying the damping characteristics of a basic rectangular interface, concepts based on innovative interface geometries and arrangements are comparatively analyzed on a simple beam as the host-structure.

chapter 9 - Comparative study
The experimental results in chapter 5, 6 and 7 are compared based on their respective configurations. The advantages and drawbacks of each approach are contrasted in a narrow scope.
chapter 10 - Application of interface damping on a real panel
An exploration of ID on a vehicle panel - spare wheel pan. Two interface types: pure damping interface and adhesive-damper hybrid interface are evaluated and compared by means of simulation.

chapter 11 - Conclusions and outlooks
A short summary of the work, closing statements and envisaging for the further development.
Chapter 2

Theories and Models of Vibration and Damping

In this chapter, the fundamentals about vibration and damping are to be introduced. The basic equations of vibration are given. The prevalent viscous and hysteretic damping model are compared with each other. Besides the theories, the evaluation algorithm of damping in the experiments are presented. Also some necessary procedures for the exact assessment are interpreted. A short review of the investigated damping approaches is given. The behavior and modeling of conventional viscoelastic materials are elucidated.

2.1 Definition of damping

Damping is defined as the phenomenon in which the mechanical vibration energy is dissipated and converted into other forms, e.g. internal thermal energy, in dynamic systems [8].

There are different types of damping models. Commonly the following damping models are widely used:

- viscous damping: has a damping force dependent on the velocity magnitude
- hysteretic damping: has a frequency independent damping force and capacity.
- coulomb damping: has a damping force independent on the velocity magnitude, but dependent on the sign of the velocity.

The damping can also be described with more complex models like the velocity-$n^{th}$ power damping [30]. But due to the simplicity of mathematical
implementation, frequently the viscous and hysteretic damping model are used in practice. Especially with the linear behavior of viscous damping, analytical solution can be easily obtained to address a variety of problems [31].

Damping can be determined both experimentally and in simulation in different ways. There are many parameters which can be applied for representing the damping value either in a monotone material or in a complex structure. A prevalent parameter reflecting the degree of damping is the loss factor \( \eta \) originated from the hysteretic damping theory. Other parameters can be easily converted into loss factor under the realistic boundary condition if the damping is not heavy (typically \( \eta < 0.2 \)) [32].

In equation 2.1, the conventional expressions and relationships between various damping parameters are presented under the precondition of light damping. These parameters will be introduced individually in the following sections in detail.

\[
\eta = 2\zeta = \tan \delta = \frac{\Delta U_h}{\pi \cdot X_0 \cdot \sigma_{\text{max}}} = \frac{1}{Q} = \frac{\lambda}{\pi} = \frac{\Delta \omega}{\omega_{\text{res}}} = \frac{\Delta x_{\text{dB}}}{27.3 \cdot f_{\text{res}} \cdot \Delta t} = \frac{1}{\sqrt{(10)^{\frac{\Delta x_{\text{dB},f}}{10}}} - 1} \cdot \frac{\Delta \omega}{\omega_{\text{res}}} \tag{2.1}
\]

- \( \eta \): Loss factor
- \( \zeta \): Damping ratio
- \( \delta \): Damping angle (phase lag in time domain)
- \( \Delta U_h \): Dissipated energy in one cycle
- \( X_0 \): Max. strain
- \( \sigma_{\text{max}} \): Max. stress
- \( Q \): Quality factor
- \( \lambda \): Logarithmic decrement
- \( \Delta x_{\text{dB},t} \): Amplitude decay of \( x \) dB in time domain
- \( f_{\text{res}} \): Resonance frequency
- \( \Delta t \): Decay time
- \( \Delta x_{\text{dB},f} \): Amplitude decay of \( n \) dB in frequency domain
- \( \Delta \omega \): Frequency range of decay in radian
- \( \omega_{\text{res}} \): Resonance frequency in radian

As the most important damping measure, loss factor is defined as the ratio between the dissipated and reversible energy in a system. The value reveals how high the damping is. Loss factor should be distinguished according to the indicated object: e.g. material loss factor \( \eta_m \) of a single material, structural
2.2 Theoretics of vibration and damping

In many cases, we analyze a multiple-degree-of-freedom (MDOF) system with a single-degree-of-freedom (SDOF) system of lumped-parameters, because it’s easy to understand and quick to handle. In this theoretical section, equations are given based on the SDOF system, which is actually also used for the evaluation through the whole thesis.

In the real world, mechanisms causing damping and energy losses are complex. Nonlinear behavior exists in most structures. However, if the damping is light, the energy loss in each vibration cycle is far more important than the way in which the damping is exactly induced [34]. For the simplicity of the mathematical implementation, the nonlinear behaviors of a system can be interpreted by equivalent linear vibration models that represent substantial characteristics of complex structures [35, 36]. Two principal linear models, which are systematically quantified, are the viscous and hysteretic damping model [8, 37]. The viscous damping model takes the assumption that the damping is frequency dependent. It is commonly used to define the damping coming from external forces. The hysteretic damping model assumes that damping is frequency independent. It is applied to interpret the intrinsic damping of materials or structures [38].

2.2.1 Viscous damping

The viscous damping assumes that the magnitude of the damping force is proportional to the velocity of the system. The model is based partially on physical observation and partially on mathematical convenience. Physically a viscous damper can be understood by a fluid dashpot [39].

The equation of motion of a viscous damped SDOF system under forced vibration condition with a cosinusoidal or sinusoidal excitation in frequency \( \omega \) of the
magnitude $F_0$ is given by:

$$m\ddot{x}(t) + c\dot{x}(t) + kx(t) = F(t) = F_0 \cos \omega t$$  \hspace{1cm} (2.2)$$

where $m$ is the mass of the system, $c$ is the viscous damping coefficient and $k$ the stiffness coefficient of the system.

By defining the natural frequency $\omega_n$ and damping ratio $\zeta$ with:

$$\omega_n = \sqrt{\frac{k}{m}}$$  \hspace{1cm} (2.3)$$
$$\zeta = \frac{c}{2\sqrt{km}} = \frac{c}{2m\omega_n}$$  \hspace{1cm} (2.4)$$
equation 2.2 can be written in:

$$\ddot{x}(t) + 2\zeta\omega_n\dot{x}(t) + \omega_n^2 x(t) = \frac{F_0}{m} \cos \omega t$$  \hspace{1cm} (2.5)$$

The total response of the system is the sum of homogeneous natural response $x_h$ and the particular enforced response $x_p$:

$$x(t) = x_h(t) + x_p(t)$$  \hspace{1cm} (2.6)$$

The homogeneous solution of the transient oscillation is:

$$x_h(t) = e^{-\frac{c}{2m}t} \left( C_1 \sin(\omega_n \sqrt{1 - \zeta^2} t) + C_2 \cos(\omega_n \sqrt{1 - \zeta^2} t) \right)$$  \hspace{1cm} (2.7)$$

where $C_1$ and $C_2$ are constants depending on the initial conditions. And the particular solution of the steady state response can be expressed by:

$$x_p(t) = X_0 \cos(\omega t - \delta)$$  \hspace{1cm} (2.8)$$

where $X_0$ is the amplitude of the system under forced vibration and $\delta$ is the phase lag between the excitation $F(t)$ and the response $x_p(t)$. The solution of $X_0$ is:

$$X_0 = \frac{F_0}{\sqrt{(k - m\omega^2)^2 + c^2\omega^2}}$$  \hspace{1cm} (2.9)$$

Using the definition of $\omega_n$, $\zeta$ and denoting the frequency ratio $r = \frac{\omega}{\omega_n}$:

$$X_0 = \frac{F_0}{k\sqrt{(1 - r^2)^2 + (2\zeta r)^2}}$$  \hspace{1cm} (2.10)$$
the solution of $\delta$ is:

$$\delta = \arctan \frac{c\omega}{k - m\omega^2} \quad (2.11)$$

similarly, we get:

$$\delta = \arctan \frac{2\zeta r}{1 - r^2} \quad (2.12)$$

The frequency transfer function can be used to determine the damping ratio [8]. By determining a certain frequency range at the resonance, the relationship between the magnitude decay of the frequency transfer function and the investigated frequency span reveals the damping ratio. Altering the equation 2.5 with Laplace transformation:

$$G_{visc}(j\omega) = \left[ \frac{\omega_n^2}{s^2 + 2\zeta \omega_n s + \omega_n^2} \right]_{s=j\omega} = \frac{\omega_n^2}{\omega_n^2 - \omega^2 + 2\zeta \omega_n \omega j} \quad (2.13)$$

where $s$ is the Laplace variable. At natural frequency, the transfer function is $G_{visc}(j\omega)|_{\omega=\omega_n} = \frac{1}{2\zeta j}$. The magnitude at this frequency is:

$$|G_{visc}(j\omega)|_{\omega=\omega_n} = \frac{1}{2\zeta} \quad (2.14)$$

Now assuming the frequency range is determined with the two points at the magnitude decay of $\frac{1}{n}$ (the points with the $\frac{1}{n^2}$ power of the peak power value), together with equation 2.13:

$$\frac{1}{n} \cdot \frac{1}{2\zeta} = \left| \frac{\omega_n^2}{\omega_n^2 - \omega^2 + 2\zeta \omega_n \omega j} \right| = \left| \frac{1}{1 - \left( \frac{\omega}{\omega_n} \right)^2 + 2j\zeta \frac{\omega}{\omega_n}} \right| \quad (2.15)$$

The transformed equation is:

$$\left( \frac{\omega}{\omega_n} \right)^4 - 2(1 - 2\zeta^2) \left( \frac{\omega}{\omega_n} \right)^2 + (1 - 4n^2\zeta^2) = 0 \quad (2.16)$$

In order to get two positive roots $\omega_1, \omega_2$, the damping ratio should remain $\zeta < \frac{1}{\sqrt{2}}$. Defining $\omega_2 > \omega_1$:

$$\left( \frac{\omega_1 - \omega_2}{\omega_n} \right)^2 = 2(1 - 2\zeta^2) - 2\sqrt{1 - 4n^2\zeta^2} \quad (2.17)$$
the general expression of the relationship between frequency span and damping (damping ratio) is thus:

\[
\frac{\Delta \omega}{\omega_n} = \sqrt{2(1 - 2\zeta^2) - 2\sqrt{1 - 4n^2\zeta^2}} \tag{2.18}
\]

In the case of remarkable damping, with \( \omega_{res} = \omega_n \sqrt{1 - 2\zeta^2} \), it can be deduced the evaluation based on the actual resonance should be:

\[
\frac{\Delta \omega}{\omega_{res}} = \frac{\sqrt{2(1 - 2\zeta^2) - 2\sqrt{1 - 4n^2\zeta^2}}}{\sqrt{1 - 2\zeta^2}} \tag{2.19}
\]

### 2.2.2 hysteretic damping

Hysteretic damping postulates that the damping force is in phase with the velocity and have a proportional magnitude to the displacement [30]. It is originally used to describe the internal damping properties of solid materials [40, 41]. The system stiffness coefficient can be described with a complex constant \( k(1 + j\eta) \), where \( j^2 = -1 \).

Hysteretic damping can be modeled with the following equation:

\[
m\ddot{x}(t) + k(1 + j\eta)x(t) = F_0 \cos \omega t \tag{2.20}
\]

or with:

\[
\ddot{x}(t) + \omega_n^2(1 + j\eta)x(t) = \frac{F_0}{m} \cos \omega t \tag{2.21}
\]

similar to equation 2.9 of viscous damping, it can be deduced for the hysteretic damping:

\[
X_0 = \frac{F_0}{\sqrt{(k - m\omega^2)^2 + k^2\eta^2}} \tag{2.22}
\]

or

\[
X_0 = \frac{F_0}{k \sqrt{(1 - r^2)^2 + \eta^2}} \tag{2.23}
\]

the phase:

\[
\delta = \arctan \frac{\eta}{1 - r^2} \tag{2.24}
\]
From the equation 2.21, the transfer function of hysteretic damping is:

\[ G_{\text{hyst}}(j\omega) = \left[ \frac{1}{ms^2 + k(1 + j\eta)} \right]_{s=j\omega} = \frac{1}{-m\omega^2 + k(1 + j\eta)} \]  

(2.25)

similarly, there is \( G(j\omega)|_{\omega=\omega_n} = \frac{1}{k\eta} \) . And its magnitude is:

\[ |G_{\text{hyst}}(j\omega)|_{\omega=\omega_n} = \frac{1}{k\eta} \]  

(2.26)

at \( \frac{1}{n} \) amplitude decay:

\[ \frac{1}{n} \cdot \frac{1}{k\eta} = \left| \frac{1}{-m\omega^2 + k + k\eta j} \right| = \frac{1}{\sqrt{(k - m\omega^2)^2 + k^2\eta^2}} \]  

(2.27)

after transformation:

\[ \left( \frac{\omega}{\omega_n} \right)^4 - 2 \left( \frac{\omega}{\omega_n} \right)^2 + \eta^2(1 - n^2) + 1 = 0 \]  

(2.28)

The roots are:

\[ \left( \frac{\omega_{1,2}}{\omega_n} \right)^2 = 1 \pm \sqrt{\eta^2(n^2 - 1)} \]  

(2.29)

Thus, the relationship between frequency span and damping (loss factor) is:

\[ \frac{\Delta\omega}{\omega_n} = \sqrt{1 + \eta \sqrt{n^2 - 1}} - \sqrt{1 - \eta \sqrt{n^2 - 1}} \]  

(2.30)

2.2.3 comparison of theoretical models

Concerning the system behavior with a certain value of \( \frac{\Delta\omega}{\omega_n} \) (a single transfer function curve), the value of damping can be evaluated with viscous or hysteretic damping theory. Actually, even under hysteretic damping, a more general formulation of the loss factor is used instead of the tedious exact solution. Figure 2.1 presents the difference between these theories for a single transfer function. With the equation 2.19 and 2.30, figure 2.1 is plotted to compare the viscous and hysteretic model in different \( \text{dB} \) decay values. It shows that for light damped behavior until \( \frac{\Delta\omega}{\omega_n} < 0.1 \), viscous and hysteretic damping have barely any difference. For 3 \( \text{dB} \) decay in logarithmic display (\( n = \frac{1}{\sqrt{2}} \) amplitude decay):

\[ \eta = 2\zeta \]  

(2.31)
Chapter 2. Theories and Models of Vibration and Damping

Figure 2.1: evaluation of damping using $\frac{\Delta \omega}{\omega_{res}}$

It is a widely used equation due to its simplicity for determining damping.

For higher damping behavior (with bigger $\Delta \omega/\omega_{res}$ value), the deviation from viscous to hysteretic damping increases. That’s the reason why in most literatures it is emphasized the damping should remain small to get the relation of loss factor in equation 2.31.

Besides the exact hysteretic damping, normally a linear function (see the exp. curves in figure 2.1 and equation 2.38) between loss factor and $\frac{\Delta \omega}{\omega_n}$ is used in the praxis for determining the loss factor. This function used for the experiment results has slight difference to the exact hysteretic solution. However, due to its convenience for calculation damping, equation 2.38 is a simple practical choice rather than the other two theoretical types with tedious derivations.

2.2.4 frequency response function

In order to investigate the dynamic behavior of a system, transfer functions in frequency domain are normally used. In the vibration field, they are called as frequency response function (FRF) [42]. There are multifarious FRFs like listed in table 2.1.
2.2. Theoretics of vibration and damping

<table>
<thead>
<tr>
<th>Name</th>
<th>Definition</th>
<th>Symbol</th>
</tr>
</thead>
<tbody>
<tr>
<td>Receptance</td>
<td>output displacement / input force</td>
<td>$Y_D = \frac{x(t)}{F(t)}$</td>
</tr>
<tr>
<td>Mobility</td>
<td>output velocity / input force</td>
<td>$Y_V = \frac{\dot{x}(t)}{F(t)}$</td>
</tr>
<tr>
<td>Accelerance</td>
<td>output acceleration / input force</td>
<td>$Y_A = \frac{\ddot{x}(t)}{F(t)}$</td>
</tr>
<tr>
<td>Dynamic stiffness</td>
<td>output displacement / input force</td>
<td>$Z_D = \frac{F(t)}{x(t)}$</td>
</tr>
<tr>
<td>Impedance</td>
<td>output velocity / input force</td>
<td>$Z_V = \frac{F(t)}{\dot{x}(t)}$</td>
</tr>
<tr>
<td>Inertance</td>
<td>output acceleration / input force</td>
<td>$Z_A = \frac{F(t)}{\ddot{x}(t)}$</td>
</tr>
</tbody>
</table>

Table 2.1: various FRFs

With these definitions, their relationships can be deduced by:

\[
\omega^2 |Y_D| = \omega |Y_V| = |Y_A| \tag{2.32}
\]

\[
|Z_D| = \omega |Z_V| = \omega^2 |Z_A| \tag{2.33}
\]

In figure 2.2, the measured dynamic stiffness / inertance are plotted in solid lines, their with $\omega$ multiplied / divided curves in dash lines respectively. It can be observed that they can be converted to each other easily according to equation 2.32 and 2.33 with a constant offset value.

Since the evaluation take different output signals, the resonance frequencies of the FRFs can be different. Their values are listed in table 2.2.

<table>
<thead>
<tr>
<th>Type of resonance</th>
<th>without damping</th>
<th>with damping</th>
</tr>
</thead>
<tbody>
<tr>
<td>free vibration</td>
<td>$\omega_n$</td>
<td>$\omega_n \sqrt{1 - \zeta^2}$</td>
</tr>
<tr>
<td>enforced displacement resonance</td>
<td>$\omega_n$</td>
<td>$\omega_n \sqrt{1 - 2\zeta^2}$</td>
</tr>
<tr>
<td>enforced velocity resonance</td>
<td>$\omega_n$</td>
<td>$\omega_n$</td>
</tr>
<tr>
<td>enforced acceleration resonance</td>
<td>$\omega_n$</td>
<td>$\frac{\omega_n}{\sqrt{1-2\zeta^2}}$</td>
</tr>
</tbody>
</table>

Table 2.2: resonance frequencies $\omega_{res}$ of various FRFs

From the table it can be observed, if the damping of the system is light, the resonance frequencies are almost the same (see figure 2.2) and have only immaterial difference. Inversely, remarkable deviation of resonance frequency can appear in well damped structure like plotted in figure 2.3: E.g., the eigenfrequency of accelerance is higher than receptance. The question is now which FRF should be chosen as the evaluation criterion. For a defined system whose geometry and
properties are not modifiable, principally the structural failure is dominantly dependent on the vibration amplitude (the risk of big displacements), whereas the human comfort is more dependent on the acceleration of the structure. Therefore, the receptance should be taken concerning the mechanical aspects and the accelerance regarding the comfort. A balanced evaluation between these two situations is the usage of the mobility.

FRF are normally plotted in logarithmic values, for instance, the receptance is expressed as:

\[
FRF_{Y_{D,SDOF}} = 20 \cdot \log_{10} |Y_D| \tag{2.34}
\]

In a MDOF system, the vibration / damping cannot be reflected with a single point, because the local vibration circumstances at various positions are also different. The common way is to calculate the root mean square (RMS) transfer function out of several places and use this mean FRF curve to interpret the global vibration / damping behavior. The mean FRF of a MDOF system can be defined as:

\[
FRF_{Y_{D,MDOF}} = 20 \cdot \log_{10} \left( \frac{\sqrt{\sum_{i=1}^{p} |Y_{D,i}|^2}}{p} \right) \tag{2.35}
\]

where \( p \) refers to the total number of the measured points.
2.3 Measurements and estimation of damping

Damping value can be determined in the experiments with different configurations. Mainly they can be measured under two conditions: free or forced vibration. The measured signal can be either directly evaluated in the time domain or they can be first converted with fourier transformation into frequency domain and then be evaluated. Since the evaluation process in the frequency domain is identical for both free and enforced vibration conditions, actually there are three main configurations which are frequently used:

- configuration in frequency domain
- configuration in time domain under free vibration condition
- configuration in time domain under forced vibration condition

2.3.1 damping estimation in frequency domain

The analysis in frequency domain under the forced vibration condition can be realized by using an electromagnetic shaker introducing continuous excitation. The measurement under free vibration condition can be done by an impact hammer with a transient excitation [43]. For both cases, the FRF curve will be plotted in the same way. A typical resonance peak can be observed in figure 2.4.
According to equation 2.30, we get the exact solution of the loss factor with:

$$\eta = \frac{1}{\sqrt{n^2 - 1}} \cdot \frac{\Delta \omega}{\omega_{res}} \cdot \sqrt{1 - \frac{1}{4} \left( \frac{\Delta \omega}{\omega_{res}} \right)^2}$$  \hspace{1cm} (2.36)

Since the high order term $\frac{1}{4} \left( \frac{\Delta \omega}{\omega_{res}} \right)^2$ is negligible:

$$\eta = \frac{1}{\sqrt{n^2 - 1}} \cdot \frac{\Delta \omega}{\omega_{res}}$$  \hspace{1cm} (2.37)

By transferring $n$ in logarithmic value with $n = 10^{\frac{\Delta x_{dB,f}}{20}}$, the loss factor can be expressed with:

$$\eta = \frac{1}{\sqrt{10^{\frac{\Delta x_{dB,f}}{10}} - 1}} \cdot \frac{\Delta \omega}{\omega_{res}}$$  \hspace{1cm} (2.38)

which is the prevalent equation we use for calculating $\eta$. The evaluation of the damping with this equation is called as fractional power bandwidth method.
(FPBW). Normally in the experiments, the value of $\Delta x_{dB,f}$ is recommended to be chosen between 0.5 and 3 dB [29], if it is not further treated with curve-fitting procedure (see section 2.3.4).

Simplifying the equation further with $\Delta x_{dB,f} = 3$ dB:

$$\eta \approx \frac{\Delta \omega_{3dB}}{\omega_{res}}$$  \hspace{1cm} (2.39)

This is the equation we mostly see in the literature [36], where $\Delta \omega_{3dB}$ denotes the frequency span of 3 dB decay in frequency domain. It is generally called as half-power bandwidth method (HPBW).

### 2.3.2 damping estimation in time domain with free vibration

The free vibration condition in time domain is suitable for measuring the damping in simple structures, e.g., a beam, whereas the damping of the system should not be too high (typically $\eta < 0.01$) [44]. An initial displacement will be imposed on the structure. By releasing the initial displacement, the vibratory magnitude of the displacement is recorded as the function of time.

![Figure 2.5: oscillatory deflection as function of time with free vibration](image)

Figure 2.5: oscillatory deflection as function of time with free vibration
An important parameter - the logarithmic decrement is defined as:

\[ \lambda = \ln \frac{x_{i-1}}{x_i} = \frac{1}{i} \ln \frac{x_1}{x_i} \]  

(2.40)

where \( i \) is like in figure 2.5 illustrated the index of the deflection wave and \( x_i \) the wave amplitude.

The loss factor can be expressed with:

\[ \eta = \frac{\lambda}{\pi} = \frac{1}{\pi i} \cdot \ln \frac{x_1}{x_i} = \frac{20 \log \frac{x_1}{x_i}}{27.3i} = \frac{20 \log \frac{x_1}{x_i}}{27.3 \Delta t f_{res}} \]  

(2.41)

\( \Delta t \) and \( f_{res} \) refer to the decay time and the resonance frequency of the structure respectively.

Sometimes the decay plot is recorded in logarithmic values. By taking the decay \( 20 \log \frac{x_1}{x_i} = 60 \text{ dB} \), the equation 2.41 can be simplified to:

\[ \eta = \frac{2.2}{f T_R} \]  

(2.42)

\( T_R \) is the time of decaying 60 dB, which is also called the reverberation time [44].

Or in the case the amplitude is decayed by half with \( \frac{x_1}{x_i} = 2 \) (6 dB decay):

\[ \eta = \frac{0.22}{f T_{1/2}} \]  

(2.43)

\( T_{1/2} \) is the time of decaying 6 dB.

### 2.3.3 damping estimation in time domain with forced vibration

As it was formulated in section 2.2.1, the response of a SDOF system under harmonic excitation is also an oscillatory motion but with a delayed phase. This characteristic is used in the experiments to determine the loss factor. It is widely used on the dynamic-mechanical-analysis (DMA) device [45]. Particularly, this method is suited for measuring loss factors greater than 0.1 [46]. But it is to mention that this principle is only valid under low frequencies, i.e. if the phase lag is tested at a frequency which is already close to the base resonance of the system, the results can be erroneous (see section 3.2.3).

By recording the excitation force signal \( F(t) \) and the response displacement signal \( x(t) \) as functions of time in one plot, the loss factor can be measured in this situation with:

\[ \eta = \tan \delta \]  

(2.44)
2.3. Measurements and estimation of damping

Figure 2.6: phase lag between excitation and response

If the excitation and response signal in figure 2.6 are plotted by using the strain $x$ as the X-axis and stress $\sigma$ as the Y-axis, a hysteresis loop like in figure 2.7 can be illustrated from the system’s motion. The loss factor can be expressed in this plot with:

$$\eta = \frac{\Delta U_h}{2\pi U_S} = \frac{\Delta U_h}{2\pi \cdot \frac{1}{2} \sigma_{max} X_0^2} = \frac{\Delta U_h}{\pi \cdot X_0 \cdot \sigma_{max}}$$ (2.45)

$\Delta U_h$ is the dissipated energy in one cycle, also the enclosed area of the ellipse in figure 2.7. $U_S$ is the stored reversible energy in one cycle. $\sigma_{max}$ and $X_0$ are the maximal stress and strain respectively.

2.3.4 Curve fitting procedure

Since the theory of evaluating loss factor with FPBW method based on the definition of a SDOF system, the “correct” loss factor should be calculated with a SDOF-shape preserving resonance curve, which is normally not obeyed with the mean FRF curve acquired from equation 2.35. Therefore, additional measures are entailed to fit the realistic FRF curve into a SDOF system curve for the assessment of modal damping. A curve-fitting tool using least square algorithm can be utilized for this purpose. The evaluation of the loss factor
utilizes merely the magnitude of the FRF.

Referring to the equation 2.9 and table 2.1, the modulus of the receptance FRF $Y_D$ should obey:

\[
Y_D = 20 \cdot \log_{10} \left| \frac{X_0}{F_0} \right| = 20 \cdot \log_{10} \left| \frac{1}{k \sqrt{(1 - r^2)^2 + (2\zeta r)^2}} \right| \quad (2.46)
\]

together with:

\[
\frac{r}{\omega_{res}} = \omega
\quad (2.47)
\]

Where $\omega$ denotes the varying frequency here, $r$ is the frequency ratio between the varying and resonance frequency.

The RMS curve of a MDOF system in equation 2.35 should be fitted according to these two equations. The reason of fitting these parameters is: Normally the loss factors evaluated directly from the original FRF through the FPBW method vary with the chosen range of the resonance peak since it’s not a SDOF system curve. Through the fitting process, the modal damping $\zeta$ is addressed with a unique value. The evaluation of the loss factor becomes independent to the selected frequency range (wider frequency range choice for the evaluation),
which facilitates the damping evaluation with an accurate prediction. Also some flaws in the experimental curve like distorted curve shape at some places can be fixed with the modal fitting algorithm easily (see section C).

By using the least square optimization algorithm of Levenberg-Marquadt [47, 48], the modal parameter \( \omega_{\text{res}}, k \) and \( \zeta \) will be determined in the equations 2.46 and 2.47. The fitted curve with the addressed parameters \( \omega_{\text{res}}, k \) and \( \zeta \) will be further interpolated as shown in figure 2.8. The accurate resonance frequency, amplitude and loss factor will be evaluated with this interpolated curve. Actually the loss factor and the resonance frequency can already be acquired with the fitted parameter in equation 2.46 without data interpolation. Their values don’t change noticeably after the interpolation by using FPBW method. Only the exact amplitude of the resonance must be acquired through the interpolation.

Similarly, the modulus of mobility and accelerance can be expressed respectively by:
\[ |Y_V| = 20 \cdot \log_{10} \left| \frac{\omega X_0}{F_0} \right| = 20 \cdot \log_{10} \left| \frac{\omega}{k \sqrt{(1 - r^2)^2 + (2 \zeta r)^2}} \right| \] (2.48)

\[ |Y_A| = 20 \cdot \log_{10} \left| \frac{\omega^2 X_0}{F_0} \right| = 20 \cdot \log_{10} \left| \frac{\omega^2}{k \sqrt{(1 - r^2)^2 + (2 \zeta r)^2}} \right| \] (2.49)

## 2.4 Damping concepts

An overview of the physical principals of damping is presented in figure 2.9. Various damping approaches based on these principles have been documented in varieties of scientific contributions. In this section, the damping approaches are described, which are investigated in this thesis.

Viscoelastic Damping: viscoelastic materials (VEM), especially rubbers have the highest loss factor among all damping materials. The value reaches typically up to about 1 or even higher [8]. A popular damping materials for automotive
2.4. Damping concepts

and civil engineering is the butyl rubber. Bulky metal-rubber damping and isolation components are often used in vehicles at various mounts to reduce the structure-borne noise, e.g. the decoupled pulley in the drive strain, engine mount and torque support in the power train, blade bush, airbag absorber and crankshaft mount in the chassis [49, 3]. The properties of the VEM vary strongly with the temperature-frequency boundaries, which should be specified and fully considered for the design of a viscoelastic device. Viscoelastic damping can be applied additionally with several other damping approaches, e.g., it can be combined with piezoelectric and magnetic damping. The classical method of damping plate structures is to layer VEM on the surface of the plate. According to the deformation state of the viscoelastic layer they can be divided into free layer damping (FLD) and constrained layer damping (CLD) [31]. In FLD one single viscoelastic layer is coated, in which extensional / compressional stresses exist during the bending vibration of the plate structure. In CLD one additional thin stiff layer (typically a metal sheet) is capped on the top of the viscoelastic layer so that the bending vibration of the plate structure induces dominantly shear stress in the viscoelastic layer. The shear deformation of the damping material in CLD configuration raises the efficiency of the treatment considerably compared to the FLD. Commercial damping sheets with a single VEM layer or additional also with an aluminum constraining layer are available for the FLD and CLD treatments respectively. Rubbers are also sprayable to the panel surfaces by robots [16]. Oberst [50] analyzed the behavior of a FLD beam. Based on his work, Ross [21] et al. completed and generalized the analytic model to CLD. The performance enhancement of CLD was proposed in [24] by segmenting the constraining layer into several sections. As a widespread standard damping technique, viscoelastic damping is one of the key approaches to be investigated in this thesis. The investigated interface damping (ID) technique is also a variation of viscoelastic damping.

Piezoelectric Damping: A variety of damping approaches use piezoelectric elements. The physical properties of piezoelectric transducers and their application aiming at vibration control are described by Moheimanie in [51]. Shunt damping treatments [52, 53, 54, 55, 56] employ the mechanical and electrical coupling effect of the piezo elements to dissipate energy with a shunt connected to them. Niederberger summarized different shunt technologies including classical resonant shunts, switching shunts and shunts applied on active fiber composites [57]. Besides the shunted piezo treatments, the piezoelectric elements can be also used in combination with CLD treatment. A typical configuration is called active constrained layer damping (ACLD). There, a piezo-sensor is bonded directly on the substrate and the collocated constraining layer is built by a piezo-actuator patch. According to the bending state of the structure, the piezo-actuator patch makes a deformation, that is negatively proportional to the strain generated in
the sensor patch, so that the shear effect in the viscoelastic layer is increased. Trindade and Stanway reviewed diverse contributions of ACLD and their configurations / results [58, 59]. Since plates are normally partially treated with ACLD, the positioning of the ACLD patch can be optimized with the modal strain energy approach [60]. Like the shunt damping technologies, different control algorithms of the piezo-actuator are influential to the performances of ACLD. Besides the above mentioned proportional control, Gandhi proposed a velocity control of the actuator. The transmitted active force afforded by the piezo-actuator damps the substrate dominantly with the derivative feedback control [61]. Because ACLD combines the merits of both piezoelectric and viscoelastic damping, it became another research emphasis of this thesis with the results from the screening phase.

**Magnetic Damping:** By applying magnetic damping, electromagnetic forces are generated between the magnetic elements on the vibrating structure and magnetic elements from the surrounding structures. These forces can attenuate the vibration. An important feature of magnetic damping is that the damping force is dependent on the vibration amplitude (indirectly the distance between magnetic damping elements). With the increased structure displacement, the damping force become significant in a semi-active way. Gospodaric [62] investigated the vibrations on a cantilever beam, which was actively damped by two lateral electromagnets. Moreover, in combination with magnetic elements, the performance of viscoelastic damping mechanisms may be augmented. Ebrahim and Baz [63] introduced magnetic constrained layer damping (MCLD), in which the segments of the viscoelastic layers are constrained with thin magnets instead of metal sheets. The deformations of the viscoelastic cores are increased with the magnetic interactions between the segments. A finite element model (FEM) model of MCLD was developed by Ruzzene in [64]. It is even possible to operate the MCLD actively with electromagnets (EMCLD) as was proposed by Niu et al. in [65]. The comparison of the passive and active MCLD based on a clamped-clamped beam was performed by Zheng et al. [66].

**Impact (Particle) Damper:** A primitive impact damper is an auxiliary mass that is to be employed for transferring momentum between the primary structure and this mass itself [67, 68]. Based on this principle, particle dampers are developed with better performances due to the non-elastic collisions and friction between particles and also due to the interaction of the particles to their containers. The particle dampers (PD) are commonly divided into three types: simple particle dampers, bean bag dampers (BBD) and non-obstructive particle dampers (NOPD). The bean bag damper introduced by Popplewell [69] used an additional plastic bag wrapping the particles in comparison to simple particle dampers. It was found that the bean bag operates quietly thanks to the resilience of the bag and has a better performance than a single rigid impact slug.
by elongating the duration of the impulse exchange. The non-obstructive particle dampers introduced by Panossian [70] is applied by means of drilling some small holes at appropriate locations inside the vibrating structure and filling them with particles. Compared to other treatments, the modeling and simulation of a whole mechanical system with PD is difficult. Normal FEM handling continuum simulations are not suitable to portray the behaviors of discontinuous microparticles. The discrete element method (DEM) is a relatively new approach to analyze the dynamic behavior of particles [71]. Partial combination of FEM and DEM was applied in [72]. However, due to the computing efficiency, it is not yet a tractable tool to confront the vibration problems. Owing to the deficient simulation methodologies, PD is mainly investigated in experiments in this thesis. The structure of PD is relative simple. The properties of typical particles (metal or sand) change barely with the environment variation like temperature. That implies, PD could have a steady efficiency under different boundary conditions. These features make PD an attractive treatment for overcoming some shortcomings of viscoelastic dampers. Therefore, PD is the third treatment that is underlined in this contribution.

Coulomb/electrostatic Damper: A Coulomb damping technique, which is investigated in this thesis instead of common Coulomb friction dampers, has been originally proposed by Bergamini [73] based on electrostatic stress in a multilayer structure. In this approach a thin dielectric sandwich core is covered with two conductive skins as electrodes. Through the electric potential existing on the two surfaces of this dielectric layer, an electrostatic stress can be generated across the thickness direction, which leads to the normal stress. By adjusting the voltage on the electrodes, the normal stress can be varied. As a consequence, the maximal shear stress due to friction can also be varied. If the interlayer shear stress exceeds the maximum, which holds the layers connected, then sliding and frictions occur between the core and skins, contributing energy dissipation. In his research, a multilayer cantilever beam, which is over two meters long, was damped with a loss factor of 0.16. Owing to the novelty of this damping device, limited literature can be found. By tailoring the damping, this method modifies the bending stiffness of the structure significantly [74]. A high voltage up to several hundreds or even over one thousand volts is needed to attain a decent damping value. The break-down of the dielectric material can be a killing factor for this method, because the structure will be then hard to repair.

Eddy Current Damper: The eddy current damper (ECD) is a similar damper to the magnetic dampers. It utilizes eddy current to generate an electromagnetic damping force, which is induced when a conductor moves through a constant magnetic field or when the conductor stays in a time-varying magnetic field. This
induced force is proportional opposite to the motion velocity of the component, which reveals the induction force is actually equivalent to a viscoelastic damping force. This physical phenomenon has been widely applied on the circular eddy current brake, whereas the application on panels as a damper hasn’t been deeply investigated. Sodano [75, 76] et al. have applied conducting copper sheets and permanent magnets on a cantilever beam to suppress vibrations. A deep insight of ECD’s behavior is provided by [77]. Ebrahim [78, 79] et al. proposed an ECD damper for the automotive suspension system and conceived the optimum arrangement of the permanent magnets to augment the magnetic flux density. An advantage of ECD is that the damping force is generated under non-contacting condition like magnetic damper. Thus, most mechanical properties of the original system are not modified. In contrast to VEM damper, the performance of ECD doesn’t degrade over time. ECD can be a suitable approach for rigid systems, e.g., for the high-speed elevator cars, whose weight is not a critical aspect.

**MR fluid Damper:** Magnetorheological fluid (MR) is a smart fluid, whose rheological behavior can be controlled through the application of a variable external magnetic field. It comprises magnetically polarizable particles, which are dispersed into a viscous fluid like silicone oil. The MR fluid can be used in a wide temperature range [80]. The commercial MR dampers are successfully applied on the stay-cable bridges and vehicle suspensions [81]. Usually they are applied at low frequencies. The MR damper can be designed with rotational and translational operation mode. A thorough portrayal of commercial MR dampers is reported in [82]. By varying currents in the coil of the damper, the friction forces generated in the orifices in the damper are adjustable. In consequence, the damping performance can be tailored. A basic analytic model of MR dampers was developed by Bouc-Wen [83]. Based on this model, Spencer et al. [84] proposed a new phenomenological model overcoming some shortcomings of several reviewed models. A comparative study of various control algorithms on the MR is presented in [85].

### 2.5 Behavior and modeling of viscoelastic materials

As important damping materials, the behaviors of viscoelastic materials (polymers) are nonlinear. The phenomenon of their environmental dependence, the analytical model tackling their nonlinearity and the methodology for quantifying the storage modulus and loss factor of the VEM is introduced in this section.
2.5. Behavior and modeling of viscoelastic materials

2.5.1 Behavior dependence of viscoelastic materials

The properties (mainly storage modulus and loss factor) of viscoelastic material are highly dependent on temperature and frequency [31]. Typical curves of their dependence are depicted in figure 2.10 and 2.11.

![Figure 2.10: temperature dependence of the storage modulus and loss factor in VEM [36]](image)

In figure 2.10 it can be seen, at a constant frequency, the storage modulus decreases with the increasing temperature from glassy to rubberlike region, whilst typically the loss factor has its maximum peak value in the transition region, where the storage modulus drastically declines.

In figure 2.11, under the same temperature and with the increasing frequency, the storage modulus increases continuously. The loss factor has its maximum at a certain frequency.

These two varying tendencies of VEM’s property can be summarized to a single dependence relationship. With a shift function, the temperature dependence can be converted and expressed as the variation of the frequency (so-called reduced frequency). The complete model of the storage modulus and loss factor can be then expressed by an unique plot - the nomogram [86] as illustrated in figure 2.12. Discrete data points in the nomogram can be displayed in the Wicket plot [31] in order to check the data quality. Normally if the acquired data in the
experiments have a good quality, they form an unique inverted $U$ curve in the Wicket plot.

2.5.2 Fractional derivative model

The mathematical description of the curves in the nomogram can be expressed by a concise model: the fractional derivative model [87].
In this model, the storage modulus and loss factor are expressed by the complex modulus $E^*$, whose real part $\text{Re}(E^*)$ represents the storage modulus and imaginary part $\text{Im}(E^*)$ represents the loss modulus. The loss factor is equal to $\text{Im}(E^*)/\text{Re}(E^*)$. A basic 1$^{st}$ order model can be described with the complex modulus $E^*$ of the VEM in:

$$E^* = \frac{a_1 + b_1 \cdot (i\omega\alpha(T))^\beta}{1 + c_1 \cdot (i\omega\alpha(T))^\beta}$$  \hspace{1cm} (2.50)$$

In the equation, $a_1$, $b_1$ and $c_1$ can be complex, whereas $\beta$ is a real number. $\omega$ is the angular frequency and $\alpha(T)$ refers to the equation according to the Arrhenius shift relationship:

$$\log_{10}[\alpha(T)] = T_A \left( \frac{1}{T} - \frac{1}{T_0} \right)$$  \hspace{1cm} (2.51)$$

$T$ is the current temperature and $T_0$ is the arbitrarily selected temperature reference. They are both in degrees absolute (Kelvin). $T_A$ represents the slope for the shift factor.

If a VEM has more than one transition region, a higher order fractional model can be used [31]. For example, the complex modulus of the 2$^{nd}$ model can be described then with:

$$E^* = \frac{a_1 + b_1 \cdot (i\omega\alpha(T))^\beta + b_2 \cdot (i\omega\alpha(T))^{2\beta}}{1 + c_1 \cdot (i\omega\alpha(T))^\beta + c_2 \cdot (i\omega\alpha(T))^{2\beta}}$$  \hspace{1cm} (2.52)$$

Also here $b_2$ and $c_2$ can be complex.

### 2.5.3 Quantifying of storage modulus and loss factor

The properties of a real VEM can be obtained in two ways. A stiff VEM can be directly measured by the DMA device, whilst a very soft VEM can only be determined indirectly with a laminated composite beam.

One frequently used approach for acquiring the material properties of soft VEM is the RKU-method [21]. The basic solution was published by Oberst in [50], in which an analytical model of a two-layer FLD beam was developed. Based on this model, Ross, Kerwin and Ungar generalized the model to a three layer CLD beam with the additional constrained layer. Both models were originally
developed to assess the structural damping with known material parameters. They are now often applied inversely. With the measured structural damping of the composite beam, in which the VEM is laminated, the material loss factor and storage modulus of the VEM can be derived. Equations of RKU-method based on different FLD and CLD cantilever beams can be found also in the ASTM standard E756 [29].

Since the RKU-method takes some assumptions in the derivation and simplifies several boundary conditions, the derived material properties from this approach could be sometimes spurious or unrealistic. In consequence, in this thesis another approach developed in the master thesis of Studer [88] by means of curve fitting in FE-analysis was applied to attain the realistic material properties of VEM.

In principle, this approach determines the material properties in the same manner as the inversed RKU-method. Experiment results of the composite beam must be available for the curve-fitting. With the known geometry of the beam, the inherent relationship between the material and composite loss factors (also the relationship between the material storage modulus and the composite eigenfrequencies) are established by abstractive functions with the FE-simulation results. The material values can be than extracted from the functions and experiment data inversely. Although these processes appear to be tedious, they offer satisfactory results. This approach comprises following five steps:

- **acquisition of experimental results**: some constrained layer (CLD) beams using butyl are measured in Oberst beam tests [29] under different temperatures. The experimental system eigenfrequencies $\omega_{sys,exp}$ and loss factors $\eta_{sys,exp}$ are recorded and calculated respectively.

- **acquisition of simulation results and functions**: FE-analysis of the beams are executed based on the geometry and boundary conditions of the experiment configuration in the first step. The properties of the sub-strate and constraining layer are known whereas only the VEM properties remained unknown. By inputing arbitrary VEM Young’s modulus $E'_{mat,sim}$ and loss factor $\eta_{mat,sim}$, the structural eigenfrequency $\omega_{sys,sim}$ and system loss factor $\eta_{sys,sim}$ are recorded. Polynomial functions with unknown coefficients are built for the description of the FEM model with $\omega_{sys,sim} = f_1(E'_{mat,sim}, \eta_{mat,sim})$ and $\eta_{sys,sim} = f_2(E'_{mat,sim}, \eta_{mat,sim})$. All coefficients of the polynomials in the function $f_1$ and $f_2$ are fitted with a variety of the inputs $E'_{mat,sim}$, $\eta_{mat,sim}$ and outputs $\omega_{sys,sim}$, $\eta_{sys,sim}$. Thus $f_1$ and $f_2$ can be fully defined.

- **application of experimental results in the simulation functions**: the material values $E'_{mat,exp}$ and $\eta_{mat,exp}$ are fitted from the defined func-
tion $f_1$ and $f_2$ with the input of experimental system eigenfrequency $\omega_{sys,exp}$ and loss factor $\eta_{sys,exp}$ from the first step. The fitted $E'_{mat,exp}$ and $\eta_{mat,exp}$ served as discrete points of the ”realistic” material properties out of the experiments.

- **curve-fitting procedure for the VEM modeling:** a fractional derivative model is applied to describe the material property of VEM. The parameters of the fractional model are fitted with the derived values $E'_{mat,exp}$ and $\eta_{mat,exp}$ from the previous step. Until this step the complete material model is accomplished.

- **attaining a specific model from a generalized model:** the experiment temperature is input into the material model. The final curve of the storage modulus and loss factors of the VEM at the assigned temperature is finally generated.
Chapter 3

Experimental

In this chapter, two experimental setups for the damping measurements are introduced. Also the tested specimens on each setup are described. The results in chapter 4 can be referred to the specimen arrangements in section 3.2.2. The experimental results in chapter 5, 6 and 7 can be referred to the specimen configurations in section 3.3.2. The test conditions of each setup are enunciated. Additionally, some important aspects for the damper arrangements on plates are explained.

3.1 Introduction

In the first two project phases, the damping approaches were investigated mainly in experiments. Two test rigs have been developed for the respective phases. The biggest difference of the two test rigs is that the first one aims at measuring beam samples, while the second one is designed for testing plate specimens. These two configuration are complementary at many aspects like measurable frequency range and testing conditions. A number of equipments are used for both test rigs.

For the topic of ride comfort in automotive engineering, substantial mechanical vibration and structure-borne noise exist at low frequencies. Up to about 400 Hz, over ca. 95% of the vibrations can be perceived by human [89, 90, 3]. Above this frequency limit, there is only rest marginal perceptable portion due to airborne noise, which can be effectively damped or isolated in general. Thus, the critical frequency range up to 400 Hz was determined as the frequency range of investigation for all experiments.
3.2 Test rig for beams

The first test rig is designed according to the configuration of a DMA device. It eases the comparison of numerous damping treatments on a beam specimen and was thus applied for the screening study. As introduced in the theory part, this measurement equipment can be utilized only at low non-resonant frequencies. The signals are processed and evaluated in time domain.

3.2.1 Experimental setup

The damping behavior of a beam specimen can be determined under different loading conditions, e.g. single or dual cantilever, tension / compression, 3 or 4 point bending. In order to be able to investigate the influence of the interface damping, a three point bending setup with interchangeable support elements was chosen, which allow flexible testing of different specimen configurations [91].

![Figure 3.1: setup for beam specimens: a) test rig configuration; b) tested samples with the experiment bench](image)

Specimens are excited in the middle using a 9 N electromagnetic shaker (1). The applied force is measured by two MMF KF24 piezo transducers (2) at the support. The displacement response in the middle of the specimen is acquired by a laser displacement meter LC-2400A (3). For this purpose, a power amplifier KEPCO 20-5M (4) is connected to the shaker and a charge amplifier Kistler 5017A (5) is linked to the force transducers. The shaker is fixed to an outer frame. In order to avoid the vibration interference between the shaking and measuring system, the inner frame holding the sample, is installed separately from the outer frame. The force sensors are embedded under the support elements in the inner frame. The frames have been designed to ensure their lowest eigenfrequencies are beyond the testing frequency range of interest (483 and 527 Hz for the inner and outer frame respectively). Collocated to the excitation, the laser, which is also installed separately away from the outer and inner frames,
is placed on the opposite side under the specimen. The NI PXI 4462 (6) and 6229 (7) systems were used for signal generation and acquisition. For the post processing in Labview, a dynamic low pass filter is integrated into the program. The value of this high order filter is set only several Hertz (typically in the range of 10 Hz) upon the operating frequency so that the excitation and response of the sample can be exactly represented.

3.2.2 Beam sample arrangements

The measured specimens were fabricated in the dimension of 200 × 40 × 1 mm in order to keep the specimens remain beam-like and reduce the effect of torsion modes. If necessary, the specimen size could be extended to an area of 400 × 400 mm so that it is also possible to measure plate behavior under very low frequencies. The dimension of the specimen is larger than on common DMA devices, so that different damping mechanisms can be easily applied and compared.

Three variants of specimen as illustrated in figure 3.3 were investigated. At surface treatments, the damping materials are bonded in the center or through almost the whole length of the specimen to the supporting points. At marginal damping, the damping devices are positioned in the vicinity of the interface joint, which is in this configuration the two triangular supports for the beam. In some arrangements, auxiliary elements like magnets are fixed externally. Instead of supporting the specimen with two triangular parts, in case of interface
Figure 3.3: specimen variants of beams

treatments, direct linkage of the damping mechanisms at the interface joint is applied. For all three variants, the specimen is excited and measured at the center point.

3.2.3 Test conditions

The whole test rig is assembled on a massive steel test bench to avoid additional peripheral vibration problems. All samples were measured under constant room temperature of 24 °C.

The force and displacement signals are recorded as a function of time. The phase lag is directly read out from this signal-time diagram. The accurate positioning of the shaker tip is a precondition for the precise measurement. Before each experiment, an undamped specimen is used for adjusting the shaker tip position. The shaker is positioned using the output voltage offset function from the power amplifier KEPCO 20-5M, which imposes the offset of the shaker position in the vertical direction. In the signal-time diagram, the two signals should be adjusted to an overlapped position. A strain-stress diagram is depicted to check the positioning. In the case of an undamped beam, a single line indicating elastic behavior can be observed in this strain-stress diagram for the ideal positioning. If the shaker tip is adjusted too high, the specimen is not constantly in contact with the movable part of the shaker. This results in instability of the signal. Conversely, if the shaker is positioned too low, the
specimen is preloaded with a great bending stress. That induces much friction on the contact edges between the specimen and the support system. The energy dissipation through this friction causes also a phase lag in the system, which leads to spurious measurement results. Apart from these aspects, the shaker should be positioned vertically in the middle of the sample in order to avoid torsion modes.

The frequency scanning for each sample begins at 10 $Hz$ with the interval of 10 $Hz$. As mentioned before, the samples should only be measured in non-resonant condition. Below the first eigenfrequency, no system phase shift due to resonance will be yielded and the results are reliable. Above the first eigenfrequency, the system phase angle will be gradually influenced by the higher modes. The results are still applicable, however with continuously decreasing precision. Before the second eigenfrequency is reached, the accumulated system phase shift becomes significant and the measured damping angle will be erroneous. Due to different sample stiffness of the investigated configurations, their first eigenfrequencies are different. That means, the measurable ranges of the specimens are different. E.g., the specimen with electrostatic damper with low stiffness can be measured until 200 $Hz$ only, while the damped honey comb sandwiches with particle damper having high stiffness can be measured up to 400 $Hz$. Normally one can recognize the region near the resonance through the instable or highly deformed strain-stress ellipse. At the same time, a decrease of the damping phase lag can be apparently observed because of the drastic phase shift in region of resonance. Within the measurable range, i.e., from 10 $Hz$ to several hundred $Hz$ (according to the stiffness of the sample, typically from 110-400 $Hz$), average values were calculated for the loss factors $\eta_c$ of each specimen.

3.3 Test rig for plates

The second test rig is used for the investigation in the second project phase. It is designed for the measurements on plate at high resonant frequencies. The specimens are tested with a sine-sweep signal, which is a traditional method to predict the modal parameters of a structure. In our investigation, we implemented the modal testing in a freely supported condition (free-free), because it is notoriously difficult to establish other support conditions appropriately for complex structures in the practice [42]. The signals are analyzed in frequency domain and evaluated with FRF.
3.3.1 Experimental setup

The experimental setup chain is illustrated in figure 3.4. The specimen (1) is hung at both upper corners with two elastic wires which are fixed to two rigid points. The whole experiment is monitored with a Labview program (2). This Labview program outputs the excitation through a signal generation card (3) with a sine sweep signal in $10-400 \, Hz$ with the interval of $1 \, Hz$. This signal is amplified by a power amplifier (4) and fed further to the shaker (5) which excites the specimen from a foundation. On the tip of the connection rod between the shaker and the specimen, an impedance head (6) is installed to detect the excitation force. The induced charge signal is converted to a voltage signal with the charge amplifier (7). The displacement is recorded by a laser vibrometer (8) that fastened on another foundation separated from the shaker and the specimen, in order to make sure that the acquired signal is not disturbed by the oscillation from the shaker (details seeing figure 3.5). Both force and displacement signals are filtered with a low pass (9) and then recorded by the signal acquisition card (10) which transmits the data to the Labview program. Independent from the main measuring chain, another amplification loop is needed for the ACLD treatment by using two amplification stages (see section 6).

3.3.2 Plate sample arrangements

In the second project phase, many specimens with partially treated surface dampers (PD and ACLD) and interface dampers (ID) were measured. An undamped plate was measured at first as the reference to estimate the performances of the dampers.

The undamped specimen is made of aluminum and consists of two components: The frame and the plate. The frame is a sandwich structure with an aluminum honeycomb core and two aluminum facings. The frame has a rectangular hole in the middle. The plate is glued to the surface on one facing. Both parts are centrically collocated as indicated in figure 3.6. The dimensions can be seen in table 3.1. Further parameters of the applied dampers will be presented in chapter 5, 6 and 7 with their particular configurations.

The vibration and also the structure borne noise of panels in automotive are mostly induced by the vibration from its peripheral structures (mainly from the chassis in which the panels are mounted). The aim of using the sandwich frame is to yield a realistic environment for this kind of vibration transmission. Thus, the sample is excited on the frame that represents the periphery like chassis and the response is measured on the plate. For the ACLD and PD samples, the plate is glued by a thin epoxy layer to the frame (rigid interface), while instead VEM
Figure 3.4: measurement chain of plate specimens
Figure 3.5: setup for plate specimens: a) shaker ; b) laser

is used for the ID sample at the interface of plate and frame (compliant interface).

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>outer dimension of the frame</td>
<td>600 × 480 mm</td>
</tr>
<tr>
<td>inner dimension of the frame</td>
<td>400 × 300 mm</td>
</tr>
<tr>
<td>thickness of honeycomb core</td>
<td>40 mm</td>
</tr>
<tr>
<td>honeycomb cell size</td>
<td>ϕ6</td>
</tr>
<tr>
<td>honeycomb density</td>
<td>54 – 80 kg/m³</td>
</tr>
<tr>
<td>face thickness of the frame</td>
<td>2 mm</td>
</tr>
<tr>
<td>panel dimension</td>
<td>440 × 340 × 1 mm</td>
</tr>
</tbody>
</table>

Table 3.1: geometry of the plate sample

Furthermore, we would like to transmit most of the vibration into the plate and have little influence from the frame. For determining the mode patterns of the whole specimen, a modal analysis was done in Ansys under free-free condition as in the experiments. According to the results, there are totally 10 modes existing up to 400 Hz as indicated in figure 3.7 (only the nodes on the plate were extracted in the figure). Then a harmonic analysis with the same boundary conditions was performed discretely at these 10 eigenfrequencies to acquire the exact amplitudes of these modes. As shown in figure 3.7, the immense stiffness difference between the sandwich frame and the monolithic plate guarantees that the frame possesses only one eigenfrequency (mode 6)
3.3. Test rig for plates

Figure 3.6: plate sample configurations
up to 400 $Hz$, whilst the plate has already other 9 modes in this frequency range.

![Figure 3.7: eigenfrequencies of the plate in the simulation](image)

### 3.3.3 Optimization of damper size and comparability

Because some plate specimens are partially treated with surface dampers, the optimization of the damper size and location becomes another essential issue for the plate experiments. In addition, the position of the dampers should be arranged in the manner that the performances of the surface dampers can be compared reasonably.

In order to compare the damping approaches ACLD and PD under similar conditions, the size of the sectional dampers are chosen with similar values: The applied commercial piezo-patch has a sectional size of $61 \times 35\ mm$. It results in that the size of ACLD damper has also the same sectional area. Therefore, PD’s area is chosen with a similar value of $60 \times 30\ mm$. The typical edge width of the bonded area on a vehicle panel is about $20\ mm$. The area of ID is thus chosen
Next aspect for the comparison of the different approaches is to consider the position of the dampers. For ID, no sophisticated positioning is required. The damper will be simply applied homogeneously at the whole interface edges. For the surface treatments, panels are typically treated with local surface dampers which do not cover its whole surface. The localization of the surface dampers is an important issue for their performances. In order to find the optimal damper location, the vibration behavior of an undamped plate was at first investigated. The amplitudes of all modes up to $400 \, Hz$ were accumulated into one plot. From this plot it can be seen on which position the plate has the highest amplitude through the whole frequency range. In order to get the comprehensive behavior of the plate, both receptance and mobility were superposed. Figure 3.8 shows the overlapped receptance and mobility. The entirely accumulated amplitudes from mode 1 to 10 on the left side distributes in the same pattern as the first mode, which reveals that the first mode is apparently dominant among all the modes. In order to take other modes also into account, the amplitudes were additionally accumulated apart from the first mode on the right side of the diagram. We can observe that still the middle region of the plate has the highest mobility. The receptance reaches also 80% of the highest value at this place. Thus, the midpoint is the best position for PD, because the damping effectiveness of PD is based on the vibration amplitudes. Since ACLD should be arranged by the modal strain energy approach [60] instead of the vibration amplitude, the strain energy on the plate of the modes up to $400 \, Hz$ was also accumulated like the process mentioned above. In figure 3.9 it can be seen, the maximal strain energy of all modes is actually concentrated in a narrow region at the longitudinal edges of the plate, whereas the midpoint has only about 70% of the maximal strain energy. Apart from mode 1, the midpoint becomes however the most dominant position. Considering the dimension of the piezo patch, dominance at all other modes and the comparison with PD, the midpoint is a good choice for the placement of ACLD, although it may hamper the effectiveness of ACLD at mode 1.

Concerning all these aspects, we decided to place the sectional dampers (PD and ACLD) in the middle of the plate covering an area of ca. $60 \times 30 \, mm$. As indicated in figure 3.7, the midpoint of the plate is antinode at mode 1, 4 and 9. For these modes, the damper performs significantly due to the great plate displacement / velocity / modal strain energy. Thus, by using this “direct” method and placing the damper at the midpoint, the 10 modes of the plate can be divided into 3 sorts:
• Damped modes: mode 1, 4 and 9 with antinodes at the midpoint. These modes are supposed to be greatly damped with sectional damper in the middle (PD and ACLD)

• Undamped modes: mode 2, 3, 5, 7, 8 and 10 with nodes at the midpoint. These modes are supposed to have no damping with sectional damper in the middle (PD and ACLD)

• Global modes: mode 6 sourcing from the eigenmode (torsion) of the frame

Figure 3.8: overlapped amplitudes of receptance and mobility

It is to mention that this “direct” method considers the most dominant modes in the structure. However, it loses the vibration controllability with dampers at modes with node position at the midpoint of the plate. Furthermore, other modes with insignificant amplitudes will hardly be concerned, since the superposition is merely an additive value in which small terms are relatively immaterial. If global optimization is wished, the methodology based on controllability gramian [92] can be applied. This algorithm takes the places with the greatest mean nodal energies at all modes for monitoring and suppressing the vibration. The difference between the two algorithms is that with the determined position of the direct
method, the 3 mentioned modes can be damped with maximized damper performances, because the damper is directly located at the anti-node. The difference of loss factors between the damped and undamped modes becomes distinct. If the method based on gramian is used, although every mode will be damped, but the compromise between sacrificing and balancing performances among different modes will be undertaken. The difference between modes will be reduced. For a sectional damper contributing little damping, it is hard to distinguish its own damping efficiency to the slight inherent damping from the measurement setup. To make sure that all three investigated damping approaches could be measured with significant loss factors and be compared with their best performances, the direct method was therefore used for the experiments in the second project phase.
3.3.4 Test conditions

The frequency of the sweep signal must be varied slowly enough to reach the
steady-state response of a system [42]. For instance, with the scanning interval
of each 0.1 Hz, the acquired vibration curve is normally closer to the reality
than with 1 Hz interval. In order to evaluate the precise behavior of the system,
the plates are to be tested also with possibly high resolutions for the frequency
scanning. In this situation, the required time for each measurement will be
greatly prolonged. It is important to find an optimum resolution with which
the behavior of the system can still be recorded without significant deviation
to the original vibration but with a reasonable effort for the measuring time.
For this purpose, a study for investigating the optimal resolution frequency was
conducted. Detailed information is reported in appendix C. The results show,
with the resolution of 1 Hz, the system behavior can already be quite precisely
detected together with the curve-fitting tool (see section 2.3.4). Consequently,
the specimens were measured from 10-400 Hz with the resolution of 1 Hz.

The behavior of a whole plate must be assessed by many measurement points
on the plate. A 5 × 5 point-grid was chosen as the measuring points. These
points are indicated as white spots in figure 3.8. The grid locates in the middle
area of the plate, where over 60% of the accumulated vibration amplitudes are
concentrated. The horizontal and vertical intervals between each two measuring
points are 50 and 40 mm respectively. For each mode, notable displacements
can be measured from the majority of these points. The measured 25 receptance
FRF (laser / impedance head) from 10-400 Hz were then as described in section
2.3.4, converted to a single FRF of RMS receptance. This receptance FRF was
further converted to a mobility FRF. To be more precise, there is one mobility
FRF curve for the final evaluation of each sample. The loss factor of each mode
was evaluated from this mobility FRF with the introduced curve-fitting tool.
Chapter 4

Screening of Damping Approaches

This chapter presents the results of various damping treatments introduced in chapter 2. The used setup and sample configurations can be referred to section 3.2. The efficiency of these approaches are then compared in relation with their weights. Some interesting approaches were selected for the further research phase.

4.1 Introduction

For the first step of investigating damping technologies which could be applied on panel structures, it is essential to obtain an overview of different damper categories and to compare their characteristics under possibly similar boundary conditions. Therefore, various damping approaches are investigated experimentally in this chapter. The damping materials or mechanisms are attached on simple beam structures to facilitate the measurements. The specimens are measured in a continuous frequency range. The focus of the study is to identify materials and mechanisms with low weight and outstanding damping performance. Surface treatments covering a considerable area on the panel structure, marginal treatments acting close to the edge of the panel structure and interface treatments set at the interface of the panel structure to its surrounding components were investigated (see figure 3.3).

As a remark for the results, a tiny phase lag can be measured on the reference specimen (an undamped aluminum beam) due to the friction between the sample and the supports. This can hardly be eliminated because of the sensitivity of the preload adjustment. The results of the other specimens contain the similar
preload phase lag. Since the measured damped specimens have much higher loss factors than the value of this undamped beam, this tiny phase lag is tolerable.

### 4.2 Viscoelastic damping

Three typical polymers widely applied in industry were used for the free and constrained layer damping. All these polymers have a density between ca. 1.7-1.9 g/cm$^3$. Two stiff bituminous polymers with the $E$-modulus (under test condition) about 0.5-0.8 GPa for type $a$) and 1.7-2.1 GPa for type $b$) can be applied for both FLD and CLD. The other soft butyl polymer with the $E$-modulus of 2-13 MPa shows only efficiency for CLD. For all CLD configurations, an aluminum foil with the thickness of 0.12 – 0.15 mm was used as the constraining layer. Additionally, the bituminous polymer $a$) was also tested with the configuration at interface.

<table>
<thead>
<tr>
<th>Sample No.</th>
<th>damping approaches</th>
<th>polymer type</th>
<th>comment</th>
<th>total weight [g]</th>
<th>loss factor $\eta_c [%]$</th>
<th>variant</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>base beam</td>
<td>-</td>
<td>-</td>
<td>21.5</td>
<td>0.010</td>
<td>-</td>
</tr>
<tr>
<td>2</td>
<td>FLD</td>
<td>bituminous polymer $a$)</td>
<td>damper volume $150 \times 40 \times 3$ mm</td>
<td>59.5</td>
<td>0.124</td>
<td>(a)</td>
</tr>
<tr>
<td>3</td>
<td>CLD</td>
<td>bituminous polymer $a$)</td>
<td>damper volume $150 \times 40 \times 2$ mm</td>
<td>47.0</td>
<td>0.145</td>
<td>(a)</td>
</tr>
<tr>
<td>4</td>
<td>CLD</td>
<td>bituminous polymer $b$)</td>
<td>damper volume $150 \times 40 \times 3$ mm</td>
<td>56.7</td>
<td>0.164</td>
<td>(a)</td>
</tr>
<tr>
<td>5</td>
<td>interface damping</td>
<td>bituminous polymer $a$)</td>
<td>damper volume $20 \times 40 \times 3$ mm</td>
<td>31.8</td>
<td>0.060</td>
<td>(c)</td>
</tr>
<tr>
<td>6</td>
<td>FLD</td>
<td>butyl rubber</td>
<td>damper volume $150 \times 40 \times 1.5$ mm</td>
<td>34.4</td>
<td>0.050</td>
<td>(a)</td>
</tr>
<tr>
<td>7</td>
<td>CLD</td>
<td>butyl rubber</td>
<td>damper volume $150 \times 40 \times 1.5$ mm</td>
<td>39.6</td>
<td>0.340</td>
<td>(a)</td>
</tr>
</tbody>
</table>

Table 4.1: results of viscoelastic dampers

Since the bituminous polymer is quite stiff, strain energies can be well dissipated in FLD configuration with sample No.2. Comparing No.2 and No.3, even less bituminous polymer was applied in the CLD configuration, but the measured loss factor is still higher than FLD. However the difference is not significant. By using the bituminous polymer $a$) at the interface, a phase lag caused by the in-
terface was able to be captured, even there was only few polymer incorporated to the specimen. The butyl rubber is relative soft and can barely absorb strain energy under extensional stress like the configuration of No.6. The CLD configuration using butyl polymer on No.7 provides considerable damping in comparison to FLD on No.6. This butyl rubber for typical CLD arrangement shows the best absolute performance among all investigated damping treatments. With these phenomena, it can be concluded for the viscoelastic damping:

- the performance of FLD and CLD are strongly dependent on the damping material.
- CLD is typically more efficient than FLD, the difference with soft polymers is bigger than with stiff polymers.
- interface damping provides moderate damping with few additional material only.

### 4.3 Piezoelectric damping

The configuration of ACLD specimens was designed similar to the CLD approach. One piezo sensor with the capacitance of 150 $nF$ and a blocking force of 90 $N$ was attached on the bottom of the specimen, then the viscoelastic and constraining layer were bonded to the sensor. Finally one same piezo plate was bonded on the top of the constraining layer as actuator. The gain value between the sensor and actuator is totally 500 by using two amplification devices (direct position feedback). The displacement of the specimen was directly recorded from the surface of the piezo actuator.

<table>
<thead>
<tr>
<th>Sample No.</th>
<th>damping approaches</th>
<th>comment</th>
<th>total weight [g]</th>
<th>loss factor $\eta_c$ [-]</th>
<th>variant</th>
</tr>
</thead>
<tbody>
<tr>
<td>8</td>
<td>(A)CLD</td>
<td>damper 61 $\times$ 35 $\times$ 1.5 mm with butyl rubber</td>
<td>37.5</td>
<td>0.260</td>
<td>(a)</td>
</tr>
<tr>
<td>9</td>
<td>ACLD</td>
<td>damper 61 $\times$ 35 $\times$ 1.5 mm with butyl rubber</td>
<td>117.6</td>
<td>0.620</td>
<td>(a)</td>
</tr>
</tbody>
</table>

Table 4.2: results of piezoelectric dampers

Due to the outstanding damping efficiency of butyl rubber, the passive (A)CLD specimen without voltage control affords already a high loss factor.
By the application of the voltage up to 250 V, a tremendous phase lag was measured. This phase lag was associated with the interactive motion of the actuator and induced moment on the structure and at the supports, which may affect the correct damping angle of the structure. But from this measured phase lag one may still expect a great damping effect. Although the ACLD alone is lightweight, the electronic parts of ACLD have a negative influence to the weight efficiency as can be seen from No.9.

4.4 Magnetic damping

Some specimens using magnetic forces were investigated with the marginal treatment variant shown in figure 3.3 (b) with auxiliary fixed magnets. For each sample there were totally four magnetic elements assembled. Two $\phi 20 \times 2$ permanent magnets (PM) with the remanence of 1.3 T were glued on the bottom of the specimen near the support points. The other two auxiliary PM or electromagnets (EM) with the size of $\phi 20 \times 30$ and 12 W, which exert repulsive forces to the ones glued on the sample, were fixed with additional frames separately below the specimen. The magnetic field was adjusted with different distances between PM and EM or the voltage values on EM as listed in table 4.3. In addition to these samples, a specimen using extensional MCLD was also tested with four PM at the same location.

Due to the high stiffness of the bituminous polymer a) as used in sample No.2, the extensional deformation of the viscoelastic layer is very limited with the configuration of No.10. That results in its low loss factor. The inevitable high weight of solenoid in EM on sample No.12-No.17 hampers the effectiveness of the lightweight performance significantly. The adjustable magnetic field imposed to the system incurs different damping force and efficiencies. From figure 4.1 it can be seen that with the increasing voltage or decreasing distance between the PM and the EM, the loss factor will be augmented nonlinearly. The characteristics of magnetic dampers are:

- can be only used where the environment is not sensitive to electric or magnetic disturbance.
- nonlinear damping behavior in terms of the vibration amplitude.
- compact design with PM-PM considering weight and volume, but inefficient with PM-EM design due to the high weight of EM.
### magnetic damping

<table>
<thead>
<tr>
<th>Sample No.</th>
<th>damping approaches</th>
<th>comment</th>
<th>total weight [g]</th>
<th>loss factor $\eta_c$ [-]</th>
<th>variant</th>
</tr>
</thead>
<tbody>
<tr>
<td>10</td>
<td>MCLD</td>
<td>$4 \times \phi 20 \times 2 \text{ mm PM}, \text{ damper with bituminous polymer } a), \ 2 \times \phi 20 \times 3 \text{ mm}$</td>
<td>44.2</td>
<td>0.062</td>
<td>(b)</td>
</tr>
<tr>
<td>11</td>
<td>PM-PM damper</td>
<td>distance between two PM 7 mm</td>
<td>40.7</td>
<td>0.098</td>
<td>(b)</td>
</tr>
<tr>
<td>12</td>
<td>PM-EM damper</td>
<td>distance between two PM 7 mm, $V=6 \text{ V}$</td>
<td>173.2</td>
<td>0.074</td>
<td>(b)</td>
</tr>
<tr>
<td>13</td>
<td>PM-EM damper</td>
<td>distance between two PM 7 mm, $V=9 \text{ V}$</td>
<td>173.2</td>
<td>0.091</td>
<td>(b)</td>
</tr>
<tr>
<td>14</td>
<td>PM-EM damper</td>
<td>distance between two PM 5 mm, $V=6 \text{ V}$</td>
<td>173.2</td>
<td>0.079</td>
<td>(b)</td>
</tr>
<tr>
<td>15</td>
<td>PM-EM damper</td>
<td>distance between two PM 5 mm, $V=9 \text{ V}$</td>
<td>173.2</td>
<td>0.091</td>
<td>(b)</td>
</tr>
<tr>
<td>16</td>
<td>PM-EM damper</td>
<td>distance between two PM 3 mm, $V=6 \text{ V}$</td>
<td>173.2</td>
<td>0.099</td>
<td>(b)</td>
</tr>
<tr>
<td>17</td>
<td>PM-EM damper</td>
<td>distance between two PM 3 mm, $V=9 \text{ V}$</td>
<td>173.2</td>
<td>0.103</td>
<td>(b)</td>
</tr>
</tbody>
</table>

Table 4.3: results of magnetic dampers

![Figure 4.1: performance of magnetic damping with PM-EM](image-url)
4.5 Particle damping

Particle dampers with sand and steel powder were attached to the specimens with two types of structures. The first type was applied to an aluminum beam like all other investigated specimens. The second type was applied on sandwich structures. A sandwich with honey comb core was filled with particles in some of its hollow cells, which is not comparable to other samples.

4.5.1 Regular particle damper

In the first type, one external container was fixed on the bottom of the specimen in the centre. Two containers with different dimensions were filled simply with particles or a bean bag containing same quantity or volume of particles as shown in table 4.4.

The particle dampers No.18 and No.19 using same quantity of particle weight show similar performance in simply filled configuration and BBD with slim containers ($20 \times 20 \times 22 \ mm$). The loose bean bags were barely pressed in the vertical direction inside the container. The chances of particle collisions for both arrangements are almost equal which leads to the comparable loss factors. On the No.20 and No.21, the same quantity of particles as used on No.18 and No.19 is packed into a flat container ($40 \times 25 \times 4 \ mm$). The flat container packed the particles more compact, the chances of the collision and impulse exchange between particles and the container cover walls as well is increased in the vibration direction. Therefore the measured loss factors with this flat container are generally higher than the values with the slim container. Also simple particle damper and BBD configuration were compared with No.20 and No.21. Here they still show small difference. But it is to mention that the bean bag of No.21 was more tightly sealed than the bag of No. 19 and it was slightly pressed in the container. The bean bag behaved more like a solid. Thus the performance of No.21 is lower than No.20, in which the particles could move freely in the container. On No.22 and No.23 the same volume of particles were applied to the slim container again. Due to the higher density / weight of steel particles, more impact energy was absorbed in No.23 than No.22 with same volume. The behaviors of these samples show:

- the shape of the particle container is important for the performance, it should be adjusted to an ideal dimension that maximizes the impulse exchange.

- the quality of BBD is dependent on the degree of the bag tightness and bag shape, it showed no significant improvement in comparison to simple particle damper in this study.
4.5. Particle damping

<table>
<thead>
<tr>
<th>Sample No.</th>
<th>damping approaches</th>
<th>particles</th>
<th>comment</th>
<th>total weight [g]</th>
<th>loss factor $\eta_c$ [-]</th>
<th>variant</th>
</tr>
</thead>
<tbody>
<tr>
<td>18</td>
<td>particle damper</td>
<td>sand</td>
<td>container volume $20 \times 20 \times 22 \text{ mm}$, with particles of 10% total weight (sample)</td>
<td>34.1</td>
<td>0.065</td>
<td>(a)</td>
</tr>
<tr>
<td>19</td>
<td>BBD</td>
<td>sand</td>
<td>container volume $20 \times 20 \times 22 \text{ mm}$, with particles of 10% total weight (sample)</td>
<td>34.1</td>
<td>0.064</td>
<td>(a)</td>
</tr>
<tr>
<td>20</td>
<td>particle damper</td>
<td>steel</td>
<td>container volume $40 \times 25 \times 4 \text{ mm}$, with particles of 10% total weight (sample)</td>
<td>34.2</td>
<td>0.100</td>
<td>(a)</td>
</tr>
<tr>
<td>21</td>
<td>BBD</td>
<td>steel</td>
<td>container volume $40 \times 25 \times 4 \text{ mm}$, with particles of 10% total weight (sample)</td>
<td>34.2</td>
<td>0.088</td>
<td>(a)</td>
</tr>
<tr>
<td>22</td>
<td>particle damper</td>
<td>sand</td>
<td>container volume $20 \times 20 \times 22 \text{ mm}$, with particles of 50% height of the particle container</td>
<td>35.5</td>
<td>0.075</td>
<td>(a)</td>
</tr>
<tr>
<td>23</td>
<td>particle damper</td>
<td>steel</td>
<td>container volume $20 \times 20 \times 22 \text{ mm}$, with particles of 50% height of the particle container</td>
<td>40.5</td>
<td>0.092</td>
<td>(a)</td>
</tr>
<tr>
<td>24</td>
<td>particle damper</td>
<td>steel</td>
<td>container volume $40 \times 25 \times 4 \text{ mm}$, with particles of 100% filled container</td>
<td>40.1</td>
<td>0.121</td>
<td>(a)</td>
</tr>
</tbody>
</table>

Table 4.4: results of particle dampers

- with higher density / weight / quantity of the particles, the performance will be improved.

- particle dampers do not have strong temperature dependence for the damping and thus have reliable performance under harsh environment.
4.5.2 Particle damper for honeycomb sandwich

For the sandwich structures, the particles with the same weight of 5 g were filled into the sandwich cells with the filling ratios in thickness direction as shown in figure 4.2 from 0 to 100% with the core thicknesses of 2, 4 and 7 mm respectively. The mass ratio between the particles and the sandwich is about 0.1.

![Figure 4.2: filled honey comb sandwich with different filling ratios](image)

The sandwich particle dampers could not be compared to other samples, because this configuration has a significantly higher bending stiffness than the other specimens based on the bare aluminum beam. Their results are illustrated in figure 4.3. As it can be observed from the figure, with the increase of particle density (sand to steel), the damping is augmented about up to 0.02 under the same filling ratio. By increasing the core thickness, the chances of the contact transfer of the particles will be improved and thus result in an improvement of loss factor for about 0.01-0.03. The sandwich particle dampers show following characteristics:

- the performance is better with a higher filling ratio in the vibration direction or close to the excitation.
- if more particles are distributed into a single room, the performance is better.
- unlike the other dampers, for stiff structures like honeycomb sandwich, the PD can also perform well, where the local deformation is small.
- steel particles performs better than sand.
4.6 Coulomb / electrostatic damping

These results show that PD can be applied even to very stiff structures. In an additional study, PD was integrated into a stiff sandwich plate, which has aluminum facings and a PET-foam core. The measurements and results are listed in appendix B.

![Figure 4.3: loss factor of sandwich particle damper](image)

4.6 Coulomb / electrostatic damping

One dielectric layer (PI-film) with the thickness of 25 or 75 µm was glued onto the whole top surface of the aluminum specimen. Above this glued substrate, one steel beam with the same geometry as the aluminum specimen was placed. The excitation was exerted on the top of the steel beam. These two metal sheets serve as electrodes. The maximum applied voltage for the electrostatic dampers were 1.8 kV and 3.6 kV for the 25 and 75 µm polyimide films respectively. The normal electrostatic stress provided by the accumulated charges on the two surfaces of the dielectric film produces friction between the PI film and steel beam.
Table 4.5: results of electrostatic dampers

<table>
<thead>
<tr>
<th>Sample No.</th>
<th>damping approaches</th>
<th>comment</th>
<th>total weight [g]</th>
<th>loss factor $\eta_c$ [-]</th>
<th>variant</th>
</tr>
</thead>
<tbody>
<tr>
<td>25</td>
<td>multi-layer electrostatic damper</td>
<td>25 $\mu$m insulation layer, max. loss factor at $V=1200$ V</td>
<td>70.9</td>
<td>0.078</td>
<td>(a)</td>
</tr>
<tr>
<td>26</td>
<td>multi-layer electrostatic damper</td>
<td>75 $\mu$m insulation layer, max. loss factor at $V=2700$ V</td>
<td>71.7</td>
<td>0.075</td>
<td>(a)</td>
</tr>
</tbody>
</table>

We can see in figure 4.4, the loss factors are raised for both samples with increasing electric potential at the beginning, which reveals with larger electrostatic stress, bigger friction forces can be provided. A maximum of the loss factor will be reached, until the sliding friction disappears and only static friction exists with the increasing electrostatic stress. Then the loss factor drops down. The maximal loss factors of both samples have only slight difference. The voltage at the maximum loss factor of No.26 is higher than No. 25, which means for a thicker layer, a stronger potential is needed to generate the similar electrostatic stress than a thin layer. It can be concluded from the multi-layer electrostatic damper:

- less thickness of the dielectric layer induces abrasion, damage and breakdown of the insulator, higher thickness requires extremely high electric potential, a compromise must be made for the thickness selection of this dielectric layer.

- the optimum value of the loss factor is decided by the threshold value of the sliding friction.

### 4.7 Eddy current damping

Three types of eddy current damper (figure 4.5) using same materials but different configurations and dimensions were investigated. All of them were set in the vicinity of the two support points like variant (b) of marginal treatment in figure 3.3. In the first type, two $\phi 20 \times 2$ mm permanent magnets (PM) disks were bonded below the specimen and they were inserted into a copper tube with the inner diameter of $\phi 22$ mm. In the second type, eight small $\phi 8 \times 3$ mm PM arrays were bonded like the first type onto the specimen and they were inserted
4.7. Eddy current damping

Figure 4.4: result curves of the electrostatic damper

into two massive copper blocks \((40 \times 40 \times 15 \text{ mm}^3)\) with also eight holes of \(\phi 10\ \text{mm}\). In the third type, two \(40 \times 40 \times 2 \text{ mm}^3\) copper sheets were glued downwards and PM were placed 3 mm beneath the copper sheets. The PM have similar remanences as the one used for magnetic dampers.

These tested ECD configurations are not effective. Both magnetic mass and inductive part are too heavy for lightweight design. On the tube dampers No.27 and No.28, few conductor / magnets were used. They are compactly built compared to other ECD dampers. Damping forces can be introduced in the whole vibration cycle, because the total insertion of the magnets into the copper tube provides a stable damping force constantly during the vibration. Although the loss factors are not the best ones, they are more weight efficient than the other ECD arrangements.

The damping force of ECD is proportional to the conductivity of the conductor and provided magnetic flux from the magnetic part. The used copper and NdFeB magnets have already the best efficiency from this point of view. It’s hard to raise the damping through the material efficiency. Augmenting the volume of the conductor / magnet can be one solution. In order to check if by applying bulky conductor volume, the damping can be further improved, it is investigated on No.29 with massive block conductors. By employing these massive conductors, the improvement of damping values is quite limited concerning
the added weight. Redundant material of the conductor doesn’t contribute to the damping significantly.

For the sheet ECD, the damping force is not tremendous in the whole vibration cycle like the tube ECD, because the force is dependent on the vibration position of the specimen. At the position far from the magnets during the vibration, the damping force is lower and conversely it is higher at a position closer to the magnets like magnetic damper. Therefore, although stronger magnets were used for sheet ECD, the measured loss factors on the sheet ECD were similar to the tube ECD, but with relative low weight efficiencies. In order to test the influence of adding more magnets to the loss factors, twice PM were applied on No.31 than No.30. At the cost of supplementing 15 $g$ magnets into the system, the improvement of the loss factor for 0.01 can be perceived in No.31 compared to No.30. The relative velocity of the damper also affects the damping force. This effect can be compared with No.31 and No.32. They use the same configuration. No.31 was excited with a normal force as all other samples. No.32 was excited with a force whose amplitude is three times higher than usual. With this great excitation force and velocity, a higher loss factor was achieved. Hence, the application of ECD is more efficient for high speed vibrations. Following aspects are important for the design of ECD:

- ECD performs better with high conductivity of the conductor and remanence of the magnets.
### 4.8. MR fluid damping

**Eddy current damping**

<table>
<thead>
<tr>
<th>Sample No.</th>
<th>damping approaches</th>
<th>type</th>
<th>comment</th>
<th>total weight [g]</th>
<th>loss factor $\eta_c$ [-]</th>
<th>variant</th>
</tr>
</thead>
<tbody>
<tr>
<td>27</td>
<td>ECD</td>
<td>tube</td>
<td>$2 \times$ copper tube $\phi_{25 \times 25,mm}$, $1 \times$ PM $\phi_{20 \times 2,mm}$</td>
<td>42.1</td>
<td>0.044</td>
<td>(b)</td>
</tr>
<tr>
<td>28</td>
<td>ECD</td>
<td>tube</td>
<td>$2 \times$ copper tube $\phi_{25 \times 25,mm}$, $2 \times$ PM $\phi_{20 \times 2,mm}$</td>
<td>51.7</td>
<td>0.062</td>
<td>(b)</td>
</tr>
<tr>
<td>29</td>
<td>ECD</td>
<td>block</td>
<td>$2 \times$ copper block $40 \times 40 \times 15,mm$, $8 \times$ PM $\phi_{8 \times 3,mm}$ array</td>
<td>377.0</td>
<td>0.063</td>
<td>(b)</td>
</tr>
<tr>
<td>30</td>
<td>ECD</td>
<td>sheet</td>
<td>$2 \times$ copper sheet $40 \times 40 \times 2,mm$, $2 \times$ PM $20 \times 20 \times 5,mm$</td>
<td>108.8</td>
<td>0.054</td>
<td>(b)</td>
</tr>
<tr>
<td>31</td>
<td>ECD</td>
<td>sheet</td>
<td>$2 \times$ copper sheet $40 \times 40 \times 2,mm$, $4 \times$ PM $20 \times 20 \times 5,mm$</td>
<td>123.8</td>
<td>0.065</td>
<td>(b)</td>
</tr>
<tr>
<td>32</td>
<td>ECD</td>
<td>sheet</td>
<td>$2 \times$ copper sheet $40 \times 40 \times 2,mm$, $4 \times$ PM $20 \times 20 \times 5,mm$, with stronger excitation force</td>
<td>123.8</td>
<td>0.072</td>
<td>(b)</td>
</tr>
</tbody>
</table>

Table 4.6: results of eddy current dampers

- the positioning of the conductor / magnet is important. A bigger distance deteriorates the damping efficiency because of the lack of inductance for eddy current. Inversely, with very small clearance, additional friction might be generated between them and the systems could be stuck.

- the damping force is dependent on the vibration behavior. ECD works better for larger displacement / velocity / excitation.

- the configuration of the conductor / magnets is essential for the performance. The compact tube ECD has the best performance due to the constant existing damping force. The block ECD is ineffective due to the heavy weight and unfavorable distribution of the conductor. The sheet dampers are not efficient due to the unstable damping force and amplitude dependence of the vibration.
4.8 MR fluid damping

Two MR dampers \((20 \times 20 \times 26 \ mm^3)\) made of brass were mounted at the interface of the specimen. The damper design is similar to commercially available MR dampers (figure 4.6): The piston rod in the damper is on the upper end connected to the specimen according to the variant (c) of interface treatment in figure 3.3 and on the bottom end connected to a plate with orifices making friction forces. By adjusting the magnetic field from the PM attached to the outer damper walls, variable damping forces can be obtained. The applied MR fluid has a density of \(2.3 \times 10^3 \ kg/m^3\) and the particle percentage of 72.9% of weight. The applied PM are composed of the same NdFeB material as used for ECD dampers.

![Figure 4.6: sectional view of the MR damper](image)

### Table 4.7: results of magnetorheological dampers

<table>
<thead>
<tr>
<th>Sample No.</th>
<th>damping approaches</th>
<th>comment</th>
<th>total weight [g]</th>
<th>loss factor (\eta_c) [-]</th>
<th>variant</th>
</tr>
</thead>
<tbody>
<tr>
<td>33</td>
<td>MR fluid</td>
<td>without external magnetic field</td>
<td>101.0</td>
<td>0.061</td>
<td>(c)</td>
</tr>
<tr>
<td>34</td>
<td>MR fluid</td>
<td>with four (20 \times 4 \times 2\ mm) PM on the container wall</td>
<td>105.8</td>
<td>0.074</td>
<td>(c)</td>
</tr>
<tr>
<td>35</td>
<td>MR fluid</td>
<td>with two (20 \times 20 \times 5\ mm) PM with (6\ mm) spacer</td>
<td>161.0</td>
<td>0.091</td>
<td>(c)</td>
</tr>
<tr>
<td>36</td>
<td>MR fluid</td>
<td>with two (20 \times 20 \times 5\ mm) PM on the container wall</td>
<td>161.0</td>
<td>0.110</td>
<td>(c)</td>
</tr>
</tbody>
</table>
The high weight ratio of the particles in the MR fluid and the inevitable weight from the container cause a high basic weight of this approach. In No.33, even no magnetic parts were used, with which the MR fluid should be controlled principally, the weight of the sample is already higher than many other damping approaches. By adding only 4.8 g magnets, the loss factor of No.34 is improved by 17%. Also here in order to check the influence of imposing stronger magnetic field to the contribution of the loss factor, No.35 and No.36 were tested with bigger magnets. In No.35, the magnets were placed 6 mm away from the container wall with spacers (plastic frames). The loss factor was improved with 0.03 compared to No.33 without magnets. In No.36, the magnets with the same dimension of No.35 were directly bonded to the container wall. Further 17% improvement could be measured in comparison to No.35. Among all investigated samples, the loss factor increase with the augmentation of magnetic fields. Principally if the friction force provided from the MR fluid is big enough to stick the rod, the decrease of loss factor should be observed due to the static friction. But with the configurations in this section, this force of the threshold was not reached. From the results it can be seen:

- the achieved loss factors of MR dampers depend strongly on the applied magnetic field and its distribution. The highest loss factor measured with the strongest magnetic field was almost 80% higher than the one without magnetic field.

- the enhancement of magnetic field is firmly related with supplementing weight of the auxiliary solenoid or permanent magnets. Accompanied with high percentage of particle weight in MR fluid, this method is not really weight efficient.

- at optimum point, where the magnetic field maximizes the sliding friction with the particles should be designed for the damper.

- the performance of MR damper is tunable, if solenoid is used for the control.

### 4.9 Comparison

In figure 4.7, the results aforementioned are summarized. The loss factors are plotted with their physical principles in a) and boundary variants in b). In a), we may observe that the performances of viscoelastic dampers with bituminous or butyl polymers are excellent in general. Compared to them, the particle dampers show also decent efficiencies but without the drawback of temperature sensitivity. The other methods possess moderate performances owing to their mechanical complexity. Among them, the ECD and electrostatic damper have good performance. The MR fluid and magnetic damper have poor efficiency due
to the high weight of magnets.

In b) it can be seen, the variant 'a' of surface treatment exhibits most promising results. The variant 'c' of interface treatment has the middle efficiency. Nevertheless, viscoelastic sample in variant 'c'(sample No.5 in table 4.1) performs fair damping which may be valuable for interface application concerning the amount of the applied material. The variant 'b' of marginal treatment is most ineffective.

![Figure 4.7: specific damping efficiencies on a) damping principles, b) variants](image)

**4.10 Conclusion**

In this chapter, the performances of different damping approaches for beam structures were experimentally compared. Both structural and interface damping treatments were investigated. Special attention was paid to their efficiency related to weight. Specimens treated with various approaches were tested in 3 point bending condition. The phase lag between excitation and response of the specimen was measured in time domain and the structural loss factor was extracted. About 40 different configurations in passive and active designs,
including viscoelastic treatments, piezoelectric, magnetic, particle, coulomb / electrostatic, eddy current and magnetorheological fluid dampers were tested in the frequency range of interest up to 400 $Hz$.

According to the measured system loss factors, the assessment of the specific loss factors for each approach is given. Damping with classical viscoelastic materials offers most excellent results. The interface damping shows its potential with viscoelastic material by proper interface design. The piezoelectric elements show promising potential concerning their low weight and possible active forces. Owing to the high density and weight of the metallic parts, the performances of the magnetic and MR fluid dampers are generally moderate. Particle dampers are simple and provide reliable damping which are close to the viscoelastic damper but independent of temperature change. The electrostatic damper is not efficient in either damping or weight. Through accurate locating, the eddy current dampers may supply higher efficiencies with compact design.

![Figure 4.8: estimation of damping techniques](image)

- FLD: free layer
- CLD: constrained layer
- PM: permanent magnet
- PD: particle
- BBBD: bean bag
- MCLD: magnet constrained layer
- IF: interface
- ECD: eddy current
- PIP: passive piezo shunt
- MR: magnetorheological
- EM: electromagnet
- ACLD: active constrained
- ES: electrostatic
The estimation of the damper performance ranges is summarized in figure 4.8. Concerning the practical exploitation of these various damping techniques, the particle damper is outstanding for economic application and has also decent efficiency. The design of interface damping via viscoelastic material can be a new method for plate damping in comparison to conventional surface-cover treatments. The active constrained layer damper seems to be an excellent solution for lightweight design. For these reasons, these three approaches were further studied in the following research phase.
Chapter 5

Particle Damping Treatment

The experimental results of PD in the second project phase are presented in this chapter. Six different PD are compared with each other. The applied test rig and base specimen configurations are described in section 3.3.

5.1 Damper configurations

In order to survey the parameters of particle damping, six different particle dampers were manufactured as listed in table 5.1. The cross-section areas of the dampers are 60 \( \text{mm} \) in length and 30 \( \text{mm} \) in width. Only the depths are varied. In three of them, additional honeycomb cells with the cell size of 3.2 \( \text{mm} \) are bonded inside the damper container (see figure 5.1). The applied particles have the size between ca. 0.3-0.9 \( \text{mm} \).

![Figure 5.1: particle damper containers](image)

The purpose of using honeycomb cells in the container is to investigate the influence of container size to the damping efficiency between the dampers with and without any interim cell walls. The augmentation of the damping with
Chapter 5. Particle Damping Treatment

<table>
<thead>
<tr>
<th>No.</th>
<th>type</th>
<th>damper depth</th>
<th>total damper weight [g]</th>
<th>weight of particle [g]</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>simple</td>
<td>2 mm</td>
<td>25.9</td>
<td>8.3</td>
</tr>
<tr>
<td>2</td>
<td>simple</td>
<td>2 × 2 mm</td>
<td>45.2</td>
<td>15.6</td>
</tr>
<tr>
<td>3</td>
<td>honeycomb</td>
<td>2 mm</td>
<td>24.7</td>
<td>6.8</td>
</tr>
<tr>
<td>4</td>
<td>honeycomb</td>
<td>2 × 2 mm</td>
<td>44.7</td>
<td>14.0</td>
</tr>
<tr>
<td>5</td>
<td>simple</td>
<td>4 mm</td>
<td>40.9</td>
<td>16.2</td>
</tr>
<tr>
<td>6</td>
<td>honeycomb</td>
<td>4 mm</td>
<td>39.1</td>
<td>13.2</td>
</tr>
<tr>
<td>(7)</td>
<td>honeycomb substitute</td>
<td>2 mm</td>
<td>24.6</td>
<td>0</td>
</tr>
<tr>
<td>(8)</td>
<td>honeycomb substitute</td>
<td>2 × 2 mm</td>
<td>44.6</td>
<td>0</td>
</tr>
<tr>
<td>(9)</td>
<td>honeycomb substitute</td>
<td>4 mm</td>
<td>38.9</td>
<td>0</td>
</tr>
</tbody>
</table>

Table 5.1: particle damper configurations

doubled quantity of particles can be compared with the 2 mm and 4 mm thick samples (No.1 vs. No.5, No.3 vs. No.6). By separating a 4 mm damper into 2 × 2 mm with a cover wall in the depth direction (No.2 vs. No.5, No.4 vs. No.6), the relationship of energy dissipation between the particles and between particles and cover wall can be determined. There were further three damper substitutes tested which have the same weight as the honeycomb dampers but don’t contain any particles. With these substitutes, the performance of the particle dampers can be compared with the undamped structure transversely under the same weight and stiffness condition.

The plate was bonded to the frame by a thin epoxy layer. Because there are a lot of PD configurations, in order to minimize the internal damping difference of different specimens, the different PD were bonded to the same specimen flexibly. On the contact surface between PD and plate, a thin double-sided tape was applied. On the contrary side of PD, it was firmly fixed to the plate surface with a thin polyimide foil. After each measurement, the PD was removed and replaced by a new one to be measured.

5.2 Results

Due to the plenty number of measured PD specimens, the comparison is arranged in the following sequence:

- comparison of simple PD
- comparison of honeycomb PD
• comparison of honeycomb PD and weight substitute specimens

Comparison of simple PD

The simple PD are not efficient (see figure 5.2) in the experiments. Their loss factors at the damped modes are less than $2 \times 10^{-2}$ on all samples. These values are merely slightly larger than an undamped plate. The difference of damping between samples is not distinct. The low damping is caused by the test setup: The flat particle damper is placed vertically in the experiment. The specimen was excited in the horizontal direction with the shaker. The particles in the bottom layers are compressed by the particles in the upper layers through the gravitation and the upper part of the damper is hollow. During the horizontal vibration, particles are more strongly pressed and tightened with the increase of the damper length downwards. The force chains generated by the compression between particles provide static instead of sliding frictions, which hampers the motion of particles and the energy dissipation as well. Only at the top part in the damper cavity, minor numbers of particles vibrate freely. Due to the diminishing of the majority of effective particles, the simple PD is added in fact as nothing else than a weight slug with slight damping effects. Thus, these simple PD don’t perform appropriately in the tested configuration and cannot present the real behavior of PD due to the deficient placement and damper structure design. The deviation of the loss factors can be quite large compared to the slight measured damping, because the exact quantities of the unconstrained particles in each PD stay unknown. For the vibration in horizontal direction, the big length-depth ratio of the dampers augments the compression effect caused by gravity, which should be avoided.

<table>
<thead>
<tr>
<th>No.</th>
<th>damper</th>
<th>mode 1</th>
<th>mode 4</th>
<th>mode 8/9</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>amp. [dB]</td>
<td>$\eta_c [10^{-2}]$</td>
<td>amp. [dB]</td>
</tr>
<tr>
<td>1</td>
<td>2 mm</td>
<td>74.5</td>
<td>1.47</td>
<td>55.5</td>
</tr>
<tr>
<td>2</td>
<td>2 x 2mm</td>
<td>72.2</td>
<td>1.97</td>
<td>56.7</td>
</tr>
<tr>
<td>5</td>
<td>4 mm</td>
<td>72.9</td>
<td>1.68</td>
<td>54.3</td>
</tr>
</tbody>
</table>

Table 5.2: simple PD results at damped modes

Comparison of honeycomb PD

In contrast to the simple PD with a single container cavity for particles, the influence of the container size was investigated with the honeycomb dampers, in which the same quantity particles are separated in several small sections instead of a single big cavity. With many 3.2 mm honeycomb cells, the damper is
divided into more than 100 separate small cavities. In this case, although there are almost same weight of particles in this honeycomb damper as the simple PD, most of the particles in each cell are not tightly compressed by the gravity due to the moderate amount. By locating in vertical direction, the particles are still able to move in the small honeycomb cells during the horizontal vibration. All these active particles contribute to significant damping, as can be seen in table 5.3. The measured loss factors of the honeycomb PD are 1.8-3.9 times higher than those simple PD at the damped modes. With doubled weight, the loss factors of the 4 mm honeycomb damper are improved by 11-29% compared to the 2 mm honeycomb damper. Hence, separating cavities in the proper direction is meaningful to avoid the compression caused by gravity.

The influence of the cover wall (see specimen description) was determined with the specimens of $2 \times 2$ mm honeycomb and 4 mm honeycomb damper (No.4 and No.6 in figure 5.1). With even less weight of particles in No.6, it performs even better than No.4. The cover wall reduces the amount of the particles and their chances of collision with each other in a single closed cavity. This results in the degradation of the damper performance. Therefore, with the same amount of particles, the damper with a single cavity in the vibration direction surpasses those with several cavities, if most particles are not highly constrained for their motions. i.e., separating cavities across the vibration direction is detrimental for
5.2. Results

![Graph showing FRF of honeycomb PD](image)

**Figure 5.3: FRF of honeycomb PD**

<table>
<thead>
<tr>
<th>No.</th>
<th>damper</th>
<th>mode 1</th>
<th>mode 4</th>
<th>mode 8/9</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>amp. [dB]</td>
<td>$\eta_c$ $[10^{-2}]$</td>
<td>amp. [dB]</td>
</tr>
<tr>
<td>3</td>
<td>2 mm honeycomb</td>
<td>65.8</td>
<td>4.68</td>
<td>51.5</td>
</tr>
<tr>
<td>4</td>
<td>2 x 2mm honeycomb</td>
<td>64.7</td>
<td>5.48</td>
<td>52.2</td>
</tr>
<tr>
<td>6</td>
<td>4 mm honeycomb</td>
<td>63.4</td>
<td>6.58</td>
<td>51.4</td>
</tr>
</tbody>
</table>

**Table 5.3: results of honeycomb PD at damped modes**

the damping performance.

**Comparison of honeycomb PD and weight substitute specimens**

As the honeycomb PD (No.3,4 and 6) performs best, they were further compared with the weight substitute samples No.7-9, in order to survey the absolute performance of honeycomb PD in contrast to undamped structures. On the substitute samples, one single metal slug with the same weight as honeycomb damper is bonded at the midpoint of the plate. Because both weight and stiffness of the two type samples are the same, accurate comparisons of the damping can
be directly observed in figure 5.4 and table 5.4.

We still observe the damped modes 1, 4 and 9. The honeycomb PD are generally very effective for suppressing the first mode. A decay of 13-16 $dB$ with the honeycomb PD corresponds to an amplitude attenuation of 78-84%. For mode 4 and 9, the amplitudes of particle damper samples were reduced in the range of 2-5 $dB$ compared to the weight substitute samples. The loss factors were improved by 27-60%.

![Figure 5.4: FRF of the honeycomb PD vs. undamped sample with same weight](image)

The loss factors of the undamped and global modes are listed in table 5.5. Like the behavior of ACLD at these modes, the difference of the values is also indistinct with PD. No additional damping can be attained at undamped and
Table 5.4: results of honeycomb PD with equivalent weight samples at damped modes

Table 5.5: loss factors of honeycomb PD at undamped and global modes

An insight of the damping efficiency between undamped sample, with simple PD and honeycomb PD damped sample (4 mm thick weight slug / damper) is given by figure 5.5. In each plot, the mean mobility of the accumulated magnitude on the 25 measured points from 10-400 Hz is shown. Like shown in figure 3.8, the vibration of an undamped plate (substitute) looks like the dominant 1st mode. The sample with simple PD has similar behavior. But the amplitude at the midpoint is slightly mitigated and the magnitudes at each point are globally lower than the undamped sample. The concave surface of the honeycomb damper sample shows remarkable amplitude attenuation at the midpoint. Even the points in its vicinity are correspondingly damped. The
vibrations at each point are further damped in contrast to simple PD.

![Figure 5.5: vibration shapes of the 4 mm PD specimens](image)

**5.3 Conclusion**

Concisely, the characteristics of PD, that were observed in the experiments and from the results, can be concluded as following:

- the appropriate applied PD has superior damping capability with relative good stability (insensitive to temperature or frequency).
- the behavior of PD is dependent on the vibration direction which should be highly considered for the optimal shape design and placement of the PD. The vibration direction is the effective direction of applying PD. PD should have a big depth-length ratio, i.e., possess possibly less cavities in the vibration direction and more particles should be filled into one cavity in order to raise the efficiency.
- the effect of compression of the tightened particles should be generally avoided in any configuration. A cavity with a filling ratio of 90-95% is optimal [93].
- the cavities should be vertically kept with proper size for avoiding gravity compression in the case of horizontal excitation.
- cellular insert is a solution to solve the ”anisotropic” damping behavior of PD
- at damped modes, PD performs especially excellent for the first mode. Modes of higher order can still be well damped. PD should be applied at the position with great displacement / velocity.
• at undamped and global modes, the damping effect of PD is negligible
• structure-borne noise can be effectively damped by PD
Chapter 6

Active Constrained Layer Damping Treatment

The research of ACLD in the second project phase is represented in this chapter. The results of four different ACLD are shown. The applied experimental setup and base specimen configurations can be referred to section 3.3.

6.1 Damper configurations

The ACLD system consists of the damping material (VEM), sensor, actuator and the amplification loop for the actuator. The VEM used in ACLD is the same one as used for ID (see chapter 7). As sensor and actuator, two identical DuraAct™ patches [94] were used. The properties of the used piezoelectric sensor and actuator are shown in table 6.1.

<p>| | |</p>
<table>
<thead>
<tr>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>operating voltage</strong></td>
<td>−50 ~ 200 V</td>
</tr>
<tr>
<td><strong>holding force</strong></td>
<td>90 N</td>
</tr>
<tr>
<td><strong>electrical capacitance</strong></td>
<td>150 nF</td>
</tr>
<tr>
<td><strong>size</strong></td>
<td>61 × 35 × 0.4 mm</td>
</tr>
</tbody>
</table>

Table 6.1: properties of piezoelectric elements

The ACLD is controlled with position feedback algorithm. The amplification loop is composed of two steps (see figure 3.4): the first step is adjustable for the voltage gain, whereas the second step is an amplifier with fixed gain. For both steps, the proportional amplification of displacement was used as control law.
The first step is a non-inverting op-amp. With different excitation amplitudes, the induced voltages on the sensor are different. By adjusting the gain of the first step, the voltages fed to the actuator can be adapted, so that the voltages imposed on the actuator can be controlled in a designed range. Moreover, with the adjustable gain, the influence of different input voltages to the damping performance was investigated. Three gain levels were applied: 2, 6 and 12. The second amplification step is ACX 1224 with a fixed gain of 20 and a voltage output limit of 200 V. These two amplifiers are connected serially. The total gain of the voltage signal is the product of these two separate gains.

<table>
<thead>
<tr>
<th>No.</th>
<th>specimen</th>
<th>total gain [-]</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>(A)CLD</td>
<td>without active control</td>
</tr>
<tr>
<td>2</td>
<td>ACLD 40</td>
<td>40</td>
</tr>
<tr>
<td>3</td>
<td>ACLD 120</td>
<td>120</td>
</tr>
<tr>
<td>4</td>
<td>ACLD 240</td>
<td>240</td>
</tr>
</tbody>
</table>

Table 6.2: ACLD specimens

The applied specimens of ACLD are listed in table 6.2. The first specimen (A)CLD uses exactly the same configuration as the other ACLD samples. Only it was measured without any active control (open loop). The piezo actuator works in this case merely as a passive constraining layer. The other three specimens employ minimal, middle and max gain for the voltage amplification (closed loop).

The plate was bonded to the frame by a thin epoxy layer. Also the piezo sensor patch was bonded to the plate by epoxy. The actuator, VEM and sensor were bonded with each other merely by the butyl rubber without any adhesives like mentioned in the section of ID.

6.2 Results

ACLD is an improved variant of CLD. The dimensions of the piezo patches are normally identical to the VEM. i.e., its configuration and characteristics as well are similar to CLD. Only the efficiency is raised in contrast to CLD due to the active actuating of the VEM. The responses of the ACLD specimens are indicated in figure 6.1.

An influential aspect for the results of ACLD is that with the configuration in this thesis, the first mode can not be damped with the best performance, since
the strain energy is not maximal at the midpoint (reaching about only 70% of the maximum). Thus the conclusion at low frequencies, especially considering the first mode, must be limited for the specific case. Apart from the first mode, the position of the ACLD patch was chosen appropriately.

Firstly, we observe the vibration behaviors of the specimens under non-resonant frequencies. In the low frequency range (10-150 Hz), all FRF curves of No.1-4 overlap with each other. The open/close loop control have barely any influence to their vibration behavior. It reveals that at low frequencies, where the modal wave length of the structure is considerably longer than the length of the CLD segment, no constraining effect is exerted on the VEM layer and the CLD/ACLD treatment has a poor efficiency, if the damper is not positioned at the locations of high strain energy. Systematical analysis and experimental results about this effect can be also found in [95, 22, 24]. Upon 150 Hz, the wave lengths of the structure get shorter. The ACLD becomes more effective. The vibration amplitudes of the passive (A)CLD specimens are globally higher than the close-loop controlled ACLD in the investigated frequency range. Due to the limited frequency range in our test, the wave lengths of the structure are still longer than the size of the piezo actuator. In addition, the sensor and the actuator were located at the nodal position for many of the modes. Consequently, the instability of ACLD due to the spillover effect [96] could not be observed from the
results and the close-loop ACLD performs overall better than open-loop (A)CLD.

Next, the responses of resonant frequencies will be observed. The FRF of the damped modes (1, 4 and 9) are shown in figure 6.2. On the ACLD 240 sample, the amplitudes are reduced by 5.2 \( \text{dB} \) (mode 1), 4.2 \( \text{dB} \) (mode 4) and 2 \( \text{dB} \) (mode 9) in comparison to the (A)CLD sample respectively. Although the amplitudes are mitigated, the damping efficiency of ACLD remains moderate for the surveyed samples. At mode 1, the loss factor increases slightly with the amplified voltage in spite of the perceivable amplitude mitigation. The improvement of the loss factor with increasing amplification at mode 4 is pronounced. At mode 9, the differences are not marked. In order to understand these effects, the mode shapes and the position of the actuators at these modes are shown in figure 6.3.

Figure 6.2: FRF of damped modes (1, 4 and 9) with ACLD

At mode 1, the dimension of the ACLD damper is too small to induce sufficient constraining effect in the VEM. Also the strain energy at the middle is moderate compared to the edge region, which leads to the inefficiency at this mode. At mode 4, the piezo locates between two consecutive nodes. The half wavelength between these two nodes is close to the piezo patch’s longitudinal dimension. The piezo sensor is strongly bended and outputs a great voltage. The wavelength is in the adequate range for the damping, which results in the high loss factor at this mode. The deflection of the plate at mode 9 is in the same
order as mode 4. However, the piezo sensor experiences a small curvature at this mode, because the wave form is perpendicular to that of mode 4 and therefore is not optimal to actuate ACLD damping elements. The minor deflections of the sensor and VEM cause the resultant low damping. It can be seen that in the damped modes, actually only some of them can be effectively damped with one ACLD configuration.

The responses at the undamped and global modes are shown in figure 6.4. At mode 2, due to the low frequency, all curves overlap with each other. Above mode 2, all other modes including the global mode have almost the same behavior. The curves of the ACLD overlap at the resonances. The amplitudes of the passive (A)CLD are higher than the ACLD. Except the difference between (A)CLD and ACLD, no improvements can be seen by enhancing the active voltage. The calculated loss factors are listed in table 6.4. There is no significant difference between the undamped and ACLD specimens. As expected, ACLD as a sectional damper doesn’t contribute noticeably to the damping of the undamped and global modes.

### Table 6.3: results of ACLD at damped modes

<table>
<thead>
<tr>
<th>damper</th>
<th>mode 1</th>
<th>mode 4</th>
<th>mode 8/9</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>amp. [dB]</td>
<td>$\eta_c [10^{-2}]$</td>
<td>amp. [dB]</td>
</tr>
<tr>
<td>(A)CLD</td>
<td>74.9</td>
<td>1.71</td>
<td>52.1</td>
</tr>
<tr>
<td>ACLD gain 40</td>
<td>74.1</td>
<td>1.65</td>
<td>48.8</td>
</tr>
<tr>
<td>ACLD gain 120</td>
<td>71.4</td>
<td>2.05</td>
<td>47.9</td>
</tr>
<tr>
<td>ACLD gain 240</td>
<td>69.6</td>
<td>2.50</td>
<td>47.9</td>
</tr>
</tbody>
</table>

Figure 6.3: strain ratio of the piezo elements for mode 1, 4 and 9
Figure 6.4: FRF of undamped and global modes with ACLD
Table 6.4: loss factors of ACLD at undamped and global modes

<table>
<thead>
<tr>
<th>specimen</th>
<th>mode 2</th>
<th>mode 3</th>
<th>mode 5</th>
<th>mode 6</th>
<th>mode 7</th>
<th>mode 10</th>
</tr>
</thead>
<tbody>
<tr>
<td>undamped</td>
<td>0.89</td>
<td>1.66</td>
<td>0.85</td>
<td>0.84</td>
<td>1.16</td>
<td>0.13</td>
</tr>
<tr>
<td>(A)CLD</td>
<td>1.46</td>
<td>1.06</td>
<td>0.99</td>
<td>0.99</td>
<td>1.20</td>
<td>1.02</td>
</tr>
<tr>
<td>ACLD gain 40</td>
<td>1.45</td>
<td>1.01</td>
<td>1.05</td>
<td>1.04</td>
<td>1.17</td>
<td>1.08</td>
</tr>
<tr>
<td>ACLD gain 120</td>
<td>1.52</td>
<td>1.00</td>
<td>1.07</td>
<td>1.05</td>
<td>1.12</td>
<td>1.14</td>
</tr>
<tr>
<td>ACLD gain 240</td>
<td>1.48</td>
<td>0.95</td>
<td>1.12</td>
<td>1.07</td>
<td>1.11</td>
<td>1.11</td>
</tr>
</tbody>
</table>

### 6.3 Conclusion

The characteristics of ACLD can be deduced from the aforementioned effects and some experimental experience:

- ACLD is generally ineffective in low frequency range, where the structure modal wavelength is bigger than the sensor dimension by far. At resonances the improvement through close-loop is limited whilst at non-resonant frequencies no difference can be determined. However, the efficiency should be higher than measured in this paper since the position of ACLD elements was sub-optimal at mode 1.

- in high frequency range with short modal wavelengths of the structure, ACLD show globally better performance. At resonances, significant damping can be observed.

- the performance of ACLD is sensitive to the location. Both high relative strain in the sensor and the adequate structure wavelength in the longitudinal direction of the sensor / actuator are necessary for a good damping efficiency.

- the bending stiffness of the panel should be higher than the piezo element so that the sensor undertakes sufficient strain.

- for the optimization of CLD or ACLD, sophisticated consideration should be made by concerning different passive and active parameters. E.g. the shear parameter $g^*$ [31]

- ACLD should be designed for addressing the damping of some specific modes. Like in our test case, few damped modes can be mitigated whilst undamped and global modes are barely influenced by the damper.
Chapter 7

Interface Damping Treatment

This chapter encompasses the experimental investigation of ID in the second project phase and the parametric study by means of a FE model developed in the third project phase. This chapter can be divided mainly into three parts. First of all, the experiments with ID are shown in section 7.3. Next a FE model of the ID specimen is built in Ansys® and validated with the experiments in section 7.4. In the last step in section 7.5, a parametric study of ID’s performance with respect to the material properties of VEM and interface geometry is conducted.

7.1 Introduction

As one of the essential aims of this thesis, the possibility of introducing damping into the system through its interface must be emphasized investigated. By attaching VEM as the adhesive joint between the panel and its peripheral components, a soft and flexible interface is formed instead of bonding the panel directly with stiff adhesive to other structures. This compliant joint influences the vibration transmitted from the exciter (auto body) to the receiver (auto panel). Two aspects can be adjusted at the interface to improve the vibration behavior of the receiver: isolation and damping, which are afforded by the VEM joint concurrently. On one hand, for the isolation, a common method is to set some isolators with low stiffness at the interface. Principally, springs are used. Due to the distinct stiffness difference between VEM and the conjunction pair, VEM components can also be used as isolator. Some monographs have discussed the functionality of isolation devices for rigid machines or buildings [97, 43, 98]. But the research on the receiver behaviors of flexible structures like panels is scarce. On the other hand, for the damping, VEM sheets can be mounted on the bolted,
riveted or spot-welded structures to augment damping into the system [97, 32]. Also a variety of contributions investigated the dynamic behaviors of the adherends and adhesives by employing pure VEM at the interface. In the papers [99, 100, 101], the influence of the interface geometry / material properties on the system natural frequency and mode shape on a cantilever beam with a single lap joint was studied. They don’t describe the damping effect of the interface, however. In other papers, the loss factors or the frequency response function (FRF) of the system were investigated. Prucz [102] studied the damping effect of a rhombic interface joint using VEM. Apalak [103] determined experimentally the loss factors of an adhesively bonded double containment cantilever beam based on the interface / beam geometry variation. Kaya [104] investigated the receptance at the single lap joint by introducing a system modal loss factor. Vaziri [105] achieved a comprehensive parametric study of the interface joint on the system dynamic response. Rao [106] developed a closed analytic model of the lap joint on a pin-pin beam. Nevertheless, most of the aforementioned literature analyzed the behavior of a cantilever beam system with a lap joint. The more complex damping behavior of the interface joint on a plate structure was not studied. Also, these papers concentrated mostly on the dynamic behavior of the interface, instead of enunciating the global vibration and damping on the exciter or receiver systematically. Furthermore, the realistic boundary and application for vehicles were not implemented (auto body as exciter and panel as receiver). Therefore, some further investigations concerning these aspects are necessary for a better understanding of interface damping’s characteristics.

The result in the first project phase of screening implies already the viability of ID. In this chapter, ID is thus further investigated through some detailed surveys regarding the above mentioned points.

7.2 Properties of the applied viscoelastic material

For all investigated ID in this chapter, a commercial butyl rubber (VEM) supplied by an industrial partner was applied. The butyl is a typical material used for CLD, whereas its exact material properties were originally unknown. In order to have an accurate comparison between experiment and simulation, it is crucial to have the precise material properties of this butyl rubber.

Based on some preliminary results in Obest beam tests [29], the butyl is modeled according to the method with five steps stated in section 2.5.3. Some discrete derived $E_{mat,exp}'$ and $\eta_{mat,exp}$ points acquired from the third step are
plotted in the Wicket plot of figure 7.1.

Figure 7.1: Wicket plot of the fitted material data \( (E'_\text{mat,exp}, \eta_{\text{mat,exp}}) \) at different temperatures

Since these points do not form an unique inverted \( U \) curve, the 1\(^{st}\) order fractional derivative model (see section 2.5.2) is not precise enough to describe the behavior of the VEM. The deviation between the fitted material curve and the discrete values of \( E'_\text{mat,exp} \) and \( \eta_{\text{mat,exp}} \) is remarkable in the 1\(^{st}\) order model. Consequently, a 2\(^{nd}\) order fractional derivative model was used to diminish apparent deviations between the data fitted under different temperatures. Here again, the complex modulus can be described in the 2\(^{nd}\) model as:

\[
E^* = \frac{a_1 + b_1 \cdot (i\omega\alpha(T))^\beta + b_2 \cdot (i\omega\alpha(T))^{2\beta}}{1 + c_1 \cdot (i\omega\alpha(T))^\beta + c_2 \cdot (i\omega\alpha(T))^{2\beta}} \quad (7.1)
\]

The fitted values of all the coefficients are listed in table 7.1. The general description of the butyl rubber’s properties using the 2\(^{nd}\) order fractional model is plotted in figure 7.2. The data utilized in the simulation under the temperature of experiments were derived from the general model and are shown in figure 7.3.
Chapter 7. Interface Damping Treatment

<p>| | |</p>
<table>
<thead>
<tr>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>$a_1$</td>
<td>1.60e0</td>
</tr>
<tr>
<td>$b_1$</td>
<td>4.19e-4</td>
</tr>
<tr>
<td>$b_2$</td>
<td>2.15e-1</td>
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<tr>
<td>$c_1$</td>
<td>8.88e-3</td>
</tr>
<tr>
<td>$c_2$</td>
<td>1.00e-6</td>
</tr>
<tr>
<td>$\beta$</td>
<td>7.75e-1</td>
</tr>
<tr>
<td>$T_A$</td>
<td>4.29e3</td>
</tr>
<tr>
<td>$T_0$</td>
<td>293.15 K</td>
</tr>
</tbody>
</table>

Table 7.1: values of the coefficients in the $2^{nd}$ order fractional model

![Figure 7.2: $2^{nd}$ order fractional derivative model of VEM](image)

Figure 7.2: $2^{nd}$ order fractional derivative model of VEM

7.3 Experiment

The results presented in this section were attained in the second project phase. Further information of the test rig and base sample can be seen in section 3.3.

7.3.1 Damper configurations

Limited by the production process, the VEM is only available in the thickness of 1.4-1.5 mm. In the experiment, an undamped sample and two ID samples as shown in figure 7.4 were tested to investigate the effect of interface thickness. In the ID samples, one is bonded with one layer butyl, and the other is bonded with two layers, which are overlapped across the thickness direction. Partial gaps between these two butyl layers could not be completely circumvented. The
7.3. Experiment

Figure 7.3: storage modulus and loss factor of the VEM at experiment temperature (24 °C)

applied butyl rubber is optimized for automotive application and has superior adhesion properties. In order to avoid any unknown effects caused by gluing, the surfaces of the frame and the plate were roughed by sandpaper and then directly bonded together by the butyl rubber without additional adhesive. The specimens were measured under ambient temperature at 24 °C. The overlapped width of the VEM between the plate and the frame is 20 mm on each side and the outer dimension of the overlapped interface is the same as the plate (440 × 340 mm).

Figure 7.4: measured ID specimens

7.3.2 Results

With the vibration imposed on the frame, remarkable bending or torsion deformation at the flexible interface is induced in the compliant VEM, which causes damping. From the FRF in figure 7.5 it can be observed that the resonance peak at each mode of the sample with single layer ID is markedly reduced with 10-20 dB compared to the undamped configuration. By applying 2 layers of VEM, the
vibration attenuation is even stronger (further 3-4 \( dB \) reduction compared to the single VEM layer). The loss factors of ID are listed in table 7.2. On the sample with single layer VEM, all modes are stably damped. The difference between the loss factors at different frequencies is not tremendous. The values between 2.2-3 \([10^{-2}]\) reveal that the amplitude reduction comes not only from the mass effect, but also from the damping contribution. The characteristic of the ID is that the damping is globally effective for every mode through the whole frequency range of interest. Modes with small vibration amplitudes are even eliminated in the FRF by ID (like the mode at 316 \( Hz \) on the undamped specimen in figure 7.5). Concerning this aspect, ID performs well within the whole frequency range of interest in contrast to sectional treatments that damp only some modes or areas partially.

![Figure 7.5: complete FRF of ID](image)

<table>
<thead>
<tr>
<th>specimen</th>
<th>( \eta_c \ [10^{-2}] )</th>
<th>mode 1</th>
<th>mode 2</th>
<th>mode 4</th>
<th>mode 6</th>
<th>mode 8/9</th>
<th>mode 10</th>
</tr>
</thead>
<tbody>
<tr>
<td>undamped</td>
<td></td>
<td>0.73</td>
<td>0.89</td>
<td>0.48</td>
<td>0.84</td>
<td>0.65</td>
<td>0.13</td>
</tr>
<tr>
<td>ID 1 layer</td>
<td></td>
<td>2.96</td>
<td>2.44</td>
<td>2.27</td>
<td>3.05</td>
<td>2.69</td>
<td>2.42</td>
</tr>
<tr>
<td>ID 2 layers</td>
<td></td>
<td>3.89</td>
<td>3.70</td>
<td>3.49</td>
<td>6.42</td>
<td>3.40</td>
<td>3.70</td>
</tr>
</tbody>
</table>

Table 7.2: loss factors of ID for all modes
Another notable point is that at the global mode (mode 6 at about 240 Hz in figure 7.5 and table 7.2, ID has the highest loss factor among all the modes. The loss factor reaches 3.05 and 6.42 \(10^{-2}\) with 1 or 2 VEM layers respectively. The intense vibration on the frame at this mode enhances the deformation in the VEM. In automotive applications, many global modes of car bodies appear typically below 150 Hz. At these eigenfrequencies, the normal surface damping system are not effective for the panel structures because typically the deformation within the single panels is marginal. The panel swings with the frame structure in a rigid-like behavior. With ID this type of global modes could be damped.

Briefly, the characteristics of ID observed from the results can be concluded as following:

- ID is an approach which damps every mode in the investigated frequency range. The performance of ID is stable at any frequency with the applied soft VEM.

- ID is the only approach investigated that helps to damp global vibrations from the peripheral structures. Therefore, it is an approach with profound potential for damping automotive panels.

### 7.4 Simulation

According to the results in the experiments, ID is an effective and stable approach for damping panels in a wide frequency range, especially at global modes in low frequency range where normal surface add-on dampers are inefficient in the automotive structures. Thus, this thesis was focused on ID for the further research phase. In this section, the simulation results on ID in the third project phase are presented. With the correlated FE model, the mitigation characteristics of the interface in terms of geometry and material properties are systematically analyzed.

The simulation model is the same subject as the tested specimen in the experiment. The configuration of the FE model is depicted in figure 7.6. It comprises following components like the real specimen:

- aluminum plate
- butyl (VEM) Interface
- honeycomb sandwich frame

Same as the configuration in the experiment, the simulation model is analyzed under free-free condition (nowhere pinned). The excitation point is placed
on the honeycomb frame and locates at an eccentric position in both length and width direction of the frame so that every mode of the plate are excited. On the surface of the 1 mm thick aluminum plate, 25 points from the grid are evaluated.

In order to save the computational expense, the honeycomb core in the frame is simplified and modeled with an equivalent continuum core, i.e., solid elements. A comparative study on the different theories of the equivalent continuum properties is performed in [107]. Among the introduced theories, the Ashby theory [108] is the unique one which provides a comprehensive description of the substitute continuum’s properties with a generalized hexagon cell geometry. Therefore, it was applied for this thesis. The precondition of this theory is that the honeycomb core consists of a single isotropic material, e.g., aluminum. Then the real honeycomb core structure is substituted with continuum elements. The substitute is a virtual orthotropic material possessing nine equivalent elastic constants, which are derived from the cell geometry and material. Therefore, all components in the simulation model are simulated with solid elements except the plate (shell). The FE model is illustrated in figure 7.7. All elements have a maximal length of 10 mm.

Also here the damping effects in the simulation were evaluated with the similar processes as in the experiments in the following three steps:

- building of the FRF by using a unit force 1 N in harmonic analysis
- modal curve fitting of the transfer mobility FRF out of 25 points in figure 7.6 to a SDOF system response
- evaluation of the damping with the fitted SDOF-curve

### 7.4.1 Simulation of the undamped specimen

For a better understanding of the interface damper’s behavior, the undamped specimen was investigated preliminarily in the simulation by bonding the plate with a thin stiff epoxy layer (0.5 mm thick) instead of VEM.

A harmonic analysis was done in Ansys® to detect the amplitude response of these modes. According to the mode shapes in figure 3.7, their mobility FRF can be seen in figure 7.8. In table 7.3 it can be seen, most of the frequencies values in simulation are in accordance with the experiments (±5%). Only the global mode has a deviation of about 10%. It will be stated in the next section.
Figure 7.6: arrangement of the FE model with: (a) cross section view, (b) configuration
Chapter 7. Interface Damping Treatment

Figure 7.7: FE model for the ID panel

![FE model for the ID panel](image)

Figure 7.8: mobility FRF of the undamped specimen in simulation

![Mobility FRF of the undamped specimen in simulation](image)

<table>
<thead>
<tr>
<th>mode No.</th>
<th>1</th>
<th>2</th>
<th>3</th>
<th>4</th>
<th>5</th>
<th>6</th>
<th>7</th>
<th>8/9</th>
<th>10</th>
</tr>
</thead>
<tbody>
<tr>
<td>FEM</td>
<td>80</td>
<td>132</td>
<td>185</td>
<td>220</td>
<td>232</td>
<td>274</td>
<td>317</td>
<td>348</td>
<td>395</td>
</tr>
<tr>
<td>Exp.</td>
<td>85</td>
<td>136</td>
<td>184</td>
<td>223</td>
<td>229</td>
<td>247</td>
<td>316</td>
<td>341</td>
<td>389</td>
</tr>
<tr>
<td>Deviation [%]</td>
<td>5.7%</td>
<td>3.1%</td>
<td>-0.4%</td>
<td>1.3%</td>
<td>-1.3%</td>
<td>-9.7%</td>
<td>-0.2%</td>
<td>-2.1%</td>
<td>-1.6%</td>
</tr>
</tbody>
</table>

Table 7.3: comparison of the mode frequencies in experiment and simulation

7.4.2 Correlation of simulation and experiment

The specimen with interface damper was simulated with harmonic analysis as well. In No.3, the two butyl layers are physically not entirely bonded. This separating is difficult to correlate, since most joints are simulated with ideal
bonding condition. Thus, only the specimen No.2 in figure 7.4 with the single VEM layer was taken for the correlation. Also the plate is built with shell elements, whilst other components are all simulated with solid elements.

The experimental FRF of specimen No.2 is compared to the simulation in figure 7.9. Both curves are mostly coincident. Until 200 \(Hz\) the curves overlap almost with each other. At high frequencies, due to the imperfect bonding of butyl in the experiment, the experimental FRF locates on the left side of the simulation, which implies the stiffness of the system in the experiment is not as high as the simulation that assumes every component is ideally conjunct to the others. Another possibility is that between about 200 and 900 \(Hz\), there are less data points of \(E'_{mat,exp}\) and \(\eta_{mat,exp}\) generated from the polynomial function \(f_1\) and \(f_2\) than below 200 \(Hz\) for the modeling (curve-fitting procedure in section 2.5.3), thus the precision of the fitted material model decreases slightly and the model has a insignificant deviation from the reality.

![Figure 7.9: comparison of ID between experiment and simulation](image)

The parameters of the eigenmodes with dominant amplitudes in figure 7.9 are listed in table 7.4. The deviation between experiment and simulation is small at all plate modes. The simulation represents the reality quite well. However, at the global mode (mode 6), some differences can be observed like the correlation on the undamped specimen (see table 7.3). The damping in the experiment is higher and the eigenfrequency is shifted to the left side of the simulation. This
can be attributed to the modeling of the frame: The bonding layer between the honeycomb core and face sheet of the sandwich was not encompassed in the simulation, since the influence of this layer could not be estimated precisely. The inhomogeneous distribution of the glue hampered the prediction of its contribution to the total system mass / stiffness matrix. The additional mass of the glue layer pulled resultantly the eigenfrequency of this mode leftwards. However, this mode originating from the frame still reflects the reality with maximal 30% loss factor difference, whilst all other plate modes can be correlated accurately.

<table>
<thead>
<tr>
<th></th>
<th>mode 1</th>
<th>mode 2</th>
<th>mode 4</th>
<th>mode 6</th>
<th>mode 8/9</th>
<th>mode 10</th>
</tr>
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<tr>
<td><strong>experiment</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
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<tr>
<td>frequency [Hz]</td>
<td>76</td>
<td>129</td>
<td>210</td>
<td>241</td>
<td>323</td>
<td>371</td>
</tr>
<tr>
<td>amplitude [dB]</td>
<td>68.5</td>
<td>60.8</td>
<td>52.3</td>
<td>41.1</td>
<td>53.2</td>
<td>46.6</td>
</tr>
<tr>
<td>loss factor [$10^{-2}$]</td>
<td>2.96</td>
<td>2.44</td>
<td>2.27</td>
<td>3.05</td>
<td>2.69</td>
<td>2.42</td>
</tr>
<tr>
<td><strong>simulation</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>frequency [Hz]</td>
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<td>126</td>
<td>212</td>
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<td>378</td>
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<td>61.5</td>
<td>52.6</td>
<td>38.4</td>
<td>52.8</td>
<td>44.2</td>
</tr>
<tr>
<td>loss factor [$10^{-2}$]</td>
<td>2.53</td>
<td>2.26</td>
<td>2.20</td>
<td>2.02</td>
<td>2.69</td>
<td>2.72</td>
</tr>
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<td><strong>deviation</strong></td>
<td></td>
<td></td>
<td></td>
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</tr>
<tr>
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<td>-0.7%</td>
<td>-14.0%</td>
<td>-2.9%</td>
<td>-1.7%</td>
</tr>
<tr>
<td>amplitude [%]</td>
<td>-2.0%</td>
<td>-1.0%</td>
<td>-0.6%</td>
<td>7.1%</td>
<td>0.7%</td>
<td>5.4%</td>
</tr>
<tr>
<td>loss factor [%]</td>
<td>14.7%</td>
<td>7.5%</td>
<td>2.9%</td>
<td>33.8%</td>
<td>-0.1%</td>
<td>-10.9%</td>
</tr>
</tbody>
</table>

Table 7.4: loss factors of ID in experiment and simulation

By taking the compromise of saving computational expense, we deem that this simulation model provides in general acceptable precision for the investigation of the vibration on the plate. It was further utilized for the following parametric study to survey the characteristics of ID.

### 7.5 Parametric study

In this parametric study, the influence of ID on both system amplitude response and structural modal loss factor was investigated. The study consists of two parts:
the investigation of the influence of VEM properties and of the interface geometry. In the first part of the study, the material properties of the interface material were taken as virtual frequency independent values, whilst in the latter part the properties were taken from the real frequency dependent material characteristics, i.e., butyl’s behavior under ambient temperature at 24 °C. Other parameters are the same for both parts as listed in table 7.5.

<table>
<thead>
<tr>
<th></th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Storage modulus of aluminum</td>
<td>69e9 [Pa]</td>
</tr>
<tr>
<td>Material loss factor of aluminum</td>
<td>0.005 [-]</td>
</tr>
<tr>
<td>Poisson ratio of aluminum</td>
<td>0.33 [-]</td>
</tr>
<tr>
<td>Density of aluminum</td>
<td>2700 [kgm$^{-3}$]</td>
</tr>
<tr>
<td>Poisson ratio of butyl (interface material)</td>
<td>0.48 [-]</td>
</tr>
<tr>
<td>Density of butyl (interface material)</td>
<td>1556 [kgm$^{-3}$]</td>
</tr>
</tbody>
</table>

Table 7.5: material constants in the simulation

### 7.5.1 Interface material properties

In this section, the storage modulus and loss factor of the butyl (interface material / VEM) were varied with frequency independent values as retained in table 7.6. The storage modulus was assumed from 1 MPa, which is lower than the minimum modulus of the butyl under ambient temperature, to 4.5 GPa that corresponds to the value of a stiff epoxy. The value of the material loss factor was taken from an almost elastic object to the unrealistic high value of 5 (The maximum value of butyl under ambient temperature is 1.11). The harmonic analysis was done 10 × 11 times in different combinations with the values in table 7.6. Due to the great damping values ($\eta_m = 2$ or 5) in the investigation range, the modes with small modal amplitudes were eliminated and could not be evaluated with FPBW method anymore. Therefore, only the modes with dominant amplitudes were investigated. They are the plate mode 1,2,4,9,10 and global mode 6.

The structural loss factors $\eta_c$ were evaluated in figure 7.10. The values of the No.$\eta_m$ and No.$E'$ in the graphics are the same as listed in table 7.6. With the increase of the VEM storage modulus through the whole range, $\eta_c$ is declined, since less deformation exists in the VEM, resulting in poor damping. $\eta_c$ rises with the augmentation of the material loss factor $\eta_m$ until the realistic value of $\eta_m = 1.5$. With even higher virtual material loss factors, the contribution of the damping to the stiffness matrix becomes greater in the simulation (refers to the construction of damping matrix in Ansys [109]), so that it leads again to the same effect of increasing the storage modulus of the VEM. Thus, the damping is weakened under extreme high material damping values in
the simulation. Therefore it can be concluded, a soft VEM interface with high material loss value is beneficial for the structural damping in the real application.

<table>
<thead>
<tr>
<th>No.</th>
<th>1</th>
<th>2</th>
<th>3</th>
<th>4</th>
<th>5</th>
<th>6</th>
<th>7</th>
<th>8</th>
<th>9</th>
<th>10</th>
<th>11</th>
</tr>
</thead>
<tbody>
<tr>
<td>storage modulus (E') [Pa]</td>
<td>1.00 e6</td>
<td>2.11 e6</td>
<td>4.47 e6</td>
<td>9.46 e6</td>
<td>2.00 e7</td>
<td>5.00 e7</td>
<td>1.00 e8</td>
<td>5.00 e8</td>
<td>1.00 e9</td>
<td>4.50 e9</td>
<td>-</td>
</tr>
<tr>
<td>loss factor (\eta_m) [-]</td>
<td>0.005</td>
<td>0.01</td>
<td>0.02</td>
<td>0.05</td>
<td>0.1</td>
<td>0.2</td>
<td>0.5</td>
<td>1</td>
<td>1.5</td>
<td>2</td>
<td>5</td>
</tr>
</tbody>
</table>

Table 7.6: investigation values in the study of VEM properties

Regarding the structural loss factors at different modes, the shapes and \(\eta_c\) values of the surfaces are quite similar, which is in conformity with the experimental phenomena. This indicates again the damping of ID is quite stable at any frequency. The diagram of the global mode can only be plotted partially, because the resonance peaks with small amplitudes were eliminated by great interface damping values \(\eta_m\) at some positions of the abscissas. The loss factors at these positions cannot be evaluated with HPBW-method and they are left blank in the plot. Principally the shape of the surface (damping distribution in figure 7.10) at mode 6 is the same as the other modes. Comparing a same configuration with certain values of \(\eta_c\) and \(E'\) between different modes in the plots, the numerical values of the global mode are only ca. 1% lower than at other modes. Thus, in ideal situation, the damping at global mode is slightly lower than at plate modes. However, as the experimental result shown in the previous chapter, in the practical application, the local debonding of VEM enhanced probably the damping and shifted its value even a little bit higher than at usual plate modes.

The plots for the amplitude of the mobility FRF are presented in figure 7.11. In general, if the material loss factor is increased, the resonance amplitude will be reduced. However, the effect of the unrealistic great damping \((\eta_m = 5)\) arouses again sectionally the virtual stiffening of the VEM. In consequence, the amplitudes are raised at some points on the edge of the surface.

By increasing the storage modulus, inverse variation tendencies of the system amplitudes are perceivable under different loss factor ranges. At low loss factors until \(\eta_m = 0.02\), the increase of VEM storage modulus reduces the amplitude on plate slightly. Due to the negligible damping in this range, the stiffened interface has a dominant contribution to the modification of the whole system’s stiffness matrix, which results in the amplitude reduction. At high loss factors above
Figure 7.10: results of the VEM property study: structural loss factors of plates
Figure 7.11: results of the VEM property study: vibration amplitudes of plates
0.05 the increasing VEM storage modulus induces the amplitude augmentation, because the deformation of the VEM is smaller and consequently its damping is lower as well.

By altering the VEM properties, the amplitudes will be changed with similar values at any mode. The plot of the global mode is still shown partially. Also similar tendencies can be observed like the other modes. It indicates that the influence of the VEM property affects the amplitude variation of the system also steadily.

Briefly it can be conducted from these plots that a soft interface material with high material loss factor is suitable for achieving high structural damping.

### 7.5.2 Interface geometry

The influence of the interface geometry was investigated by varying the interface thickness $t_{VEM}$, width $b_{VEM}$ and the free vibrating area $b_{hole} \times l_{hole}$ of the plate (refers to figure 7.6 (a)). The values of the investigated parameters can be extracted from table 7.7. While these parameters were changed, other dimensions remained unchanged as the standard specimen No.2 (figure 7.4) in the experiment ($t_{VEM} = 1.5 \, mm, b_{VEM} = 20 \, mm, b_{hole} \times l_{hole} = 400 \times 300 \, mm$).

<table>
<thead>
<tr>
<th>Variation items</th>
<th>Variation values</th>
</tr>
</thead>
<tbody>
<tr>
<td>Interface thickness $t_{VEM}$ [mm]</td>
<td>0.5, 1, 1.5, 3, 4.5, 7, 10</td>
</tr>
<tr>
<td>Interface width $b_{VEM}$ [mm]</td>
<td>10, 20, 30, 40</td>
</tr>
<tr>
<td></td>
<td>$200 \times 150 , mm$ [0.25 times]</td>
</tr>
<tr>
<td>Size of free vibrating area $b_{hole} \times l_{hole}$ [times]</td>
<td>$400 \times 300 , mm$ [1 times]</td>
</tr>
<tr>
<td></td>
<td>$800 \times 600 , mm$ [4 times]</td>
</tr>
</tbody>
</table>

Table 7.7: variation items of the geometry study

The FRF of different interface thickness are plotted in figure 7.12. The responses of the plate modes and the global mode must be distinguished. For the plate modes, the shape and amplitudes of all resonance peaks don’t show significant differences. This indicates that by employing more VEM in the thickness direction, the vibration cannot be mitigated at the eigenmodes of the panel. For the global mode, with the increase of the VEM thickness, the peak is attenuated remarkably, i.e., for the mitigation of the panel structures in the vehicles, thickening the VEM is advantageous for damping the global modes.
Figure 7.12: mobility FRF of the geometry study (thickness variation)
from the auto body.

The modal structural loss factors of the different VEM thickness are evaluated in figure 7.13. For the plate modes, the loss factors exhibit pronounced similarity at different thicknesses, where the deviation of the loss factors is within 1% at almost all modes. The values at mode 3 are overall higher than at other plate modes due to its mode shape: The longitudinal half wave length at this mode goes through the length direction of the plate which incurs large deformation in the VEM. At the global mode, the loss factor augments proportional with the thickening of VEM. Hence, the ambivalent damping-weight performance must be compromised for the thickness design regarding the damping at global mode.

![Figure 7.13: structural loss factors of the geometry study (thickness variation)](image)

The responses of the altered interface width are illustrated in figure 7.14. At $b_{VEM} = 10 \text{ mm}$, the resonance amplitudes are much lower than other widths. The flexible interface in this case can be well sheared during the vibration. With the broadening of the VEM, the amplitudes increase strongly above 10 mm, since the relative stiff interface could hardly be deformed and thus little energy were dissipated. Above certain limit, the influence of a broad interface becomes minor again, as the small differences shown between the width 20-40 mm.
Chapter 7. Interface Damping Treatment

The extracted loss factors are shown in figure 7.15. The loss factors at 10 mm are significantly higher than the loss factors at broader interfaces, whereas a potential bonding failure should be avoided from the weak conjunction. In the range of 20-40 mm, the values are similar. At mode 3, the loss factors are still higher than at the other modes. With a low VEM thickness, the vibration at the global mode could barely be improved by varying the width. The compromise between mechanical requirements and damping must be undertaken for the width design with respect to the plate modes.

The FRF of three plates with half / normal / doubled plate length and width are presented in figure 7.16. The high stiffness of the small plate causes that only one mode locates within 400 Hz. On the big plate the resonance peaks become very sharp, while the peak of the small plate looks flat. Since additional specific modes appear on different plate sizes with the coupled sandwich frame, only the first three dominant modes (mode 1,2 and 4), which could still be recognized with their mode shapes, were compared with each other. For the small plate, the harmonic analysis was carried out until 850 Hz.

Their loss factors can be seen in figure 7.17. All three modes show identical tendency, in which the loss factors on a smaller plate are generally higher than on a bigger one. They decrease almost proportional with the increasing dimensions of the vibrating panel. On a large panel, the damping at the interface can barely
7.5. Parametric study

Figure 7.15: structural loss factors of the geometry study (width variation)

Figure 7.16: mobility FRF of the geometry study (plate size variation)
affect the vibration in the mid area of the plate. The application of ID is thus not adequate for the damping of a panel with excessive huge dimensions or extreme low bending stiffness.

<table>
<thead>
<tr>
<th>Plate Area</th>
<th>Structural Loss Factor [%]</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.25 times area</td>
<td>4</td>
</tr>
<tr>
<td>1 times area</td>
<td>2.5</td>
</tr>
<tr>
<td>4 times area</td>
<td>2</td>
</tr>
</tbody>
</table>

Figure 7.17: Structural loss factors of the geometry study (plate size variation)

From all these phenomena observed above, we can conclude that the interface geometry should stay possibly narrow and thick for the damping of all modes. With such geometry, the behavior of the VEM is optimal for the shear stress generation at the interface, resulting in damping. However, some demands should still be satisfied concerning the mechanical and weight performance. Concurrently, the ID should be applied on a panel structure within certain dimension limit, which is well fit to many automotive panels.

### 7.6 Conclusion

In this chapter, interface damping (ID) is studied both with experiments and in simulation. The results present, ID is not only suitable for damping every eigen-mode of the panel structures, but also contributes to the mitigation of global modes, which exist typically at low frequencies, from the peripheral structures. A broad frequency band can thus be well attenuated with ID, covering
the drawback of prevalent add-on dampers. The system FRF of FE-analyses were evaluated and correlated with experiment consistently. Various parameters regarding the material properties of VEM, the geometry of the interface were studied by means of simulation. The results show that a soft VEM with high material loss factor is beneficial. Concurrently, an interface with possibly narrow and thick VEM is advantageous for the damping within a wide frequency range.
Chapter 8

Novel Concepts of Interface Damper

In the previous chapter, the influence of various parameters on a simple rectangular interface was investigated. Some concepts with more complex geometry or configurations, which were envisioned to have better performances than a simple one, are devised in this chapter based on the master thesis of M. Studer [88] and are investigated in simulation. The structural loss factors $\eta_c$ are extracted on each configuration and compared with respect to the applied material / interface weight.

8.1 FE model

In order to reduce the computational expense, a simple mid-scale aluminum beam structure was taken as the investigation object for this chapter. The configuration and geometry of the specimen is illustrated in figure 8.1. This beam specimen has a free vibrating area of $260 \times 60 \times 0.5 \, mm$. It is mounted with various interface damper designs at both ends. Most of the interfaces have a total dimension of $60 \times 20 \times 4 \, mm$ including auxiliary components. Beneath the damper, the peripheral structures (simulated with simple blocks) are fixed. The VEM used in this chapter is the butyl introduced in the previous chapter. For a better overview, all other auxiliary components for the connection between the beam and the fixation were simulated with aluminum, i.e., the whole system is composed of merely butyl and aluminum. The simulations were exerted also under ambient temperature. Unlike the evaluated FRF with 25 points on the plates in the previous chapters, in this study all nodes on the beam were averaged for calculating the RMS mobility-FRF, based on which the structural loss factors were evaluated in table 8.1 and table 8.2.
Chapter 8. Novel Concepts of Interface Damper

The reason for using all nodes is that this beam model is simple enough. The evaluation of the exact structural loss factor with all nodes doesn’t raise the computational expense remarkably, which is not viable with a more complex simulation model. The beam is excited at an eccentric position indicated in figure 8.1, so that all the even bending modes and torsion modes could be excited.

![Figure 8.1: configuration of the interface design](image)

Up to 400 Hz, there are 7 modes existing on the undamped beam. Their mode shapes can be seen in figure 8.2.

![Figure 8.2: mode shapes of the beam](image)

8.2 Interface Damper configurations

Sixteen interfaces are designed as depicted in figure 8.3-8.6. The first eight concepts employed only VEM at the interfaces, in order to investigate the efficiencies of pure VEM in different shapes and distributions. Four further
8.2. Interface Damper configurations

Concepts of VEM interface (1): In figure 8.3, the standard configuration serves as the reference for the comparison to other configurations. Cut 1 has a split in the VEM layer compared to the standard. The cut aims at the augmentation of the shear in the VEM analog the segmented CLD treatment [24]. The shear deformation is minor in the region of the cut 1’s split, whereas in cut 2, the split is located at the position where the deformation is major. Based on the idea of segmenting, in the fraction concept, VEM are arranged in small lumps to raise the shear compliance of the interface.

Concepts of VEM interface (2): In the double fraction in figure 8.4, the 4mm thick VEM lumps are separated into two 2 mm thick layers by an intermediate plate which is intended to establish a two-stage isolation [43] interface for the beam. In the concepts of No.6, 7 and 8, the VEM are glued on curved surfaces. Corrugation 1 has a single curvature with the radius of 12 mm. The bending moment on the beam were expected to be dissipated by the induced shear force in the radial direction at the interface. Corrugation 2 has doubled curvatures with a smaller radius of 10 mm, which was anticipated to increase the shear in the curved surfaces. The fractional corrugation combines the characteristics of fraction and corrugation 2 with an even smaller radius of 8 mm.

Concepts of elastic springs: On the fractional CLD specimen in figure 8.5, two small CLD patches are contained at each conjunction. The beam is connected with a segment VEM layer (VEM 1) and a thin base strip to the fixation area. Another CLD segment (VEM 2 and constrained layer) is supplemented to the surface of the base strip. The construction of a conventional CLD treatment is implemented in the transverse CLD concept. The beam is connected with two strips to a slim CLD beam that is transversely disposed and fixed at the mid-area of the bottom. The corrugated spring specimen aims at shear possibly many VEM in a defined volume. Several VEM sections are integrated into the corrugated volumes of the spring. On the shear beam specimen, the beam is bonded with two strips that directly fixed at their ends. Each strip connects further two vertically disposed VEM patches that are fixed at the other surfaces. This concept was intended to generate shear into the VEM while the strip is bent by the beam.
Figure 8.3: concepts of VEM interface (1)
Figure 8.4: concepts of VEM interface (2)
Figure 8.5: concepts of elastic springs
8.2. Interface Damper configurations

Figure 8.6: concepts of VEM shear elements
Concepts of VEM shear elements: In comb 1, four strips are arranged like comb teeth and bonded with the beam (see figure 8.6). The gaps between the teeth are filled with VEM that are fixed further to the peripheral structure. Comb 2 employs also two comb-like elements that are bonded on their vertical surfaces to the VEM elements. Below the comb, a gap of 1 mm distance is kept to the fixation, so that sufficient shear stress can be induced in the VEM. The concept surround uses four small square cross-shaped aligned VEM elements at each joint. These squares are bonded to a pin, which is a part of the fixation, in the middle and surrounded by a square frame that is bonded to the beam. The 1 mm gap below the interface is kept in the configuration too. It was expected that the vibration in any direction could induce shear in the VEM at the flexible interface. Because the static behavior of the surround concept is not optimal, the VEM squares are permanently loaded with shear stress from the beam weight, thus in the advanced surround concept, a handle is added to the top of the square frame. A low-creeping elastomer \( (E' = 10 \text{ MPa}) \) is bonded between the handle and the pin. In this manner, the static stress can be taken by this elastomer part, while the dynamic loading is concentrated on the VEM.

8.3 Results

The mobility FRF of each concept is not plotted in this chapter. Only the modal structural loss factors of the seven modes are evaluated in table 8.1. Some peaks are eliminated with sufficient damping from the interfaces. "-" in the table are signed for such cases.

Standard configuration has moderate loss factors. The values of cut 1 are in the same range as the standard, which implies no improvement can be achieved by segmenting the VEM layer in the region with minor deformation. Conversely, augmentation of loss factors in comparison to the standard can be perceived in cut 2. The split at the position with dominant deformation is advantageous for the interface damping similar to the effect in CLD treatments. The fraction possesses overall higher efficiencies than the standard in spite of less incorporated VEM. Sufficient shear compliance and significant shear deformation are thus important for the damping. In the double fraction concept, the shear deformation was interrupted into two parts in thickness direction. By hampering the shear, the loss factors were drastically diminished. The beam was excited in the second test case at the position of a fixation (constraint of the fixation was removed in this case) to test the effect of isolation, but the interim plate didn’t bring any other amplitude mitigation. Therefore, the isolation concept is not functional for the interface damping on flexible structures. With the same applied VEM quantity as standard, the damping of corrugation 1 is generally
light and lower than standard, whilst corrugation 2 achieves overall higher loss factors than standard. Since the shapes of both excellent concepts, the fraction and corrugation 2, are combined in the fractional corrugation, its loss factors surpass the other concepts of VEM interface, even mode 5 is totally suppressed.

Table 8.1: modal loss factors of the concepts

<table>
<thead>
<tr>
<th>No.</th>
<th>model</th>
<th>mode 1</th>
<th>mode 2</th>
<th>mode 3</th>
<th>mode 4</th>
<th>mode 5</th>
<th>mode 6</th>
<th>mode 7</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Standard</td>
<td>3.49</td>
<td>2.98</td>
<td>1.85</td>
<td>2.66</td>
<td>3.51</td>
<td>2.72</td>
<td>2.00</td>
</tr>
<tr>
<td>2</td>
<td>Cut 1</td>
<td>3.41</td>
<td>3.04</td>
<td>1.85</td>
<td>2.67</td>
<td>3.34</td>
<td>2.73</td>
<td>2.00</td>
</tr>
<tr>
<td>3</td>
<td>Cut 2</td>
<td>3.89</td>
<td>3.33</td>
<td>2.09</td>
<td>2.92</td>
<td>4.00</td>
<td>2.95</td>
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<tr>
<td>4</td>
<td>Fraction</td>
<td>4.62</td>
<td>4.03</td>
<td>2.13</td>
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<td>4.21</td>
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<td>5</td>
<td>Double fraction</td>
<td>3.01</td>
<td>2.58</td>
<td>1.61</td>
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<td>2.61</td>
<td>2.35</td>
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<td>6</td>
<td>Corrugation 1</td>
<td>2.87</td>
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<td>2.30</td>
<td>2.34</td>
<td>1.39</td>
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<tr>
<td>7</td>
<td>Corrugation 2</td>
<td>3.97</td>
<td>3.00</td>
<td>2.09</td>
<td>3.00</td>
<td>-</td>
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<td>2.20</td>
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<td>2.55</td>
<td>3.96</td>
<td>-</td>
<td>3.51</td>
<td>2.81</td>
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<td>16.06</td>
<td>6.21</td>
<td>21.14</td>
<td>-</td>
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<td>10</td>
<td>Transverse CLD</td>
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<td>5.15</td>
<td>18.15</td>
<td>4.92</td>
<td>7.03</td>
<td>-</td>
<td>8.91</td>
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<td>11</td>
<td>Corrugated spring</td>
<td>8.73</td>
<td>6.10</td>
<td>5.26</td>
<td>10.82</td>
<td>2.68</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>12</td>
<td>Shear beam</td>
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<td>6.05</td>
<td>2.87</td>
<td>3.33</td>
<td>4.79</td>
<td>5.12</td>
<td>2.99</td>
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<tr>
<td>13</td>
<td>Comb 1</td>
<td>0.72</td>
<td>0.79</td>
<td>0.67</td>
<td>0.82</td>
<td>0.84</td>
<td>0.83</td>
<td>0.70</td>
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<tr>
<td>14</td>
<td>Comb 2</td>
<td>19.90</td>
<td>-</td>
<td>1.69</td>
<td>14.62</td>
<td>2.57</td>
<td>41.91</td>
<td>2.25</td>
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<tr>
<td>15</td>
<td>Surround</td>
<td>35.67</td>
<td>-</td>
<td>18.33</td>
<td>22.02</td>
<td>-</td>
<td>-</td>
<td>-</td>
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<tr>
<td>16</td>
<td>Advanced surround</td>
<td>34.65</td>
<td>16.46</td>
<td>-</td>
<td>21.09</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
</tbody>
</table>

The loss factors on the fractional CLD are far beyond any concepts with the inflexible VEM interfaces above. Through the separately exerted simulations with only segment VEM 1 or VEM 2 respectively, it was found that the damping comes dominantly from segment VEM 1. With the longitudinal settled CLD patches in the configuration of fractional CLD, the bending modes of the beam are well damped. In contrast, the transverse CLD has good performance at the beam torsion modes, at which the slim transverse CLD beam is bent, while it is less efficient for the beam bending modes. Thus these two configurations perform inversely at each mode. The corrugated spring operates excellent in high frequencies, wherein two modes were eliminated. On the shear beam specimen, the loss factors are inferior to other concepts based on the spring principles. The direct stiff bonding with the peripheral structure in this case is not beneficial
Table 8.2: weight specific damping efficiencies of the concepts

<table>
<thead>
<tr>
<th>No.</th>
<th>model</th>
<th>Loss factor [10^{-2}]</th>
<th>Loss factor pro VEM weight [kg^{-1}]</th>
<th>Loss factor pro interface weight [kg^{-1}]</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Standard</td>
<td>2.74</td>
<td>0.26</td>
<td>0.22</td>
</tr>
<tr>
<td>2</td>
<td>Cut 1</td>
<td>2.72</td>
<td>0.30</td>
<td>0.24</td>
</tr>
<tr>
<td>3</td>
<td>Cut 2</td>
<td>3.05</td>
<td>0.33</td>
<td>0.27</td>
</tr>
<tr>
<td>4</td>
<td>Fraction</td>
<td>3.47</td>
<td>0.94</td>
<td>0.58</td>
</tr>
<tr>
<td>5</td>
<td>Double fraction</td>
<td>2.32</td>
<td>0.63</td>
<td>0.33</td>
</tr>
<tr>
<td>6</td>
<td>Corrugation 1</td>
<td>2.13</td>
<td>0.20</td>
<td>0.16</td>
</tr>
<tr>
<td>7</td>
<td>Corrugation 2</td>
<td>2.83</td>
<td>0.27</td>
<td>0.22</td>
</tr>
<tr>
<td>8</td>
<td>Fractional corrugation</td>
<td>3.60</td>
<td>1.39</td>
<td>0.78</td>
</tr>
<tr>
<td>9</td>
<td>Fractional CLD</td>
<td>14.24</td>
<td>3.47</td>
<td>2.39</td>
</tr>
<tr>
<td>10</td>
<td>Transverse CLD</td>
<td>8.44</td>
<td>3.77</td>
<td>2.37</td>
</tr>
<tr>
<td>11</td>
<td>Corrugated spring</td>
<td>6.72</td>
<td>3.38</td>
<td>0.79</td>
</tr>
<tr>
<td>12</td>
<td>Shear beam</td>
<td>4.09</td>
<td>9.39</td>
<td>2.05</td>
</tr>
<tr>
<td>13</td>
<td>Comb 1</td>
<td>0.77</td>
<td>0.09</td>
<td>0.04</td>
</tr>
<tr>
<td>14</td>
<td>Comb 2</td>
<td>13.82</td>
<td>5.29</td>
<td>1.38</td>
</tr>
<tr>
<td>15</td>
<td>Surround</td>
<td>25.34</td>
<td>33.95</td>
<td>6.77</td>
</tr>
<tr>
<td>16</td>
<td>Advanced surround</td>
<td>24.07</td>
<td>33.24</td>
<td>5.26</td>
</tr>
</tbody>
</table>

for the damping. However, the loss factors are still higher than the standard concept due to the configuration of shearing VEM.

Comb 1 shows the worst loss factors among all the configurations with the values in the range of an undamped structure. The interface is too stiff with the massive arranged VEM which are totally supported by the fixation and loses its shear compliance. Comb 2 performs excellent for all bending modes, at which adequate shear stress can be generated in the VEM. The loss factors are competitive to the fractional CLD. Surround exhibits the best performance among all concepts. Four modes are completely suppressed. The other three evaluable modes have also the highest loss factors respectively. Therefore, the surround can be a promising design for the interface damper, which can be locally placed with strategic distribution at the interface. The advanced surround improves the static behavior of the interface at the cost of sacrificing slightly the loss factors. The shear deformation in the VEM decline with the compression / ex-
tensional force imposed on the elastomer. The damping is compensated partially.

The results in table 8.1 presents that the best concept with the highest absolute loss factor is the surround. In order to compare these concepts with their relative efficiencies as well, their mean loss factors, weight specific mean loss factors referring to the applied VEM and specific mean loss factors in respect to interface weight are calculated in table 8.2. Briefly, the first eight concepts with pure VEM interfaces are relative inefficient. The four concepts based on the elastic springs have higher mean loss factors due to their compliant interfaces. Except comb 1, the last concepts are still the best concerning weight. Especially the surround concept has still the most superb efficiency.

![Figure 8.7: FRF of the concept Surround](image)

The introduced concepts in this chapter imply that an important condition for a high modal structural loss factor is to build a flexible interface. Sufficient shear stress should be introduced in the VEM. The surround concept provides a
good solution for the interface damping. Further optimization of ID can be an interesting future research topic.
Chapter 9

Comparison Study

In this chapter the performance of the different damping approaches and specimens, which were investigated experimentally on the plates in the second project phase (chapter 5, 6 and 7), are compared with their magnitude of the FRF responses, loss factors and weight specific damping efficiencies transversely considering the frequency influence. Their primary features, advantages and drawbacks are summarized.

9.1 Comparison of damping efficiency

The FRF of the three best specimens from ID, ACLD and PD are shown in figure 9.1 for the whole frequency range. The FRF of ID show, its average magnitude is the lowest and some modes with small amplitudes are even eliminated. But for a specific mode, its performance can be inferior to PD or ACLD. However, at global mode (frame resonance at 240 Hz), ID mitigates the vibration strongly while ACLD and PD don’t affect the vibration behavior significantly. The FRF of PD exhibits, at mode 1, the magnitude is superiorly damped by PD compared to ACLD and ID. At mode 9, PD mitigates the modes still better than ACLD. The FRF magnitude of ACLD is quite high below 150 Hz. Nevertheless, it suppresses the vibration more remarkably than PD and ID at mode 4. No approach surpasses the others in terms of amplitude suppression within the whole frequency range.

The modal loss factors $\eta_c$ of all measured samples are shown in figure 9.2. Because some modes with small peaks are eliminated by damping effects, these modes without data in the graph can not be evaluated. ID with 1 layer shows stable performance through the whole frequency range. At global mode, ID
Figure 9.1: specimens with best performance of each approach

has a high loss factor whereas the other two approaches are inefficient for this mode. The effective range of ACLD concentrates on one mode in this study. The other damped modes are not or only slightly damped. At mode 4, a great difference can be seen between open and close-loop. The PD is effective at all frequencies at antinodes. At mode 1, PD provides excellent damping and outperforms ID and ACLD (due to suboptimal damper position) considerably. At mode 4, the loss factors of PD are still high and in the same order as ACLD, whilst ACLD reaches already its maximum. At mode 9, the loss factors of PD remain high in spite of different mode / wave shape of the structure.

9.2 Comparison of damping-weight efficiency

In figure 9.3 the mean loss factors of the measured samples are presented in relation to their damper weights. We can see that ACLD has the lowest weight. PD has the intermediate and ID the highest weight.

The loss factors were calculated for damped, undamped, global and all modes respectively. The noticeable point is that, sectional dampers (ACLD and
9.2. Comparison of damping-weight efficiency

![Comparison of damping-weight efficiency](image)

**Figure 9.2: loss factor of every mode**

PD) perform better only at the damped modes (1, 4 and 9). If the undamped and global modes or whole frequency range is concerned, their efficiencies drop drastically. Conversely, ID shows still similar or even better values. As a consequence, the performance of ID is actually raised compared to the sectional dampers in these cases.

ID exhibits reasonable damping at damped modes. The mean value of 1 layer lies in the same order as the ACLD and 2 layers close to PD. Thanks to its stable damping at any frequency, their efficiencies do not change greatly concerning all the modes. Inversely, for 10 modes, the damping is raised slightly, because ID damps better at global resonance modes. That makes the absolute damping of ID 1 layer already competitive to PD. ID of 2 layers even exceeds
Figure 9.3: mean values of specific loss factors in relation to weight

any other investigated dampers.

The average efficiencies of ACLD are quite limited. By applying different voltage gains, the loss factor of ACLD can be solely raised by 1.4-2.2 $[10^{-2}]$. 


above the value of an undamped plate. The difference of loss factors between open and close-loop is about 30% for the damped modes. At undamped and global modes, the loss factors are almost the same as the undamped specimen. Thus, for all modes, the difference between open and close-loop is reduced to about 20% because of the narrow frequency range of effectiveness. Consequently, ACLD has the poorest damping value over the whole frequency range among all investigated treatments.

The PD possess polarized results. Simple PD have very low damping efficiency because of the lack of active movable particles. Thus, these inappropriate configured dampers have much lower damping efficiencies than the other dampers at the damped modes. It reveals the effective use of the particle weight with efficient configuration is an important aspect for PD. The performances of the honeycomb PD are greatly improved in contrast to simple dampers at the damped modes. They have the highest values for both absolute and weight specific loss factors by reaching maximal 5 $[10^{-2}]$. But like ACLD, the loss factors of PD are negligible at the undamped and global modes. By analyzing all 10 modes, the loss factors reduce drastically by 36-37% due to the poor damping at undamped and global modes. In spite of that, their loss factors are still relative high and superior to most of the other dampers for the whole frequency range. For both simple and additional honeycomb PDs, the intermediate cover wall in $2 \times 2$ mm specimen deteriorates the damping efficiency.

9.3 Conclusion

With all the phenomena observed in chapter 5, 6, 7 and above, the characteristics of the three damping approaches can be summarized to the following points:

- the interface damper (ID) acts stably in the whole frequency range. It is especially effective on suppressing the vibrations caused by the resonances from the peripheral environment, which is beneficial for damping automotive panels. ID is suitable for the situation where global damping or isolation of every mode is necessary, while no local mitigation is specially required. Even the own weight of ID is relative heavy, but concerning the unavoidable weight of conventional adhesive at the interface, the weight efficiency of ID is still high.

- the active constrained layer damper (ACLD) has a finite frequency range for the application. The lower modes with long wave lengths would require damper elements of big area. It is effective to damp modes with patch size similar to the half wavelength of the structure. ACLD is sensitive to the localization and its actuation direction. It is adequate to damp local
deflections at specific frequency (wavelength and direction). Compared to PD, another advantage of ACLD is that its efficiency is independent of the plate orientation. ACLD is a lightweight solution whereas external power source and control systems are required.

- the particle damper (PD) has superior damping performance. In low frequencies, the efficiency is excellent while in high frequencies high damping can still be obtained. By adjusting the configuration of the particles, container or vibration direction, great difference of PD’s performance can be induced. PD has intermediate weight among the three damping approaches.

These damping approaches behave quite different with respect to the frequency range, their geometric size and the applied locations. Corresponding to the vibration behavior of the panel structure, e.g. critical frequency range, mode pattern (distribution of critical vibration positions) and other specific requirements, the damping method for the panel should be selected according to the advantages of these approaches. ID is the unique treatment, which can effectively damp intense vibrations from the peripheral excitation source. For mitigating any specific local vibration on the structure at finite frequency, ACLD is a good option. If the aim is to have a high damping primary in the low frequency range and concurrently a steady efficiency in a broad frequency range is desired, PD should be applied. By combination of global dampers (ID) and sectional dampers (PD, ACLD), it is possible to control all the modes in the frequency range of interest.
Chapter 10

Application of Interface Damping on A Real Panel

This chapter presents the research of the last project phase. ID is applied on a real automotive panel. Section 10.1 gives a short introduction. Section 10.2 introduces the test case and the investigation scope. In section 10.3, the modeling and validation of the FE-model is described and the brief evaluation procedures are interpreted. The results of harmonic analysis with different materials uniformly applied at the interface are presented in section 10.4. Combined interfaces with adhesives and damping materials are further investigated in section 10.5. The loss factors of all the arrangements are compared in the conclusion in section 10.6.

10.1 Introduction

In chapter 7, interface damping (ID) was investigated on a simple plate. A parametric study based on the interface geometry and mechanical properties was accomplished. Some innovative concepts were developed. However, its applicability on real automotive structures stayed unknown. This chapter focuses on the application of interface damper on a real test case - a spare wheel pan (SWP). A whole system composed of SWP, periphery frame and soft flexible ID (the butyl as described in the previous chapters) was characterized and compared to systems with common commercial elastic or stiff adhesives at the interface. Because it is difficult to implement the whole system experimentally due to its geometric complexity for the manufacturing, the structures are investigated in this chapter by simulation.
This chapter can be mainly divided into two steps: In the first step, the concentration is laid merely on the damping validation of ID without considering any mechanical requirements of the interface. In the second step, some interface arrangements are investigated in order to fulfill also the demands of the structural loading capability on the SWP with ID.

### 10.2 Test case

A spare wheel pan as shown in figure 10.1 was defined as the final test case for the interface damper, since it is a typical panel which is glued all around the edges into the auto body. In the case of having no spare wheel in a station wagon, the vibration on the SWP exhibits representative behavior (similar mode shape etc.) of other vibratory panels in the vehicles. The SWP is made of glass fiber mat reinforced thermoplastic (polypropylene) (GMT).

![Figure 10.1: spare wheel pan (SWP)](image)

In order to raise the eigenfrequencies of the structure, it is designed as in figure 10.2 with some concave and convex surfaces, which have a uniform thickness of 2.5 mm. The central part of the SWP has some reinforcement ribs and other small features as shown in figure 10.3. The SWP is surrounded
by a flange of bonding edge. The important geometric data of the SWP are summarized in table 10.1.

![Figure 10.2: CAD model of the SWP](image)

<table>
<thead>
<tr>
<th>Volume</th>
<th>ca. $623 \times 657 \times 75$ mm</th>
</tr>
</thead>
<tbody>
<tr>
<td>Typical thickness</td>
<td>2.5 mm</td>
</tr>
<tr>
<td>Bonding edge surface width</td>
<td>14.3 – 20 mm</td>
</tr>
<tr>
<td>Bonding edge surface area</td>
<td>ca. 0.0313 m²</td>
</tr>
<tr>
<td>Whole SWP weight</td>
<td>1.41 kg</td>
</tr>
</tbody>
</table>

Table 10.1: geometry of the SWP

The SWP is further bonded with adhesive at its interface to the metallic vehicle structure. In order to investigate the influence of interface damping, the test case must be investigated together with this structure. A simplified frame (see next section) is used in the simulation instead of the real structure in figure 10.1.

Four different adhesives from stiff to soft range were used for the investigation together with the butyl interface: Araldite® 2014-1 and LMB 6687-2 are two stiff adhesive with the Young’s modulus of $E' = 4$ GPa and $E' = 1.1$ GPa at room temperature [110]. Collano® RS 8500 and Collano A8 4810 are soft elastic adhesive with the Young’s modulus of $E' = 60$ MPa and $E' = 2$ MPa [111, 112]. The former one is stiffer than butyl rubber by 3-20 times and the latter one is softer than butyl by 1-10 times. These adhesives were applied to the SWP in the shape of a simple uniform layer with a constant thickness (no additional geometric design like in the previous chapter).
Chapter 10. Application of Interface Damping on A Real Panel

10.3 Simulation and modeling

10.3.1 FE model of the spare wheel pan

Due to the geometric complexity, the model was preprocessed in Hyperworks® (Hypermesh). From the simulation point of view, it is more efficient to model the SWP with shell elements. However, the SWP doesn’t have a constant thickness overall. Since the majority areas on the SWP has the same thickness of 2.5 mm, the panel was divided into several sections (see figure 10.4), where different element types are defined, to raise the model efficiency for the calculation. Because of the complicated geometry of the reinforcement rib net, the blue mid-part was modeled with solid tetra elements (solid187 in Ansys). The purple areas with 2.5 mm thickness were modeled with shell elements (shell281). The green areas where is glued with adhesive was attempted to be modeled with quad shell elements (shell281) dominantly in order to facilitate the further extension of the model with SWP’s surrounding field. Regarding the regular cross section geometry, the outer edge was represented as hexahedral solid elements (solid186).

Because the surrounding metallic structure of the SWP in figure 10.1 and other supporting components would demand a lot of computational expense, the SWP was “embedded” into a simplified virtual periphery. The SWP was attached by means of an adhesive layer to a thin aluminum frame whose geometry corresponds to SWP’s gluing edge surface. Principally the interface has the same geometry as the periphery frame. Only their thicknesses are different as shown in figure 10.5. The hexahedral interface elements (solid186) were dragged by the quad elements from the green areas of the SWP (see figure 10.5). In order to have a decent basis of comparison with different adhesives, the thickness of the interface was chosen with the same value of 1.5 mm. This value is on one side the typical value of produced butyl layer and on the other side viable for all
other adhesive thicknesses. These interface elements were further extruded to generate the metal frame (solid186).

Some small features in the model like fillets or small areas were simplified in Hypermesh. The tetra elements at the mid-part have the edge lengths of 3-8 mm, while other elements have the edge lengths of 5-8 mm. All the elements (shell / solid) in system are second order elements. At the T-conjunction or overlapped position of shell to solid elements, the elements share the same nodes (merged). Like in the previous survey, the frequency range for the analysis of the SWP was taken up to 400 Hz.
Chapter 10. Application of Interface Damping on A Real Panel

The material properties of the components used in the simulation are summarized in table 10.2. The material damping of all components were considered, which are defined with viscoelastic models. In an experimental and simulation pre-study [113], the material loss factors of these four adhesives except butyl were determined with the same value of about $\eta_m = 0.5 \times 10^{-2}$. This value is frequency independent according to most experimental results, which were measured under ambient temperature. Precise material models of the commercial adhesives concerning temperature variation was not developed. Similarly, the SWP and aluminum frame were also defined with the tiny empirical value of $\eta_m = 0.5 \times 10^{-2}$ to constrain the resonance peaks to an evaluable level with a reasonable frequency resolution. It is to mention that actually the typical loss factor of GMT could be higher [114]. But with the heightened SWP material loss factor, meanwhile, the observation of ID’s efficiency becomes difficult, because the interface damping effect will be weaken relatively compared to the material damping of GMT. Another consideration is that the loss factor of the applied GMT stay unknown anyway. Thus, the small material loss factor of of $\eta_m = 0.5 \times 10^{-2}$, which was taken for the investigation, facilitates the evaluation in simulation.

<table>
<thead>
<tr>
<th>component</th>
<th>material</th>
<th>density $[kg/m^3]$</th>
<th>Young’s modulus $[MPa]$</th>
<th>Poisson ratio [-]</th>
<th>material loss factor [-]</th>
</tr>
</thead>
<tbody>
<tr>
<td>SWP periphery</td>
<td>GMT</td>
<td>1150</td>
<td>5400</td>
<td>0.34</td>
<td>0.005</td>
</tr>
<tr>
<td>interface</td>
<td>aluminum</td>
<td>2700</td>
<td>69000</td>
<td>0.33</td>
<td>0.005</td>
</tr>
<tr>
<td>interface</td>
<td>Araldite 2014-1</td>
<td>1600</td>
<td>4000</td>
<td>0.35</td>
<td>0.005</td>
</tr>
<tr>
<td>interface</td>
<td>LMB 6687-2</td>
<td>1100</td>
<td>1100</td>
<td>0.35</td>
<td>0.005</td>
</tr>
<tr>
<td>interface</td>
<td>butyl</td>
<td>1556</td>
<td>1.8-21 (see figure 7.3)</td>
<td>0.48</td>
<td>0.3-1.1 (see figure 7.3)</td>
</tr>
<tr>
<td>interface</td>
<td>Collano RS 8500</td>
<td>1100</td>
<td>60</td>
<td>0.4</td>
<td>0.005</td>
</tr>
<tr>
<td>interface</td>
<td>Collano A8 4810</td>
<td>1300</td>
<td>2</td>
<td>0.4</td>
<td>0.005</td>
</tr>
</tbody>
</table>

Table 10.2: material data (model) of the components in the simulation [115, 110, 112, 111]

Another pre-investigation for determining the first eigenmode of the system was performed by altering the aluminum frame’s thickness. The first mode of the frame/system vary from 50-80 Hz with its thickness change from 10 to 50 mm. Consequently, the thickness of the frame was chosen with 10 mm, since the typical global modes of the vehicles appear rather beginning from a low eigenfrequency between 25 and 45 Hz [28].
The system was analyzed under free-free condition. No DOF was constrained in the assembly. Like in the previous study, the system was excited at a single point on the thin aluminum frame, the position of the excitation is marked with the red arrow in figure 10.6.

**10.3.2 Model Verification**

The experimental validation of the SWP model is unfortunately not possible, because there are many variants of the composite materials. The exact compound percentage between matrix and fiber on the real part is not available by the supplier. There is only another SWP model which was analyzed in a previous confidential industrial study in Abaqus [115]. In order to verify the present model in Ansys, the results of a modal analysis in Ansys on the sole SWP without interface and frame were compared with the other developed confidential model in Abaqus in table 10.3. The boundary conditions for both models were the same: All degrees of freedom of the nodes at the gluing edge for the adhesive were suppressed.

The pattern of the three modes are illustrated in figure 10.7. The results are very similar and have a deviation of maximum $4 \, \text{Hz}$ up to $180 \, \text{Hz}$. We conclude that the precision of the Ansys model is acceptable and can be used for the further investigation.
Chapter 10. Application of Interface Damping on A Real Panel

<table>
<thead>
<tr>
<th></th>
<th>Abaqus</th>
<th>Ansys</th>
</tr>
</thead>
<tbody>
<tr>
<td>mode 1</td>
<td>79 Hz</td>
<td>80 Hz</td>
</tr>
<tr>
<td>mode 2</td>
<td>124 Hz</td>
<td>127 Hz</td>
</tr>
<tr>
<td>mode 3</td>
<td>182 Hz</td>
<td>178 Hz</td>
</tr>
</tbody>
</table>

Table 10.3: comparison of the modal analysis results

![Pattern of mode 1, 2 and 3](image)

Figure 10.7: pattern of mode 1, 2 and 3

### 10.3.3 Eigenmodes within the frequency range

A modal analysis is necessitated to attain the eigenfrequencies and mode patterns of the system for the selection of points, whose vibration behavior will be evaluated. The system was analyzed with a soft interface with the Young’s modulus $E’ = 10$ MPa, which is the mean value of the butyl rubber up to 400 Hz. The values of the eigenmodes are listed in table 10.4.

<table>
<thead>
<tr>
<th>No.</th>
<th>frequency [Hz]</th>
<th>No.</th>
<th>frequency [Hz]</th>
</tr>
</thead>
<tbody>
<tr>
<td>mode 1</td>
<td>47.08</td>
<td>mode 10</td>
<td>250.35</td>
</tr>
<tr>
<td>mode 2</td>
<td>67.58</td>
<td>mode 11</td>
<td>261.51</td>
</tr>
<tr>
<td>mode 3</td>
<td>93.29</td>
<td>mode 12</td>
<td>277.67</td>
</tr>
<tr>
<td>mode 4</td>
<td>126.52</td>
<td>mode 13</td>
<td>295.30</td>
</tr>
<tr>
<td>mode 5</td>
<td>152.53</td>
<td>mode 14</td>
<td>314.76</td>
</tr>
<tr>
<td>mode 6</td>
<td>173.87</td>
<td>mode 15</td>
<td>320.74</td>
</tr>
<tr>
<td>mode 7</td>
<td>181.60</td>
<td>mode 16</td>
<td>357.10</td>
</tr>
<tr>
<td>mode 8</td>
<td>204.18</td>
<td>mode 17</td>
<td>377.44</td>
</tr>
<tr>
<td>mode 9</td>
<td>240.80</td>
<td>mode 18</td>
<td>393.44</td>
</tr>
</tbody>
</table>

Table 10.4: eigen frequencies of the SWP system

The first 16 mode patterns are shown in figure 10.8. The first and second system mode originate from the periphery (aluminum frame). The areas with
Figure 10.8: first 16 mode patterns
acute displacement are concentrated at the gluing edge of the SWP. The third and fourth system mode come from the SWP, which has the dominant displacements. The first bending mode of the SWP is the third mode of the system. It is beneficial here to have these first four modes well separated in frequency domain. The manifest frequency differences enable a better observation of the interface’s influence to the global modes (periphery modes) and panel modes individually. Beginning from the fourth system mode, at many resonances the mode shapes are mostly coupled with both periphery and SWP modes.

10.3.4 Evaluation criteria

Similar to the evaluation process on the simple plate in chapter 3, only several points on the SWP were chosen for the calculation of the mean FRF to limit the evaluation expense. First the receptance, mobility and accelerance of the 18 modes listed in table 10.4 were overlapped and compared. Since most of the components were defined with material loss factors of $\eta_m = 0.5 \times 10^{-2}$, the system damping can be also a very small value. The observability of the points should thus serve as a vital criterion for the selection. In other case, some minor damping effects at certain modes may not be observed with the inappropriately chosen points. The gramian based algorithm [92] was thus applied in this chapter together with the ”direct” method described in chapter 3.

The overlapped receptance, mobility and accelerance of the SWP’s nodes (without gluing edge) in FE-model up to 400 Hz are plotted in figure 10.9. Because of the dominance of the first SWP bending mode, the plots are illustrated in two cases: overlapped mode shape with all modes or with all modes but except mode 3. The nodes with stronger amplitudes distribute in similar pattern for receptance, mobility or accelerance. Any plot from the three types can be chosen for determining the evaluation points. The mobility was selected and further compared with the gramian algorithm.

The applied gramian algorithm defines a performance index $PIX$ to describe the necessity of local sensor position. Higher values indicate recommendation for placing sensors. $PIX$ is expressed with the following equation:

$$PIX = 2 \left( \sum_{i=1}^{n} S_i \right) \sqrt[10]{\prod_{i=1}^{n} (S_i)} \quad (10.1)$$

$S_i$ is the total energy at mode number $i$. In the harmonic analysis of Ansys, $S_i$ corresponds to the modal strain energy (SENE). The term in the root sign part
Figure 10.9: overlapped receptance, mobility and accelerance on the SWP
vanishes, if the nodal strain energy at any mode is close to zero. i.e., if a point near to a node position at any mode is chosen, the $PIX$ has an extreme low value. Only the positions possessing certain displacement (e.g. points close to antinodes) at all modes within the whole frequency range of interest have high $PIX$ values. These points can be thus utilized as the evaluation points which should reveal the system behavior at any mode.

The $PIX$ values according to the gramian algorithm are plotted together with the overlapped mobilities of the SWP in figure 10.10. The distribution of areas with the highest performances are quite dissimilar between two algorithms. Many appropriate regions for the point choice are even in contrary. This can be explained by their concept ideas: The direct overlapped mobility chooses the points with dominant peaks regardless of its observability at all modes. It is thus better for the observation of particular modes with distinct magnitudes. The
10.4. Results of uniform interfaces

Figure 10.11: choice of 30 points for the evaluation

The gramian algorithm exploits the points that have good mean vibration amplitudes whereas the peak at a single mode may not be recorded remarkably as the direct method. Inverse to the direct method, the gramian algorithm is more suitable for the observation of all modes.

A trade-off was addressed concerning both aspects. The regions, where dominant values of any principle prevail, are marked in figure 10.11. 30 points were totally selected at the crossed positions. About 10 points were chosen at places where both principle are adequate. For each single principle, further 10 points were selected respectively. The points are mostly located on relative flat and big areas which are commonly used for experiments.

The in this section introduced procedures can be exploited not only for the selection of sensor (evaluation) positions, but also for locating dampers and actuators.
10.4 Results of uniform interfaces

In this section, the results of harmonic analysis are to be presented. The investigated interface was filled uniformly with either the introduced real adhesives or butyl ID. i.e., five interfaces are compared with each other. The aim of this step is purely to validated the viability of ID on the SWP concerning its vibration behavior. Therefore, the mechanical properties will be discussed later in the next section.

Since the composite loss factor $\eta_c$ is small by defining most of the components with $\eta_m = 0.5 \times 10^{-2}$, a relative fine resolution of 0.5 Hz was used for the harmonic analyses. At some resonances, additional frequency points were interpolated to get the exact resonance peak values. Furthermore, the composite loss factors $\eta_c$ were fitted by the tool introduced in the previous chapter.

First the responses of stiff adhesives are compared. The FRF of the system using Araldite 2014-1 and LMB 6687-2 interface are plotted in figure 10.12 and 10.13. By using these two stiff adhesives, merely very slight difference can be observed. Thus in the further contrast, only the Araldite interface is compared to the compliant systems. Due to the stiff interface, there are only 17 modes
Next let’s observe the difference between stiff and soft adhesives. The FRF of Araldite 2014-1, Collano RS 8500 and Collano A8 4810 are illustrated in figure 10.14 and 10.15. Up to 200 Hz, the system amplitudes have similar levels with the different interfaces, even though the difference of interface’s Young’s-modulus value is 2000 times. For the first global mode at about 47 Hz, a pure flexible elastic interface of Collano adhesive without significant damping property induces even stronger vibrations on the SWP. In contrast, the second panel mode at about 126 Hz can be much better isolated with a more compliant interface.

From 200 to 400 Hz in figure 10.15, the amplitudes of different adhesives are still in the similar range. The mode at 385 Hz is greatly influenced through the interface property. Due to the very low stiffness of Collano A8 4810, some additional modes are incurred. The advance of isolation of a flexible panel is limited only for several modes in the whole frequency range. For the majority of the modes, there is no improvement perceivable. Concurrently, it brings even enhancement of vibrations at some modes. Thus, through the application of a compliant elastic interface, it is difficult to diagnose if the vibration on the panel appearing within the frequency range of 400 Hz.

![Figure 10.13: FRF of Araldite 2014-1 / LMB 6687-2 interface frome 200-400Hz](image-url)
Figure 10.14: FRF of Araldite 2014-1, Collano RS 8500 and Collano A8 4810 interface from 0-200 Hz

Figure 10.15: FRF of Araldite 2014-1, Collano RS 8500 and Collano A8 4810 interface from 200-400 Hz
can be attenuated in a wide frequency range compared to a stiff interface.

It can be concluded from these figures, a soft interface alone doesn’t attribute to the attenuation significantly. Only if the interface has sufficient damping characteristics, the vibration on the panel can be suppressed based concurrently on the damping and isolation.

![Graph showing FRF of Collano RS 8500 and Butyl interface from 0-200 Hz]

Figure 10.16: FRF of Collano RS 8500 / butyl interface from 0-200 Hz

The interface damper with butyl is compared to the flexible interface Collano RS 8500 in figure 10.16 and 10.17. Independent from mode shape or type, all modes within the range of interest are significantly attenuated. For the global mode 1 and 2, amplitudes of both modes were greatly reduced by almost 9 dB with butyl in contrast to Collano RS 8500. Even at the first panel bending mode (mode 3), 6 dB improvement can be observed. At higher modes, the amplitudes were suppressed by 4-12 dB.

The evaluated system composite loss factors $\eta_c$ are listed in table 10.5. Since the components or materials except butyl are all defined with a frequency independent material loss factor of $\eta_m = 0.5 \times 10^{-2}$ (dmpr in Ansys), the composite loss factor of systems with Araldite, LMB and Collano adhesives have almost constant values of $\eta_c \approx 0.5 \times 10^{-2}$ through the whole frequency range. Some modes having high loss factors were caused by the weak peak shape (not enough
In contrast, $\eta_c$ with butyl ID have overall considerable values. They arrive 1-2% at different modes. Mode 14 was almost eliminated. The values at high frequencies are slightly higher than at low frequencies.

### 10.5 Investigation on dual-component interfaces

In the last section, only the damping behavior of the SWP system was concerned. A real interface should of course meet mechanical requirements besides damping concurrently as well. The applied uniform damping interface with butyl in this thesis can however barely afford any loads. Thus, the combination of damping and structural bonding material at the interface is discussed in this section with the envision to tackle both demands.

#### 10.5.1 mechanical properties of the interface

The sealing properties and bonding strength are the most critical mechanical aspects for the interface construction. In fact, butyl is a typical material for the body joint sealing. The strength of butyl is moderate. The range of its shear
10.5. Investigation on dual-component interfaces

<table>
<thead>
<tr>
<th>mode No.</th>
<th>Araldite 2014-1</th>
<th>LMB 6687-2</th>
<th>Collano RS 8500</th>
<th>butyl</th>
<th>Collano A8 4810</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>0.59</td>
<td>0.50</td>
<td>0.54</td>
<td>1.53</td>
<td>0.58</td>
</tr>
<tr>
<td>2</td>
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<td>0.48</td>
</tr>
<tr>
<td>3</td>
<td>0.51</td>
<td>0.50</td>
<td>0.50</td>
<td>0.96</td>
<td>0.51</td>
</tr>
<tr>
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<td>0.51</td>
<td>0.94</td>
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<td>0.54</td>
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<td>1.86</td>
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<td>17</td>
<td>0.60</td>
<td>0.59</td>
<td>0.55</td>
<td>2.09</td>
<td>0.50</td>
</tr>
<tr>
<td>mean</td>
<td>0.54</td>
<td>0.54</td>
<td>0.52</td>
<td>1.38</td>
<td>0.56</td>
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</tbody>
</table>

Table 10.5: modal loss factors $\eta_c \times 10^{-2}$ of the uniform interface systems

strength can reach up to 2 MPa [116], which is below the strength of adhesives used for major automobile structures. For the bonding of automotive bodies and other components, common adhesives are: polyurethane, epoxy, PVC-plastisol, acrylic plastisols and in some cases also butyl rubber based adhesive [117]. Therefore, as a prevalent sealant, the strength of the butyl interface is the only crucial issue that should be tackled.

During driving, the acceleration / deceleration are the most critical situations, in which the interface was loaded primarily under shear stresses. The critical load $X_{critical}$ can be expressed by:

$$X_{critical} < \frac{\tau_{critical} \cdot A}{m \cdot S_{critical}}$$  (10.2)
where \( \tau_{\text{critical}} \) is the maximal shear stress of the interface, \( A \) is the area of bonding, \( m \) is the mass of the SWP and \( S_{\text{critical}} \) is the safety factor. All these parameters were measured in the experiments.

The applied butyl rubber in this thesis, which is specified for damping utilization, has however a very low shear strength (merely ca. \( 21 \) Pa). Assuming that a safety factor \( S_{\text{critical}} = 2 \) is taken, the investigated uniform butyl interface of the SWP can withstand only a load of \( 0.23 \) g, which apparently doesn’t meet the requirement in critical driving situations. In contrast, with a modified butyl interface from the literature [116], the SWP can be loaded up to \( 22 \) g.

The combination of butyl and the four investigated structural adhesives, which have much higher shear strengths (from \( 2.4 \) to \( 16.7 \) MPa) than butyl, can be a solution to strengthen the interface. Thus, half of the interface material is replaced as indicated in figure 10.18. The inner half of the bonding edge is permanently filled with butyl, whereas the outer half edge is filled with one of the other four commercial adhesives. i.e., the interface is formed by the combination of dual-components with 50% damping material and 50% structural adhesive. The volume of the whole interface remains unchanged compared to the uniform interface systems.

![Figure 10.18: cross section view of the dual-component interface](image)

Next let’s postulate that for the dual-component interface system, one half area of the SWP’s bonding edge has no loading capability (strength of butyl will be neglected due to the enormous difference to the adhesives). SWP is then entirely loaded at the other half bonding edge filled with structural adhesives. In this case, the loading capacity of the SWP can be strongly enhanced as listed in table 10.6.
10.5. Investigation on dual-component interfaces

<table>
<thead>
<tr>
<th>50 % damping material</th>
<th>50 % adhesive</th>
<th>critical loading limit</th>
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<tbody>
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<td>butyl rubber</td>
<td>Araldite 2014-1</td>
<td>93.0 g</td>
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<tr>
<td>butyl rubber</td>
<td>LMB 6687-2</td>
<td>67.1 g</td>
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<td>butyl rubber</td>
<td>Collano RS 8500</td>
<td>22.7 g</td>
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<tr>
<td>butyl rubber</td>
<td>Collano A8 4810</td>
<td>13.3 g</td>
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</table>

Table 10.6: loading ability of different interface materials

With the heightened mechanical strength, the dual-component interface are further investigated in the harmonic analysis with the same boundary conditions as the uniform interfaces for the damping evaluation.

10.5.2 results of dual-component interfaces

The harmonic analysis results of the dual-component interfaces with 50% Araldite 2014-1, LMB 6687-2 and Collano RS 8500 are illustrated in figure 10.19 and 10.20.

![Graph of FRF](image)

Figure 10.19: FRF of dual-component interfaces with Araldite 2014-1, LMB 6687-2 and Collano RS 8500 from 0-200 Hz
Figure 10.20: FRF of dual-component interfaces with Araldite 2014-1, LMB 6687-2 and Collano RS 8500 from 200-400 Hz

Figure 10.21: FRF of dual-component interfaces with Collano RS 8500 and Collano A8 4810 from 0-200 Hz
10.5. Investigation on dual-component interfaces

Figure 10.22: FRF of dual-component interfaces with Collano RS 8500 and Collano A8 4810 from 200-400 Hz

Below 200 Hz, the responses of the three different dual-interfaces are mostly overlapped. Only the magnitudes of Collano RS 8500 interface are slightly lower (within 1 dB) than the other two stiff interfaces. An amplitude suppression up to 5 dB can be observed with the dual-component interfaces compared to the case of uniform Araldite 2014-1 interface from figure 10.12 / 10.13. For the first two global modes, the amplitudes were reduced by 4.5 and 5 dB respectively. The improvement at other panel modes is comparatively minor up to 3 dB.

Like the uniform interfaces, the responses of Araldite 2014-1 and LMB 6687-2 dual-component interface are almost identical even until 400 Hz. Upon 200 Hz, the attenuation of these two dual-interfaces remain within the range of 0.5-4.5 dB in contrast to the uniform Araldite interface. Based on these two stiff dual-interfaces, a further universal improvement of 1-2 dB can be seen on the more compliant Collano RS 8500 interface up to 250 Hz.

As the Collano RS 8500 dual-interface has noticeable damping properties, it is further compared with the uniform butyl interface from figure 10.16 / 10.17 and the Collano A8 4810 dual-interfaces in figure 10.21 and 10.22. Generally the Collano A8 4810 dual-interface performs better than Collano RS 8500 by attenuating 2-4.6 dB in figure 10.21. Its behavior is quite similar as the uniform
butyl interface up to 200 Hz. Since the Collano A8 4810 dual-interface has a lower stiffness than the uniform butyl interface, its damping efficiency surpasses even slightly the uniform butyl interface by applying solely half VEM quantity.

<table>
<thead>
<tr>
<th>mode No.</th>
<th>50% Araldite 2014-1</th>
<th>50% LMB 6687-2</th>
<th>50% Collano RS 8500</th>
<th>100% butyl</th>
<th>50% Collano A8 4810</th>
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<td>0.59</td>
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<td>0.94</td>
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<td>1.00</td>
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<tr>
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<td>0.88</td>
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<td>2.53</td>
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<tr>
<td>mean</td>
<td>0.75</td>
<td>0.76</td>
<td>0.90</td>
<td>1.38</td>
<td>1.57</td>
</tr>
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</table>

Table 10.7: modal loss factors $\eta_c$ [$10^{-2}$] of the dual-component interface systems

Above 200 Hz, the advantage of Collano A8 4810 dual-interface becomes remarkable. Its resonant peaks are in most cases 1-1.6 dB lower than the uniform butyl interface. Because at high frequencies, the Young’s modulus of butyl increases intensely, the deformation at the uniform VEM interface was declined which deteriorates the dissipation. Mode 14 was eliminated by the dual-composite interface like the uniform butyl interface. In comparison to the Collano RS 8500 dual-interface, the amplitudes of Collano A8 4810 were further reduced by 2.4-6 dB.
It can be concluded that the dual-component interface can be a compromise option between damping and strength. Even with the stiffest Araldite 2014-1 dual-interface, an average reduction of resonance peaks can be achieved by 2.7 dB in contrast to the uniform Araldite interface from 0-400 Hz, while the mechanical requirements can be satisfied. An extreme case is, through the application of the dual-component interface with Collano A8 4810, both damping benefit and mechanical improvement can be reaped. It indicates that principally an adhesive with low stiffness but high fraction strength would be superior for the dual-component interface, which could promote the weight-efficiency of the VEM and possess competitive or better damping behavior as the uniform damping interface.

The structural loss factors of the systems with dual-component interfaces are listed in table 10.7. Similar to the amplitude responses, $\eta_c$ of Araldite 2014-1 and LMB 6687-2 have only light differences with the average deviation of 0.01%. Their loss factors are 0.2% higher than the undamped uniform interfaces. The Collano RS 8500 affords higher damping values reaching $\eta_c = 0.9$ with its lower stiffness. The Collano A8 4810 dual-component interface outperforms the uniform butyl interface almost entirely by gaining additional 0.2% damping with only half butyl volume.

## 10.6 Conclusion

In this chapter, the characteristics of the interface damper (ID) were investigated on a real vehicle panel structure - a spare wheel pan (SWP), by means of simulation. The SWP was bonded with different interface adhesives at its gluing edge to an aluminum frame. Various soft and stiff adhesives, which are commercially available, were investigated. Up to 400 Hz, the SWP conjuncted with conventional adhesive possess similar vibration behavior. No substantial attenuation can be achieved. However, by using the butyl ID, the SWP was remarkably damped. In the whole frequency range, the amplitudes of each resonances were reduced by 4-12 dB with an additional average composite loss factor of 0.9%, which was promoted by ID. ID can thus be a promising damping approach for many automotive panel structures, that are bonded to the primary vehicle structures.

The mechanical strength of the uniform damping interface is however not sufficient in critical driving cases. In consequence, an additional investigation on the dual-component interface with the combination of structural adhesives and damping material was conducted. For the comparison of these two interface types, the results of the uniform interface systems are summarized in figure
Chapter 10. Application of Interface Damping on A Real Panel

Figure 10.23: structural loss factors of the uniform interface systems

Figure 10.24: structural loss factors of the dual-component interface systems
10.23. Figure 10.24 presents the results of dual-component interfaces consisting of 50% adhesives and 50% butyl.

Through the whole frequency range, most systems with the interfaces employing butyl have steady damping characteristics. On all damped systems, the global mode 1 and 2 were universally attenuated with marked improvements on structural loss factors $\eta_c$ compared to the systems with uniform adhesive interfaces. The damping efficiency at panel modes from 3 to 8 are inferior to the global modes. At high frequencies above 200 Hz, the loss factors are overall raised which implies that ID is even slightly advantageous for higher modes because of shorter structural waves and eventually higher shear rates. More compliant interface materials heighten the dampening efficiency at high frequencies.

A panel system with balanced damping and mechanical properties can be therefore designed with the dual-component interfaces. The damping capability of the dual-component interfaces depends on both interface components. A compliant adhesive material with relatively high fraction strength and a soft damping material with high loss factors are the ideal choices for this application.
Chapter 11

Conclusion and Outlook

In this chapter, a brief review of the milestones in this thesis is stated. The most important achievements are concluded and outlook about the further development of damping technologies on vehicle panels is envisaged.

11.1 Review of main results

This thesis investigates alternative damping approaches which can substitute classical add-on dampers of viscoelastic material (VEM) for panel structures in automotive applications. Multifarious approaches were compared in a broad range on simple beam structures. Superior approaches were selected to a narrow scope and further experimented on plates. At last the focus was laid on a single approach, which was then systematically investigated and applied to a real structure.

In the first step, a comprehensive study on various damping approaches which may be applied to plate structures was performed experimentally. A test rig based on the DMA-principle was developed for the measurement in low frequency range. The dampers were mounted on a beam in the three configurations: surface, marginal and interface. Three approaches showed interesting results and were chosen for the further investigation. Interface damping (ID) using butyl rubber contributes to competitive damping with few VEM. Active constrained layer damping (ACLD) applying piezo elements enhances the mitigation performance of classical VEM with its advantage of lightweight. Particle damping (PD) provides a simple economic damping solution with high efficiency.
In the second step, ID, ACLD and PD were further experimented on a plate mounted to a sandwich frame representing the periphery. Another test rig capable of measuring in a higher frequency range was build and utilized. All three approaches were tested up to 400 Hz. The sectional dampers ACLD and PD exhibit excellent efficiency at certain eigenmodes, whereas they are ineffective at other modes. Even though ID is inferior to the sectional dampers at some specific modes, it reduces however globally the vibration on the plate in a wide frequency range. The average loss factor of ID surpasses the sectional dampers. ID has an additional unique advantage compared to sectional dampers: It mitigates the structure also at global modes, where the dominant vibration is introduced into the structure through the periphery. Therefore, ID was determined to be further investigated.

In the third step, a FE model was generated and correlated to the experimental results from the second phase. Based on this model, a parametric study was carried out in simulation. The influences of the interface material properties and geometry to the system’s vibration behavior were characterized. A compliant and strongly viscoelastic interface is generally beneficial. 16 novel interface concepts with exploring geometry were conceived on a beam structure and analyzed in simulation. The best concept has remarkable potential in terms of attenuation improvement and lightweight advantage.

In the last step, ID was implemented on a real vehicle panel: a spare wheel pan, by simulation. Different commercial adhesives and butyl ID were applied at the interface. They were analyzed under the same boundary conditions. The results showed, independent of the adhesive’s stiffness, the spare wheel pan presented similar responses with conventional elastic adhesives. Only the viscoelastic butyl interface achieved a pronounced amplitude reduction by max. 75% in contrast to an elastic interface. The trade-off with a dual-component interface design may satisfy both damping and mechanical strength requirements.

11.2 Conclusions

Through the systematical screening of various damping approaches based on beam specimens, the classical viscoelastic damper was successfully compared with the other approaches. The positive information for the automotive industry is that the simple classical CLD damper is still very effective among all these approaches. Nevertheless, non-conventional materials and treatments exhibit also their own potentials, which overcome the drawbacks of the classical approaches partially.
11.2. Conclusions

Unlike CLD, which contributes to marked damping only with the significant bending deformation of the substrate, the mitigation effect yielded by PD is based on the translational movements. This fact leads to that PD can provide reliable damping at many modes, independent of the mode type, as long as there is sufficient displacement / velocity during the vibration. This merit of PD can be an attractive solution also for stiff panels like sandwich composites, in which the bending strain is moderate. Another advantage of PD is that it is not sensitive to the environment change. The severe material frequency-temperature dependence of classical VEM damper do not exist on PD. PD can be thus utilized in diverse harsh environments. Furthermore, PD is an economic solution. Even ordinary metal chips can be used as damper. In consequence, PD can be a complementary approach to the classical treatments, e.g., it can make up the efficiency gap of CLD at some specific modes, under extreme temperatures or be used on vehicle sandwich panels. But an exact prediction of PD’s performance is difficult to fulfill. The dynamic behaviors of particles cannot be simulated with traditional FEM softwares. Additional discrete element method (DEM) for the specific simulation of granular materials [72, 71] is necessitated. Even though DEM has been rapidly developed, the investigation of damping by means of combined DEM and FEM remains still in a relative primitive research phase. Nevertheless, PD can thoroughly be appealing to both the vehicle damping community and the academic research with its superior performance but diverse open questions.

Because ACLD is an advanced damper version of CLD, it shares many properties of the classical dampers like high basic damping value. For its application on plates, the added value of ACLD is however limited to some specific mode patterns, since piezo-patches are normally only actuated in one direction. The damping in the perpendicular direction to the actuated direction on the plate can not be increased. The promoted efficiency can be thus only extracted, if the dimensions and direction of ACLD are well tailored for several finite modes. Although ACLD can be very effective for damping specific modes, as a killing factor, the cost of the piezoelectric patches, which can be at least a hundred times more expensive than a common passive CLD patch, inhibits its widespread use. The complexity of ACLD is another obstacle. Electric circuits are required for both sensors and actuators. Instability can be incurred by the active configuration. Also the assembly of ACLD is quite arduous. The bulk production with ACLD can be challenging. In general, ACLD is suitable for the application rather in a narrow scope at vital places where intense vibration is an absolute taboo. A generic application on vehicle panels can be a tough issue at least in the next few years.
In spite of the high damping performance of PD or ACLD under certain situations, the structure of surface treatments confines its effectiveness with the vulnerable point that the configuration of sectional dampers should be tuned in conformity with local oscillation conditions. An universal upgrade of mitigation for all modes within the entire frequency range of interest can be barely achieved.

In comparison, ID with VEM fulfills this mission as verified in this contribution. Actually the damping principle of ID is straightforward: If the interface is not excessively rigid, deformation of VEM and consequently damping can be generated at any modes, since the adherends vibrate commonly in different patterns. In chassis, the different thermal expansion coefficients of various components or materials can induce deformation differences up to several millimeters between adherends under different temperatures. These deformation tolerances at many interfaces must be normally compensated by using compliant adhesives. This implies, where the compliant adhesives are applied for the bonding, the application of ID is possible. And ID do cover a wide frequency range. This can be an important merit for gaining the ride comfort, since the excitations at low frequencies are particularly detrimental. In most literatures, the dynamic damping features of the interface haven’t been systematically studied. With the simulation results in this thesis, now we know that if the material properties of VEM are tuned according to the "soft & viscous" principle, the universal damping of the whole system can be greatly heightened. The structural damping can also be improved with the variation of the interface geometry with the principal idea of "shearing-oriented interface shape" mentioned in the last section. From the mechanical point of view, hybrid interface concept settles the problem of system strength even in the worst case if the damping material is mechanically too weak. Moreover, the overlapped area at the interface can be enlarged, so that it yields more space for the adhesive part in the hybrid interface and the whole system can be strengthened. On the other side, simulation techniques are sophisticated enough to describe most details of ID’s damping phenomena. The treatment can be universally well manipulated by simulation and in practice. At last, ID can be integrated into the production process analogical to conventional adhesives. In the author’s opinion, it is a method whose full potential can be rapidly taped in the near future and is applicable for the mass production with a high damping-cost efficiency being concurrently also decent weight-restrained.

11.3 Outlook

The efficiency of PD has been validated with several configurations in this thesis. But the influence of many parameter on the damping haven’t been examined, e.g., the particle size, shape and material. Further experimental studies on these aspects will be very helpful for a better insight of PD. Since the PD’s behav-
iors can hardly be simulated due to these mentioned parameters and current methodologies, its performance cannot be predicted before the implementation. The damping performance of PD is highly related to the weight. Of course, we want to circumvent the weight penalty with certain trade-offs. How to develop an algorithm, which can be used to approximate certain particle parameters for reaching the minimal expected damping level, will be valuable. For the simulation, DEM has still many open questions. There are considerable spaces for the improvement, especially with the development of computing efficiency. The most critical point can be the emulation of the behaviors of irregular particles, which violate a number of fundamental assumptions and theories of DEM. If the mentioned problems in the simulation can be tackled, it can be envisioned that later the combination of DEM and FEM will offer a powerful tool for the comprehensive analysis of PD.

Compared with the other two approaches, ACLD has been studied comprehensively with varieties of analytical models in the last decades. However, as a global effective approach for beams, ACLD encounters certain drawbacks for the application on plates, especially the spillover effects [96]. How to minimize and avoid this effect by developing robust control algorithms can be a crucial point. For some high quality cars, ACLD can be still an adequate solution for addressing some particular deleterious modes in a partial frequency range. For the industrial interest, how to simplify the assembly process of ACLD could be an essential topic.

Various aspects of the ID have been investigated. Nevertheless, the focus was mainly concentrated on the vibration behavior of the system. From the material science point of view, the development of a new VEM compound with the following objective properties can become an interesting subject: soft, highly viscous and with high fraction strength. From the production point of view, the bonding of ID must be integrated into the whole process chain. With the spray damping technology, maybe even a hybrid interface can be easily glued with the help of robotic equipments. Regarding the mechanical properties, behaviors like aging or creeping of the VEM interface haven’t been researched. These long-term behaviors are also essential for its application. Further optimization concerning the interface’s material and geometry properties in a real environment is entailed. Precise dimensioning of the dual-component interface concerning both oscillation and mechanical properties is necessary in the future. Besides miscellaneous metallic and composite panels, the application of ID can be extended to many other structures. For example, it can be interesting for the glazing process. Because in this case no dampers will be bonded on the surface of a glass, the damping can only be introduced through the interface.
Appendix A

Technical Drawing
Figure A.1: test rig used for the beam specimens
Figure A.2: dimension of the spare wheel pan
Appendix B

Results of PD on a Sandwich Plate
Sandwich plate dimension and material

<table>
<thead>
<tr>
<th>Component</th>
<th>Material</th>
<th>Thickness</th>
</tr>
</thead>
<tbody>
<tr>
<td>Face</td>
<td>Aluminum</td>
<td>1 mm</td>
</tr>
<tr>
<td>Core</td>
<td>PET foam</td>
<td>10 mm</td>
</tr>
</tbody>
</table>

Plate sample configuration (tested with 100-600 Hz)

1. simple sandwich  
   Weight: 1799 g

2. empty hole  
   Weight: 1805 g

3. hole filled with weight  
   Weight: 2083 g

4. hole filled with particles  
   Weight: 2013 g (particle mass ratio 0.15)
Tests of plate in frequency domain

\[ \text{FRF}_{\text{Mobility}} = \frac{v}{F} \quad \text{Velocity (laser)} \]

\[ \text{FRF}_{\text{Mobility_plate}} = 20 \cdot \log \left( \sqrt{\frac{\sum_{i=1}^{n} \frac{|v|^2}{F}}}{m} \right) \]

**Loss factor from FRF:**

\[ \eta = \frac{\Delta \omega}{\omega_{\text{res}}} \]

**Specimen - measuring points**

35 points (RMS FRF)
1. Simple sandwich (without holes)

<table>
<thead>
<tr>
<th>Peak 1</th>
<th>Peak 2</th>
<th>Peak 3</th>
<th>Peak 4</th>
<th>Peak 5</th>
</tr>
</thead>
<tbody>
<tr>
<td>Amp (dB)</td>
<td>η (%)</td>
<td>Amp (dB)</td>
<td>η (%)</td>
<td>Amp (dB)</td>
</tr>
<tr>
<td>56.12</td>
<td>1.58</td>
<td>81.95</td>
<td>0.41</td>
<td>54.74</td>
</tr>
</tbody>
</table>

Mean loss factor: 1.18 [%]

2. Sandwich without particles (hollow container)

<table>
<thead>
<tr>
<th>Peak 1</th>
<th>Peak 2</th>
<th>Peak 3</th>
<th>Peak 4</th>
<th>Peak 5</th>
<th>Peak 6</th>
<th>Peak 7</th>
<th>Peak 8</th>
<th>Peak 9</th>
</tr>
</thead>
<tbody>
<tr>
<td>Amp (dB)</td>
<td>η (%)</td>
<td>Amp (dB)</td>
<td>η (%)</td>
<td>Amp (dB)</td>
<td>η (%)</td>
<td>Amp (dB)</td>
<td>η (%)</td>
<td>Amp (dB)</td>
</tr>
<tr>
<td>69.17</td>
<td>0.96</td>
<td>49.06</td>
<td>1.86</td>
<td>65.68</td>
<td>1.22</td>
<td>57.87</td>
<td>1.28</td>
<td></td>
</tr>
</tbody>
</table>

Mean loss factor: 1.23 [%]
3. Sandwich with lumped weight (single slugs)

Mean loss factor: 1.25 [%]

4. Sandwich with particles (particles filled container)

Mean loss factor: 3.32 [%]
Chapter B. Results of PD on a Sandwich Plate

Results

Comparison of specimens 2-4

<table>
<thead>
<tr>
<th>Samples</th>
<th>Mode 1 (Amp. [dB])</th>
<th>Mode 2 (Amp. [dB])</th>
<th>Mode 3 (%)</th>
<th>Mode 4 (%)</th>
<th>Mode 5 (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Empty</td>
<td>77.33</td>
<td>51.27</td>
<td>1.07</td>
<td>63.51</td>
<td>1.07</td>
</tr>
<tr>
<td>Weight</td>
<td>70.46</td>
<td>53.09</td>
<td>1.09</td>
<td>66.12</td>
<td>0.68</td>
</tr>
<tr>
<td>Particle</td>
<td>66.63</td>
<td>48.64</td>
<td>2.88</td>
<td>50.39</td>
<td>4.33</td>
</tr>
</tbody>
</table>

Amp: Amplitude, %: Percentage
Appendix C

Resolution Test

In this study, the influence of the scanning resolution in sine-sweep test on the data quality is investigated with several significant modes (peaks) of some specimens.

Figure C.1: resonance peaks measured with different resolutions
The sample with a simple 4 mm particle damper (see chapter 5), which has a poor damping efficiency, was mainly investigated. The mode 1, 2, 4, 9 and 10, which have quite high velocity magnitudes were measured to investigate the influence of the resolution under different frequencies. Some peaks from the samples with particle dampers of better performance were also measured to survey the difference between poor and well damped structures. The study is divided into two steps. The resolution limit, which is fine enough to represent the reality, will be determined in the first step. Then the differences from coarse to fine resolutions is investigated in the second step.

In the first step, two samples with poor and excellent damping were measured as in figure C.1 with five different resolutions: 1, 0.5, 0.25, 0.1 and 0.05 Hz. From the results it can be observed, the curves of 1 and 0.5 Hz have some difference to 0.25, 0.1 and 0.05 Hz. The distortion is smaller with higher resolutions. The curves of 0.1 and 0.05 Hz are well overlapped with each other, which indicates

<table>
<thead>
<tr>
<th>peak</th>
<th>fitted parameter</th>
<th>resolution</th>
<th>max. deviation</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>1 Hz</td>
<td>0.5 Hz</td>
</tr>
<tr>
<td>PD 4mm mode 1</td>
<td>$f_{res} \ [Hz]$</td>
<td>64.64</td>
<td>64.34</td>
</tr>
<tr>
<td></td>
<td>$amp_{res} \ [dB]$</td>
<td>73.4</td>
<td>71.9</td>
</tr>
<tr>
<td></td>
<td>$\eta \ [10^{-2}]$</td>
<td>1.73</td>
<td>1.92</td>
</tr>
<tr>
<td>PD 4mm mode 2</td>
<td>$f_{res} \ [Hz]$</td>
<td>131.23</td>
<td>131.00</td>
</tr>
<tr>
<td></td>
<td>$amp_{res} \ [dB]$</td>
<td>68.5</td>
<td>69.2</td>
</tr>
<tr>
<td></td>
<td>$\eta \ [10^{-2}]$</td>
<td>0.88</td>
<td>0.74</td>
</tr>
<tr>
<td>PD 4mm mode 4</td>
<td>$f_{res} \ [Hz]$</td>
<td>204.40</td>
<td>203.99</td>
</tr>
<tr>
<td></td>
<td>$amp_{res} \ [dB]$</td>
<td>51.7</td>
<td>52.2</td>
</tr>
<tr>
<td></td>
<td>$\eta \ [10^{-2}]$</td>
<td>3.49</td>
<td>3.41</td>
</tr>
<tr>
<td>PD 4mm mode 9</td>
<td>$f_{res} \ [Hz]$</td>
<td>305.85</td>
<td>305.87</td>
</tr>
<tr>
<td></td>
<td>$amp_{res} \ [dB]$</td>
<td>52.1</td>
<td>51.9</td>
</tr>
<tr>
<td></td>
<td>$\eta \ [10^{-2}]$</td>
<td>2.61</td>
<td>2.66</td>
</tr>
<tr>
<td>PD 4mm mode 10</td>
<td>$f_{res} \ [Hz]$</td>
<td>380.02</td>
<td>380.04</td>
</tr>
<tr>
<td></td>
<td>$amp_{res} \ [dB]$</td>
<td>52.9</td>
<td>52.9</td>
</tr>
<tr>
<td></td>
<td>$\eta \ [10^{-2}]$</td>
<td>0.65</td>
<td>0.63</td>
</tr>
<tr>
<td>PD 2mm honeycomb mode 1</td>
<td>$f_{res} \ [Hz]$</td>
<td>70.03</td>
<td>70.21</td>
</tr>
<tr>
<td></td>
<td>$amp_{res} \ [dB]$</td>
<td>65.8</td>
<td>65.7</td>
</tr>
<tr>
<td></td>
<td>$\eta \ [10^{-2}]$</td>
<td>4.46</td>
<td>4.71</td>
</tr>
</tbody>
</table>

Table C.1: results of the resolution study
that 0.1 Hz is fine enough for the measurement. For this reason, the following study was measured only with the first 4 resolutions until 0.1 Hz.

In the next step, different modes of the sample with 4 mm simple particle damper were measured with these resolutions. From figure C.2, similar tendencies can be observed with the improvement of the resolutions like in figure C.1. Because these curves are mean FRF and don’t obey the FRF of a SDOF system, they were fitted with the introduced curve-fitting tool. The final data evaluated with this tool, including resonance frequencies, peak value at the resonance and loss factors are listed in table C.1. From the results it can be observed, the difference of the measured resonance frequencies and peak amplitudes are generally immaterial (less than 3%). Only the difference between the loss factors can be over 10%. Mode 1 and 2 have the greatest relative differences (15% and 23%). However, the absolute difference is very slight (0.3 [10^{-2}]). For an undamped structure, this difference is insignificant. With the increase of the frequency, the differences get smaller and the signals are more stable, because for similar loss factors, the 3-dB span is wider in higher frequencies. At mode 4 and 9, the differences are reduced to 7% and 2% respectively. Next, the difference between a poor and well damped structure was compared through mode 1 of the 4 mm simple particle damper and the 2 mm honeycomb particle damper, which has an excellent damping efficiency. On the latter one, the difference of the loss factor is only 7% (with absolute difference of 0.4 [10^{-2}]). Therefore, the different resolutions have much less influence on a better damped structure than a poor damped one. It can be concluded that various structures with different damping efficiencies have similar difference of fitted loss factors between the resolutions from 0.1 to 1 Hz. The absolute deviation of 0.3 – 0.4 [10^{-2}] is tolerable for the evaluation of damped structures. In general terms, for measuring the plates in this chapter, the differences using different scanning resolution are not critical with the use of the curve-fitting tool. Hence, all samples were measured with 1 Hz.

On the other hand, the plot of mode 2 in figure C.2 shows, the shape of the peak with low damping and coarse resolution is sometimes distorted. With the use of the fitting-tool, not only the loss factor can be assessed in a precise way, but also the distorted curve can be rectified into the right shape. The evaluated loss factor reflects finally the realistic system behavior without distortion. The value with 1 Hz is close to the value of 0.1 Hz.
Figure C.2: measured peaks in the resolution study
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Bibliography


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