

Special Topics in Structural Dynamics & Experimental Techniques, Vol. 5

Matthew Allen
Babak Moaveni
David Epp



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Proceedings of the 43rd IMAC, A Conference and Exposition on Structural Dynamics 2025



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Matthew Allen · Babak Moaveni · David Epp
Editors

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Preface

Special Topics in Structural Dynamics & Experimental Techniques represents one of twelve volumes of technical papers presented at the 43rd IMAC, A Conference and Exposition on Structural Dynamics, organized by the Society for Experimental Mechanics, and held February 10-13, 2025. The full proceedings also include volumes on Nonlinear Structures & Systems; Dynamics of Civil Structures; Model Validation and Uncertainty Quantification; Dynamic Substructuring & Transfer Path Analysis; Special Topics in Structural Dynamics & Experimental Techniques; Computer Vision & Laser Vibrometry; Dynamic Environments Testing; Sensors & Instrumentation and Aircraft/Aerospace Testing Techniques; Topics in Modal Analysis & Parameter Identification I&II; Data Science in Engineering; and Structural Health Monitoring & Machine Learning.

Each collection presents early findings from experimental and computational investigations on an important area within Structural Dynamics. *Special Topics in Structural Dynamics & Experimental Techniques* represents papers highlighting new advances and enabling technologies for Experimental Techniques, Biomedical and Bioinspired Systems, Metamaterials, Metastructures and Additive Manufacturing and Rotating Machinery.

The organizers would like to thank the authors, presenters, session organizers, and session chairs for their participation in this track.

Editor:

Matthew Allen – *Brigham Young University, UT, USA*; Babak Moaveni – *Tufts University, MA, USA*; David Epp – *Sandia National Laboratories, NM, USA*.

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Chapter 1

Using Mode Shapes from Cell Phone Videos for Machinery Health Monitoring

Shawn Richardson and Mark Richardson

Abstract A cell phone video offers a low-cost non-contacting alternative to traditional accelerometer-based methods for monitoring the health of plant operating equipment. The frequency bandwidth and video enhancement technology in modern-day cell phones have rendered them ideal for use as a non-contacting measurement device in a plantwide route-based monitoring program.

During the past 30 years, trained neural networks have been used in a variety of applications to solve problems where the number of possible solutions is overwhelming. A neural network must be trained with lots of data, and it will diagnose mechanical faults in rotating machinery more accurately as it is trained with more vibration data.

In this paper, it is first shown how the Operational Modal Analysis (OMA) mode shapes of a rotating machine are obtained by using **FRF-based** curve fitting on vibration data extracted from a cell phone video recording of a rotating machine during operation. Then, a database search method called **FaCTsTM** is used to identify various unbalance conditions of a rotating machine.

FaCTsTM functions in the same manner as a trained neural network. **FaCTsTM** uses the current mode shapes of a machine together with a shape difference indicator (SDI) to find the *closest match* of the current mode shapes with mode shapes that were previously *labeled* and *archived* in a database. **FaCTsTM** displays a bar chart of the *ten closest matches* of the current mode shapes with the labeled mode shapes, thereby defining the current mechanical condition of a machine based on its mode shapes.

Keywords Artificial Intelligence (AI) · Neural Network (NN) · Time Waveform (TWF) · Digital Fourier Transform (DFT) · Operational Modal Analysis (OMA) mode shape · Operating Deflection Shape (ODS) · Degree of Freedom (DOF) · Frames Per Second (fps) · Auto Power Spectrum (APS) · Cross Power Spectrum (XPS) · ODS-FRF (APS and Phase of an XPS) · Fault Correlation Tools (FaCTsTM) · Shape Difference Indicator (SDI)

Rotating Machine

In this paper, **FaCTsTM** is used to uniquely identify *nine different unbalance cases* of the rotating machine shown in Figure 1. The machine has a variable speed motor connected to the rotor with a rubber belt. The motor speed was adjusted so that the rotor speed was *approximately 1000 RPM throughout all the video recordings*.

Introduction

Most power plants, oil refineries, and manufacturing plants worldwide have implemented route-based machinery health monitoring programs for accessing the health of their rotating equipment. Digital vibration signals are the primary data used to detect and diagnose faults in operating equipment.

Traditionally, machine health monitoring has been done by attaching accelerometers to the surfaces of the operating equipment and acquiring vibration signals from the accelerometers with a portable digital spectrum analyzer. This method

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Fig. 1 Rotating Machine Showing Unbalance Screws Added to Its Rotors

of data acquisition is time consuming and expensive compared with a cell phone video recording. Furthermore, because it is non-contacting, a cell phone can record vibration of machine parts that are hot or inaccessible where accelerometers cannot be used.

In previous papers [3], [4] we applied various traditional signal processing methods to the **TWFs** extracted from the frames of a video to display its **ODS's** in animation. Both **TWFs** and their associated **DFTs** can be used to display either **time-based ODS's** or **frequency-based ODS's** so that a machine's deformation can be visualized at *slower speeds* with *higher amplitudes*.

In a previous paper [4], a new database search method called **FaCTsTM** was introduced, and was used to uniquely identify nine different unbalance cases of a rotating machine using only **ODS** data from the tops of the two bearing blocks on the machine. In this paper **FaCTsTM** is used to uniquely identify the same nine unbalance cases, but by using the **OMA** mode shapes of the machine.

These results demonstrate the reliability and repeatability of using **OMA** mode shapes extracted from a cell phone video for machine health monitoring. **FaCTsTM** also functions like a neural network in that it becomes more accurate as more *labeled* and *archived* **OMA** mode shapes are made available for shape comparison.

Artificial Intelligence (or Machine Learning)

Artificial Intelligence, popularly known as **AI**, uses a “*trained*” neural network (**NN**) to interpret the meaning of a set of data. Vibration data is primarily used to diagnose the health of rotating machinery. Vibration data in the form of **TWFs**, **DFTs**, **ODS-FRFs**, **ODS's** and **Mode Shapes** can be used to train an **NN**. But to accurately diagnose a mechanical fault, an **NN** must be trained with a *lot of vibration data*.

Here are a couple definitions of **AI** from a Google search on the Internet.

- A neural network is a machine learning model that uses a network of interconnected nodes, or artificial neurons, to process data in a way that mimics the human brain.
- A neural network is a method in artificial intelligence (**AI**) that teaches computers to process data in a way that is inspired by the human brain. It is a type of machine learning (**ML**) process, called deep learning, that uses intercon-

nected nodes or neurons in a layered structure that resembles the human brain. Machine Learning (ML) using an NN mimics the learning of the human brain. An NN is depicted in Figure 2

- ***Lots of labeled input data*** is required to train an NN
- To diagnose mechanical faults, an NN must be trained with data that is ***uniquely associated with a mechanical fault***
- When vibration data is input to an Inference Engine (a trained NN), it diagnoses a mechanical fault, as shown in Figure 3.

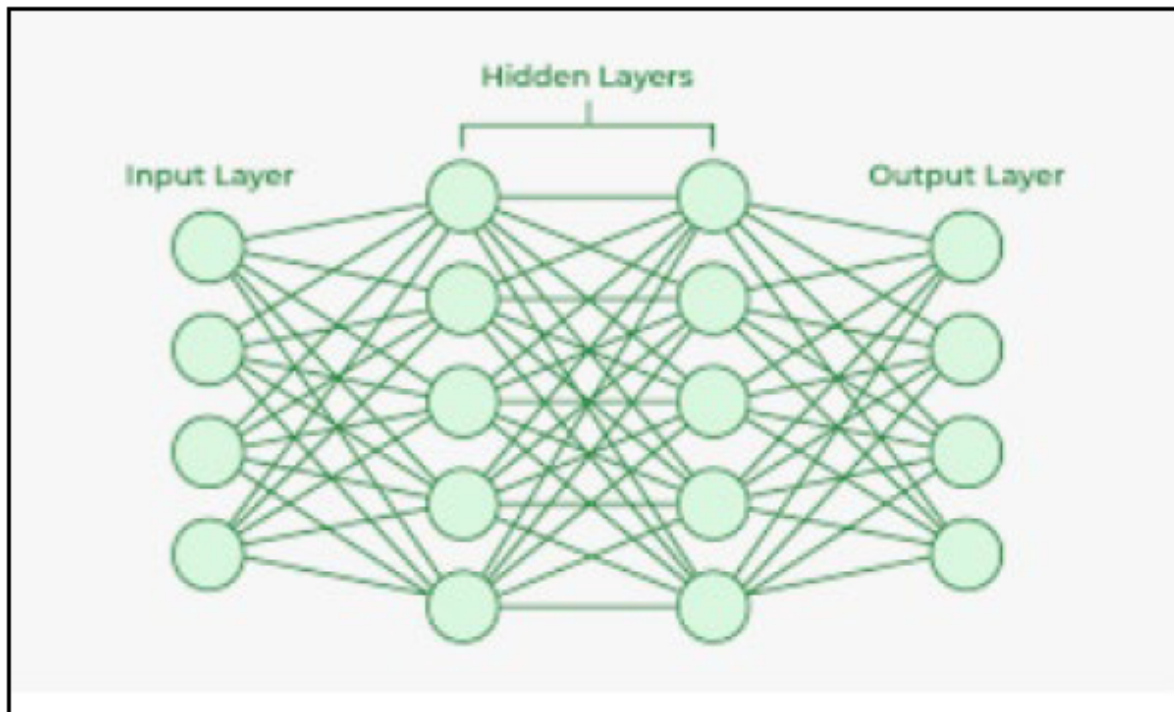


Fig. 2 A Neural Network

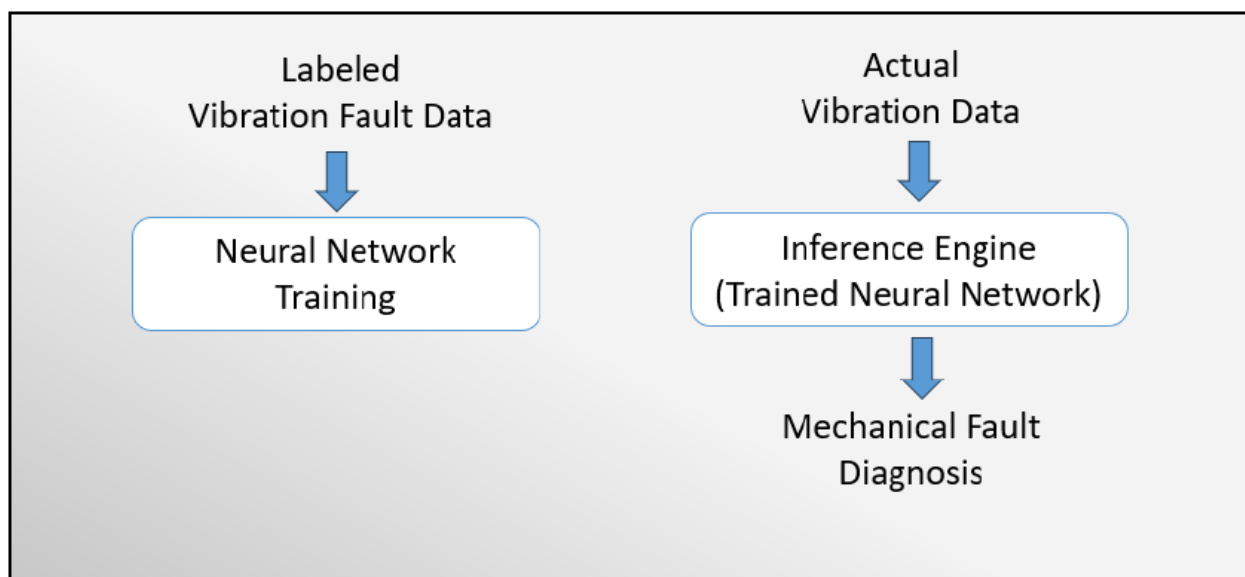


Fig. 3 NN Training & Inference Engine

TWFs & DFTs

When a video is processed in MEscape [8], a rectangular grid of points with rectangular surfaces between them is created. Frames of the video are attached to this surface during animated display of **ODS's** extracted from the video. Using a rectangular point grid, *millions of pixels* in each frame of a cell phone video are processed to extract **TWFs** for the horizontal & vertical motion of *thousands of points* in the point grid. Grid points with *little or no motion*, (like background points), are hidden and their linked **TWFs** are removed from further analysis. A point grid with background points hidden is shown in Figure 4.

A **DFT** is calculated for each **TWF** that is extracted from the video. Time-based **ODS's** are displayed in animation from the **TWFs** using a sweeping Line cursor. Frequency-based **ODS's** are displayed in animation from the **DFTs** using sine dwell modulation of the **ODS** at the cursor position. The **magnitude & phase** of the **ODS** at selected points can also be displayed, as illustrated in Figure 4.

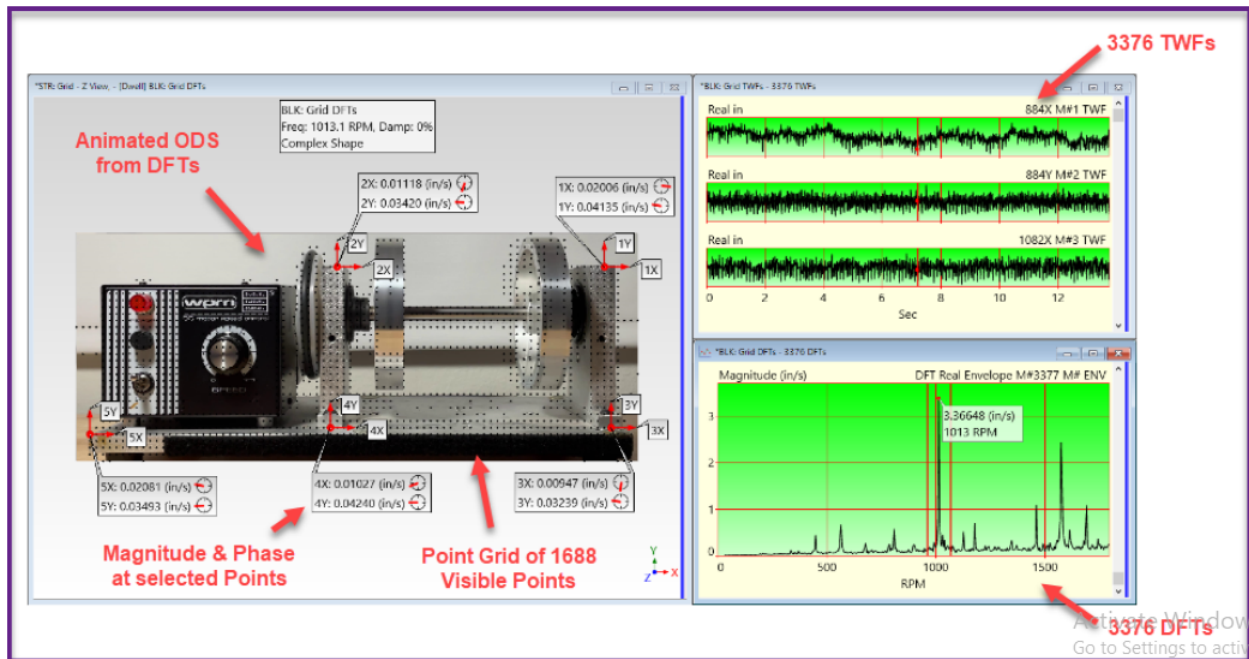


Fig. 4 First-Order ODS Animated from DFTs

ODS-FRFs

A unique frequency domain function, called an **ODS-FRF**, can be calculated from each response **TWF** extracted from a video. A set of **ODS-FRFs** calculated from the **TWFs** is typically more accurate because *spectrum averaging* can be used to reduce extraneous noise from the **ODS-FRFs**.

The **magnitude** of an **ODS-FRF** is the **APS** of the **response DOF** at a grid point. The **phase** of the **ODS-FRF** is the **phase of the XPS** between the **response DOF** and the **DOF** of any **reference** grid point.

ODS-FRFs carry the *same displacement units* as the response **TWFs** from which they are calculated. But because it is a frequency domain function, an **ODS-FRF** can be *accurately differentiated to velocity units* by multiplying it by the frequency variable.

Vibration in *velocity units* is commonly used by vibration analysts to quantify vibration levels in rotating equipment.

Law of the FFT

One of the laws of the FFT algorithm is that $\Delta f = 1/T$, where Δf is the *frequency difference between samples* of an ODS-FRF, and T is the *time length* of TWF data from which the ODS-FRF was calculated. For example, if an ODS-FRF is calculated from TWF data over a **15 second** period, the frequency resolution (Δf) of the ODS-FRF is $60/15 = 4$ RPM.

To increase the frequency resolution of an ODS-FRF, TWF data over a *longer period* T is required. Therefore, the video from which the TWFs are extracted must be recorded over a *longer period* T .

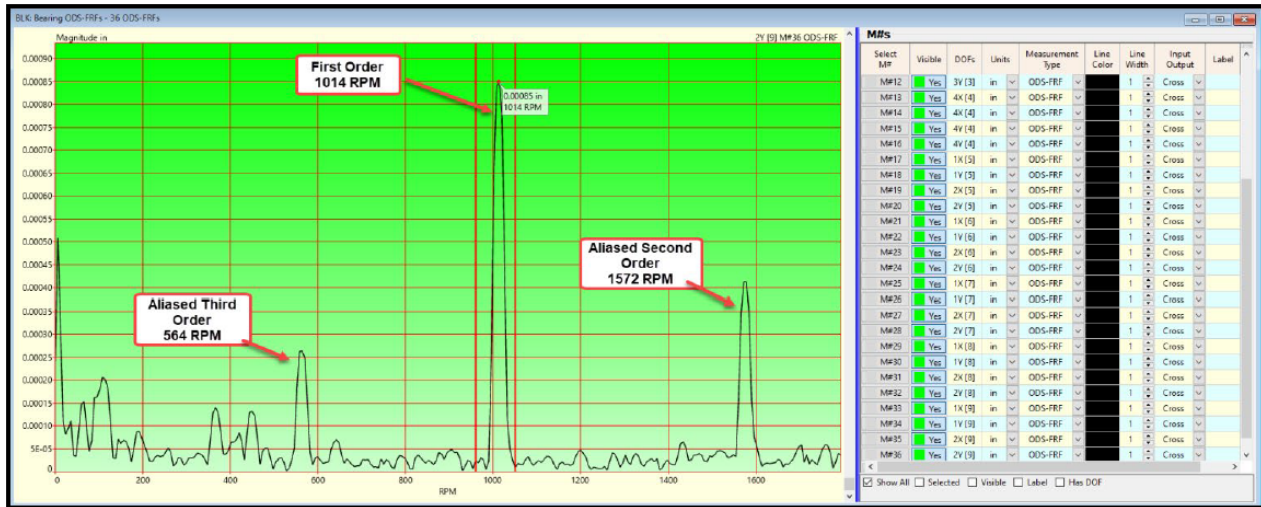


Fig. 5 ODS-FRF Showing First Three Order Peaks.

Aliased Order Peaks

A limitation of any video recording is that *anti-alias filtering cannot be used* to remove high-frequency signals from the video.

Without anti-alias filtering, machine order peaks *greater than one-half the sampling frequency*, (called F_{max}), are *folded around* F_{max} and appear at lower frequencies in the ODS-FRFs, as shown in Figure 5.

All order-related resonance peaks between F_{max} & $2 \times F_{max}$ are folded around (*wrapped around*) F_{max} and appear at a lower frequency in the frequency band (0 to F_{max}). Aliasing of higher frequencies occurs in both DFTs and ODS-FRFs.

The ODS-FRF shown in Figure 5 was calculated from a TWF that was extracted from a video that was sampled at **60 fps**, or **3600 RPM**. Therefore, the F_{max} of the ODS-FRF is **1800 RPM**. The first-order peak is at the machine running speed and is clearly visible at **1014 RPM**. The second order resonance peak should be at **2028 RPM** and third order resonance peak should be at **3042 RPM**, but they are both *folded around* **1800 RPM** and are clearly visible *at lower frequencies* in the ODS-FRF.

The aliased frequency of an order higher than F_{max} can be calculated from F_{max} and the expected order frequency.

- **Second order** aliased frequency $1800 - (2028 - 1800)$ **1572 RPM**
- **Third order** aliased frequency $1800 - (3042 - 1800)$ **558 RPM**

The **ODS-FRF** in Figure 5 was calculated from a **TWF** with a *10-second length T*. Therefore $\Delta f = (1/10)$ Hz or **60/10 = 6 RPM**. So, the aliased frequency of the third order peak is *within one* (Δf) of its calculated value.

Curve Fitting the ODS-FRFs

ODS-FRFs can be curve fit using an **FRF-based** curve fitter if they have been filtered with a special window. In MEscape [9] this filter is called a **DeConvolution window**. This filter reshapes an **ODS-FRF** so that it closely resembles an **FRF** and therefore can be curve fit using **FRF-based** curve fitting. An **FRF-based** curve fit of the three order peaks in an **ODS-FRF** is shown in Figure 6. In the upper left-hand graph, the **red curve fitting function** is overlaid on a magnitude plot of the **ODS-FRF**.

The aliased frequencies of the second and third orders are correctly estimated by curve fitting the **ODS-FRF**. More details on curve fitting **ODS-FRFs** are given in a companion paper [1].

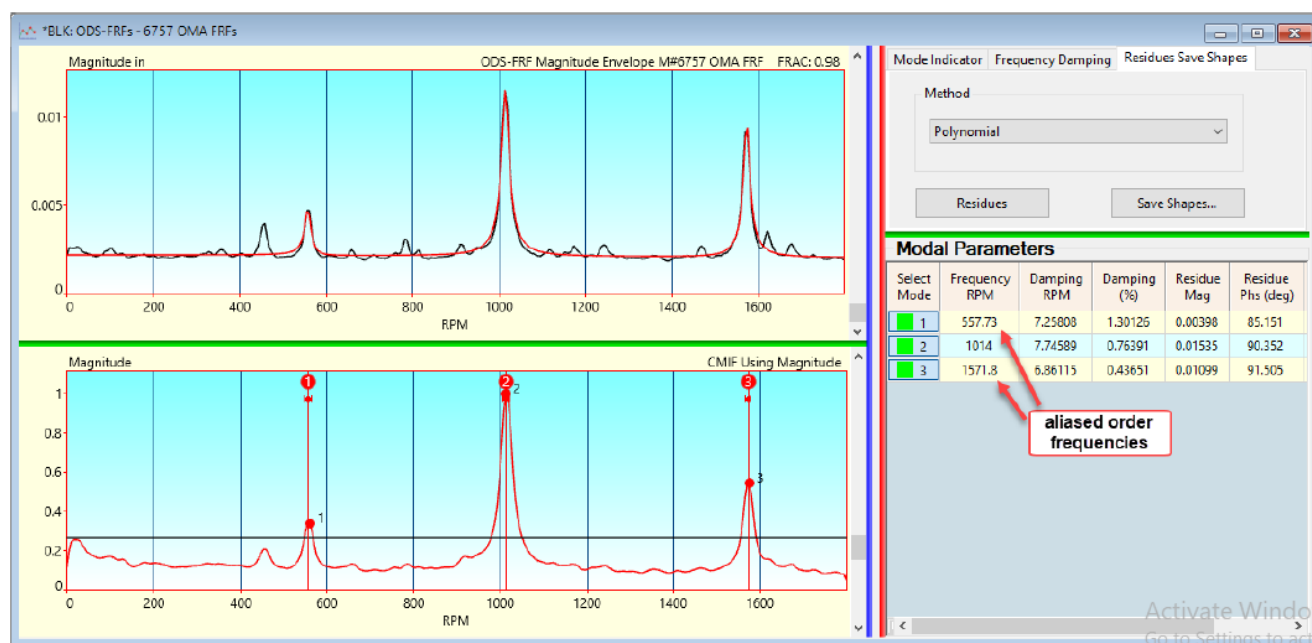


Fig. 6 Curve-fit of ODS-FRFs.

Mode Shapes at Monitored Points

Although curve fitting was applied to all the **ODS-FRFs** calculated from the videos, only the mode shape components are **five grid points** were labeled and archived in the machine database. These five points are shown in Figure 7.

Damping Removal

DeConvolution windowing is necessary before using an **FRF-based** curve fitter on a set of **ODS-FRFs**. But DeConvolution windowing *adds a specific amount of damping* to each **OMA** mode shape. Therefore, following **FRF-based** curve fitting, when the **OMA** mode shapes are stored into a Shape Table in MEscape [9], the damping added by DeConvolution windowing is removed from them. This is shown in Figure 8.

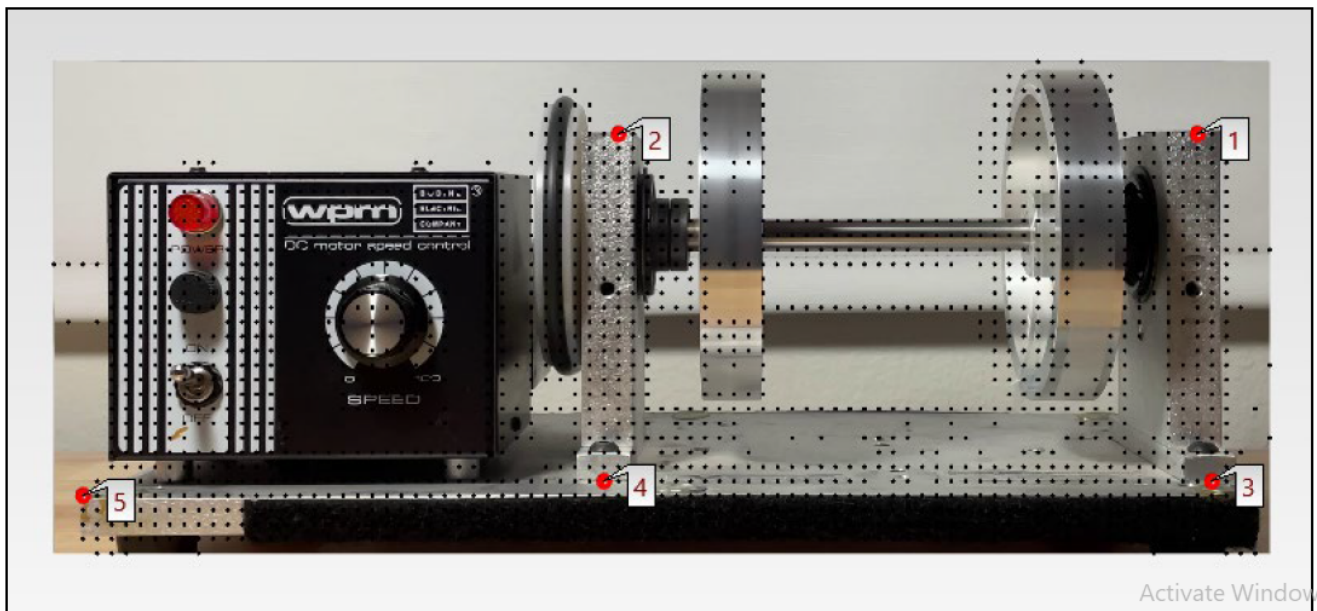


Fig. 7 Point Grid Showing Monitored Points.

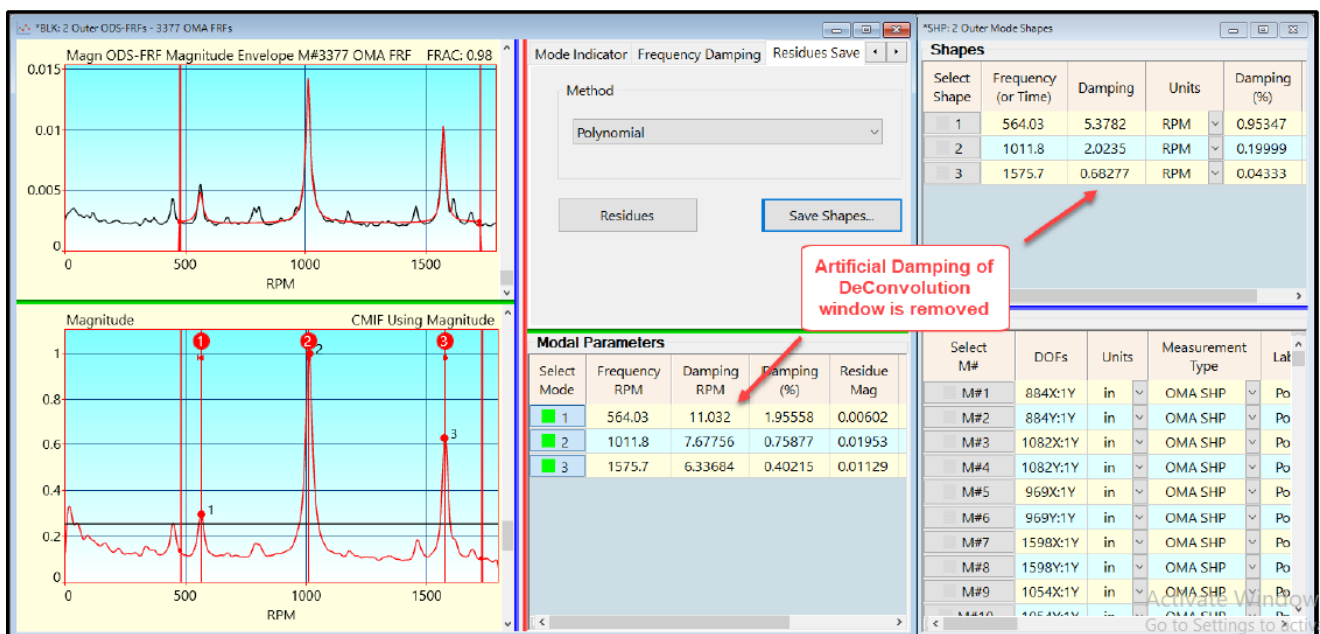


Fig. 8 Damping is Removed when OMA Mode Shapes are Saved.

FaCTs™

FaCTs™ [4] is a database search algorithm used by MScope [9]. FaCTs searches a database of *labeled* mode shapes, each *mode shape* associated with *a particular machine fault*. When a new *mode shape* is saved into the archival database, FaCTs searches the database of *labeled* mode shapes and displays a bar chart of the *ten closest matching mode shapes* together with the mechanical fault associated with the *labeled* mode shapes.

FaCTsTM uses an algorithm called the **Shape Difference Indicator (SDI)** [8]. **SDI** calculates a correlation coefficient between two complex-valued shape vectors. **FaCTs** finds the *ten closest matching* mode shapes in the archival database based on the **SDI** value between each *current* mode shape and each *labeled* mode shape.

- **FaCTs** has values **between 0.0 and 1.0**
- **FaCTs** = **1.0** → two mode shapes are *identical*
- **FaCTs** > = **0.9** → two mode shapes are *similar*
- **FaCTs** < **0.9** → two mode shapes are *different*

Baseline Case

When no unbalance screws were added to either rotor of the rotating machine in Figure 1, its mode shapes were labeled as the Baseline case. When the Baseline case is archived into the database, the **FaCTs** bar chart in Figure 9 clearly identifies it by its unique **OMA** mode shapes compared to the mode shapes of other unbalance cases. Its **FaCTs** bars with all the other unbalance cases are *much less than 1.0*.

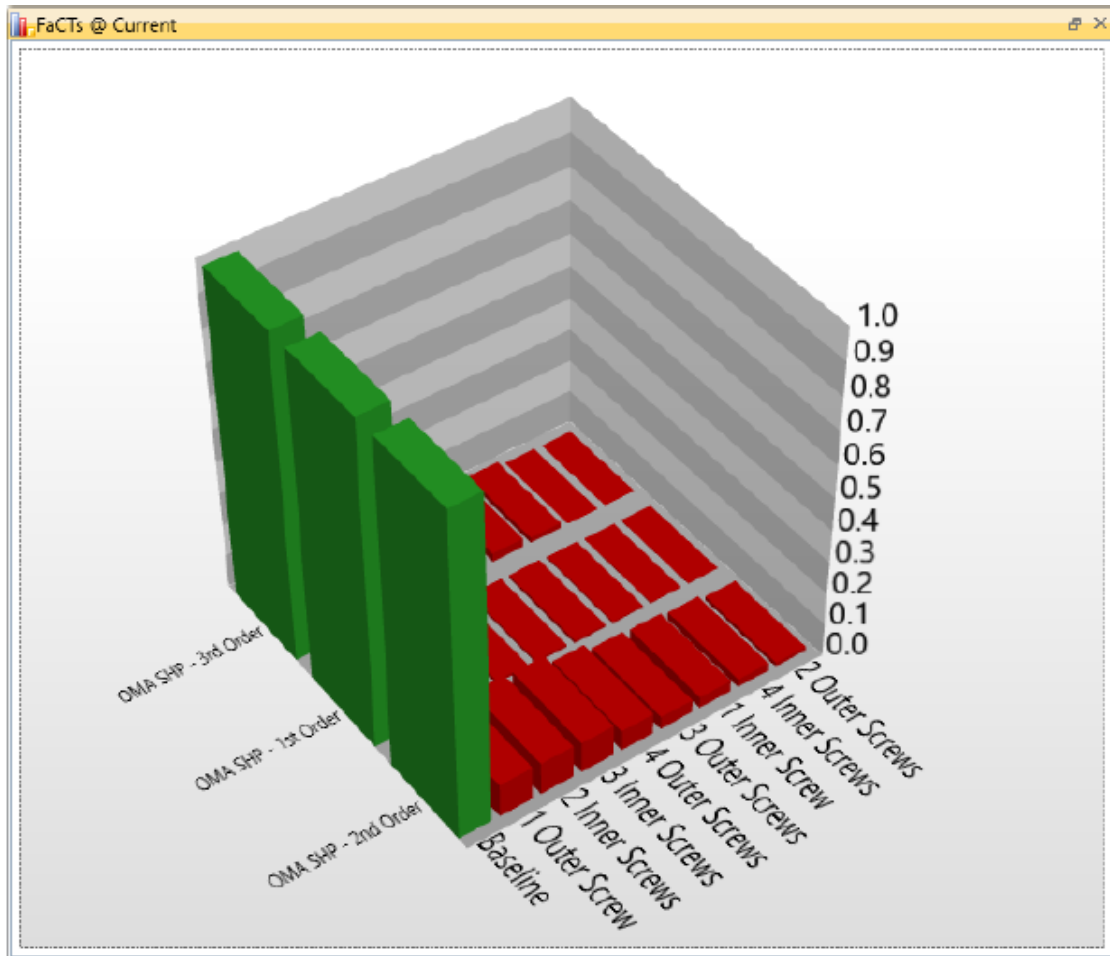


Fig. 9 Baseline OMA Mode Shapes Versus Other Eight Unbalance Cases

Figures 10 through 18 show the **FaCTs** bar charts for the *eight unbalance cases* where screws were added to either the Inner or Outer rotor. Each unbalance case was *uniquely identified* by **FaCTs** because its corresponding **OMA** mode shapes were unique when compared to the **OMA** mode shapes of the other unbalance cases. But *there is one exception*. Unbalance cases with **2 Inner screws** and **3 Inner screws** have a **FaCTs** bar of **1.0**, indicating that the **OMA** mode shapes for these two cases *are essentially the same*.

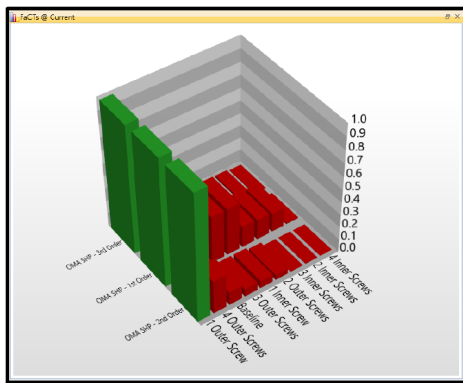


Fig. 10 One Outer Screw

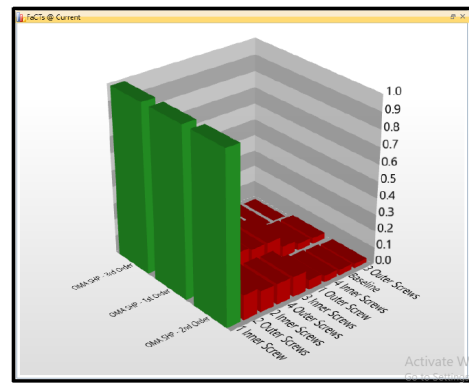


Fig. 15 One Inner Screw

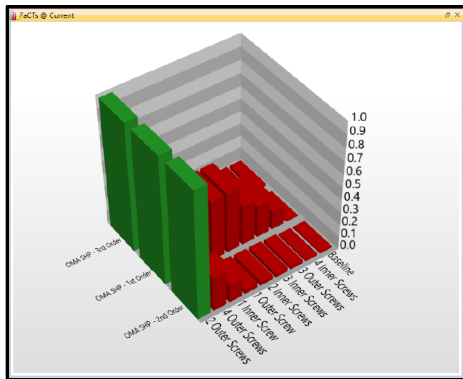


Fig. 11 Two Outer Screws

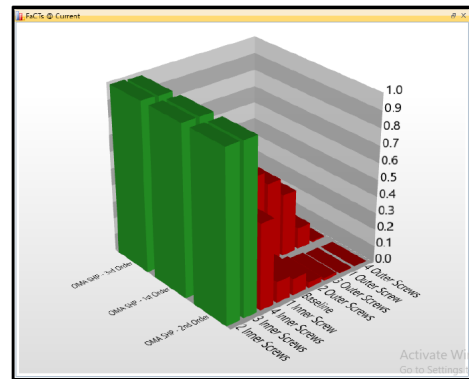


Fig. 16 Two Inner Screws

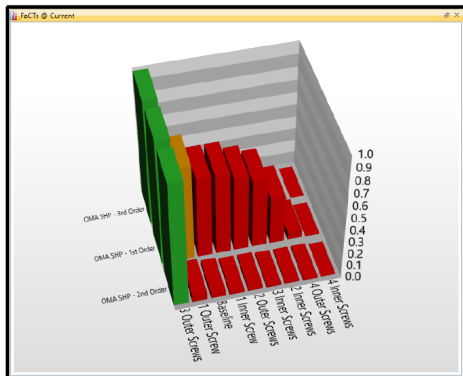


Fig. 13 Three Outer Screws

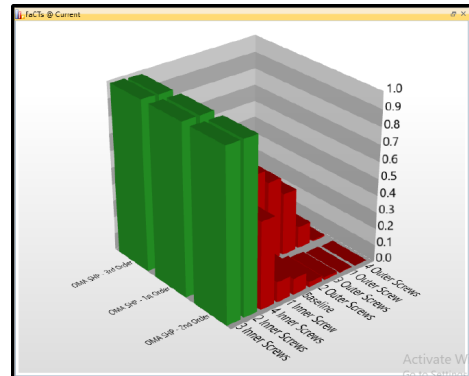


Fig. 17 Three Inner Screws

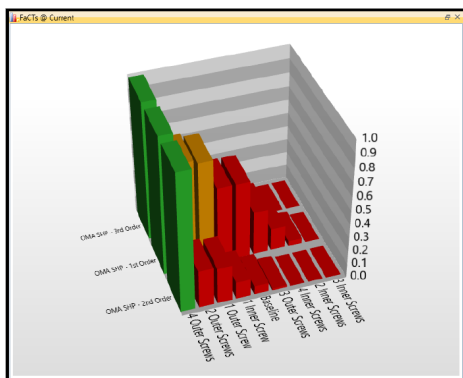


Fig. 14 Four Outer Screws

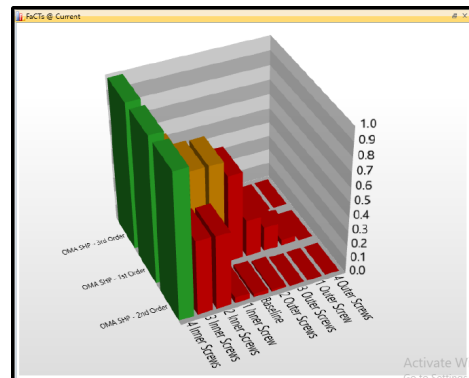


Fig. 18 Four Inner Screws

Conclusions

Nine different unbalance cases were created on a rotating machine by adding screws to its *Inner* and *Outer* rotors. The first case with *no screws added* was labeled as the **Baseline** case. In each case, a **16 second** cell phone video was recorded, with the machine running at approximately **1000 RPM**.

Using MEScope [9], **TWFs** were extracted from each cell phone video and **ODS-FRFs** were calculated from the **TWFs** for each unbalance case. To reduce noise in the **ODS-FRFs**, five *spectrum averages* and *overlap processing* were used to calculate the **ODS-FRFs**. Then the **ODS-FRFs** were curve fit using **FRF-based** curve fitting and the **OMA** mode shapes for the *first three machine orders* were labeled and stored in an archival database.

The **first order OMA** mode shape and the mode shapes of the **aliased second** and **third orders** were archived. Each set of three mode shapes was labeled with its *corresponding unbalance case*.

Then, when the **OMA** mode shapes for each case were again stored into the archival database, **FaCTs** correctly identified each case by calculating its **SDI** value with each *labeled* set of **OMA** mode shapes in the database. This method of numerically comparing a *current set* of **OMA** mode shapes with sets of *labeled OMA* mode shapes uniquely identified each of the nine unbalance cases using only *ten mode shape components* from five points on the video point grid of the machine.

This low-cost approach using cell phone videos and **FaCTs** can be used by any plant maintenance department for monitoring the health of its rotating equipment and accurately identifying machine faults.

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Chapter 2

Rotordynamic Analysis of the Turbopump Using Superelement Approach

Yunus Ozcelik, Hakan Bayraktar, and Yunus Tufek

Abstract The use of turbopumps in aerospace propulsion systems and high-performance fluid transport applications necessitates thorough rotordynamic analysis to ensure efficiency and reliability. The complexity of turbopumps, with factors like high speeds, fluid-structure interactions, and flexible components, requires advanced modeling techniques for accurate predictions. Model reduction techniques like Component Mode Synthesis (CMS) are crucial for efficient dynamic analysis of complex systems. This study analyzed the rotordynamic behavior of the turbopump using a superelement approach. The rotor was modeled using 3D elements in ANSYS, with its dynamic characteristics evaluated under constant bearing stiffness. To enhance computational efficiency, the stator was represented as a superelement and coupled with the 3D rotor model. For validation, a full 3D stator model was also developed and coupled with the rotor. This comprehensive modeling strategy enabled a detailed assessment of the system's dynamic behavior. Results validated against the full rotor–stator model demonstrated that stator flexibility has a notable effect on the rotor's critical speeds. The study highlights the effectiveness of the superelement method in reducing computational costs while maintaining the accuracy required for high-fidelity rotordynamic analysis.

Keywords Rotordynamic · Turbopump · Component Mode Synthesis · Superelement

Introduction

Turbopumps are critical subsystems in aerospace propulsion systems, commonly used in liquid rocket engines and other high-performance fluid transport applications. They consist of a pump driven by a turbine, which operates at extremely high rotational speeds to deliver the required power. Due to the high operating speeds and complex structural interactions, turbopumps are subject to various dynamic phenomena that can significantly affect their performance and reliability.

One of the most crucial aspects of turbopump design and analysis is rotordynamic, which examines the dynamic behavior of the rotating components, including the rotors, bearings, seals, and the surrounding casings. Rotordynamic analysis is essential for predicting critical speeds, stability margins, and potential failure modes such as rotor whirl and resonance, which can lead to excessive vibrations and mechanical failure. A thorough understanding and optimization of turbopump rotordynamic behavior are key to ensuring efficient, stable, and reliable operation under a wide range of operating conditions.

The high rotational speeds of turbopumps, coupled with the presence of fluid-structure interactions, bearings, seals, and flexible stator components, contribute to the complexity of their rotordynamic analysis. Factors such as rotor flexibility, bearing stiffness, and damping characteristics, as well as seal forces and casing flexibility, all play a role in determining the system's dynamic response. Identifying and addressing rotordynamic issues early in the design phase is crucial to avoiding costly failures and ensuring smooth operation throughout the entire operating range.

A rotor system must be designed to operate without excessive vibration throughout its entire speed range. The theoretical foundation of rotordynamic analysis dates to the late 18th century when Dunkerly [1] proposed that the critical speed of a rotor corresponds to the natural frequency of a simply supported beam. Early design approaches assumed rigid bearings. However, as operating speeds increased and computational methods advanced, rotor dynamics analysis evolved to account for bearing properties. It is well-established that the structure beyond the bearings, known as the foundation, tends to reduce

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the first critical speed frequency while increasing the amplification factor [2]. These studies demonstrate that considering both bearing and foundation effects in rotor dynamics analysis leads more accurate rotor dynamics predictions.

Rotordynamic analysis using one-dimensional (1D) elements, where the shaft is represented by beam or bar elements and the disks are idealized as a concentrated mass, is widely employed in industry due to its simplicity and computational efficiency [3]. However, this approach has notable limitations, particularly in its inability to model complex rotor geometries or accurately capture the effects of large flexible disks and blades. For axisymmetric rotors, two-dimensional (2D) axisymmetric harmonic elements can be used, as they consider the three-dimensional (3D) nature of the problem while reducing the number of elements needed for modeling [4].

With advancements in modeling, analysis, optimization techniques, and material science, modern rotor configurations frequently incorporate non-uniform shaft diameters, hollow components, and large flexible structures. These features necessitate more sophisticated modeling techniques to accurately predict their dynamic behavior. 2D axisymmetric harmonic elements are insufficient for representing complex, non-uniform rotor blade geometries. This limitation has led to the increased use of 3D solid and shell elements, which offer a more accurate and geometrically complete representation of the rotor. These elements allow for analysis in both the fixed and rotating reference frames, reducing the need for geometric simplifications that remain prevalent in industrial practice.

Modeling rotors with 3D shell and solid elements offers the advantage of accurately representing complex rotor geometries. However, 3D rotor models are more suitable for analysis in the rotating reference frame and tend to result in significantly larger problem sizes [5]. Despite advancements in computational resources, linear rotordynamic analyses can still take hours or even days to complete [6]. To integrate such models into the commercial engineering design of critical components, and to account for nonlinear effects from bearings and dampers, it is crucial to improve dynamic analysis performance by orders of magnitude. Crucially, this improvement must be achieved without sacrificing model complexity or solution accuracy.

The application increase in model size for rotors modeled with 3D elements necessitates the use of model reduction techniques. Component Mode Synthesis (CMS) is a model reduction technique that allows for the efficient dynamic analysis of large and complex mechanical systems by breaking them down into smaller, manageable substructures. These substructures are individually analyzed, and their dynamic properties (e.g., mass, stiffness, and damping) are characterized through modal parameters such as natural frequencies and mode shapes. CMS enables the recombination of these modes to form a reduced-order model for the entire system, significantly decreasing the computational load while preserving the essential dynamic characteristics.

The use of CMS to reduce the computational effort for analyzing large rotating machinery, particularly in turbochargers, has been demonstrated by Kim et al. [7], while maintaining accuracy in predicting critical speeds and response amplitudes. CMS methods have also been applied to analyze the dynamic behavior of rotor-bearing systems under unbalance forces, enabling the inclusion of nonlinear effects like bearing stiffness and damping [3]. CMS was employed to investigate the dynamics of a turbopump rotor system in a study conducted by Lund [8]. The research highlighted the method's ability to predict natural frequencies and mode shapes accurately, facilitating the identification of critical speeds that could lead to resonance. Similarly, the superelement method has been used in the design optimization of rotor-bearing systems to predict how component modifications would affect overall system behavior, as shown in a study by Ehrich [9]. The combination of CMS with nonlinear effects has been gaining traction in recent years. CMS with nonlinear rotordynamic phenomena, such as oil whip and shaft bending, was integrated, demonstrating the method's versatility in capturing complex real-world dynamics that are difficult to handle with traditional full-order models, by Gupta et al. [10].

In this study, the rotordynamic behavior of a turbopump, including both the rotor and stator, was investigated using a combination of 3D modeling and superelement techniques obtained through the CMS method. The rotor was initially modeled with 3D finite elements in ANSYS, and its dynamic characteristics were evaluated under the assumption of constant bearing stiffness. To further enhance the analysis, the rotor was coupled with a stator represented as a condensed superelement, simplifying the stator's influence on the rotordynamic. Furthermore, a full 3D model of the stator was created in ANSYS and coupled with the rotor to validate the reduced model. This detailed approach provided an in-depth analysis of how the interaction between the rotor and stator influences the critical speeds of the turbopump.

Background

Substructuring is a technique that condenses a group of finite elements into a single equivalent element, represented by a reduced-order matrix. This condensed element is referred to as a superelement. In analytical procedures, a superelement is treated comparably to any other element type, with the exception that it necessitates an initial substructure generation analy-

sis for its creation. Substructuring significantly reduces computational effort and solution time of large-scale problems while reducing the requirement for computational resources. This method proves particularly beneficial in nonlinear analyses and in examinations involving structures with recurring geometric configurations, such as rotordynamic analysis. In nonlinear analysis, substructuring can be implemented in the linear sections of the model, thereby eliminating the necessity for repetitive recalculation of element matrices during each equilibrium iteration. For structures characterized by repetitive patterns, a singular superelement can be generated to represent the recurring geometry and replicated across various locations, resulting in substantial savings in computational time.

In substructuring analysis, the nodal displacement vector $\{u\}$ of a substructure is expressed in terms of reduced coordinates $\{\hat{u}\}$, by following transformation [11]

$$\{u\} = [T] \{\hat{u}\}, \quad (1)$$

where $[T]$ is the transformation matrix.

In dynamic analysis, equation of motion can be expressed by,

$$[M] \{\ddot{u}\} + [C] \{\dot{u}\} + [K] \{u\} = \{F\}, \quad (2)$$

where $[M]$, $[C]$ and $[K]$ are mass, damping and stiffness matrix, respectively and $\{F\}$ is the load vector.

By substituting Eq. (1) into Eq. (2) and left multiplying by $[T]^T$, the following reduced equation is obtained,

$$[\hat{M}] \{\ddot{\hat{u}}\} + [\hat{C}] \{\dot{\hat{u}}\} + [\hat{K}] \{\hat{u}\} = \{\hat{F}\} \quad (3)$$

where, $[\hat{M}] = [T]^T [M] [T]$ is the reduced mass matrix, $[\hat{C}] = [T]^T [C] [T]$ is the reduced damping matrix, $[\hat{K}] = [T]^T [K] [T]$ is the reduced stiffness matrix and $\{\hat{F}\} = [T]^T \{F\}$ is the reduced load vector.

Static condensation, often referred to as Guyan Reduction, and Component Mode Synthesis (CMS) are both widely used techniques for substructuring in structural dynamics. Guyan Reduction is typically insufficient to fully capture the dynamic behavior of a superelement. To address this, static condensation is enhanced with the inclusion of component modes and residual vectors. When applying CMS to reduce a complex engineering model, certain loading conditions may induce deformation states that cannot be accurately represented by static reduction and component modes alone. In such cases, the Residual Vector technique is used to incorporate high-frequency contributions, compensating for this deformation discrepancy and ensuring an accurate dynamic response across all loading conditions. In this study, CMS is utilized to model the stator component of the turbopump. By applying equilibrium and compatibility conditions at the component interfaces, the method defines the dynamic behavior of the entire system model.

Dividing the matrix equation into degrees of freedom (DOFs) for the interface and interior,

$$\begin{aligned} \{u\} &= \begin{Bmatrix} \{u_m\} \\ \{u_c\} \end{Bmatrix}, \quad [M] = \begin{bmatrix} [M_{mm}] & [M_{mc}] \\ [M_{cm}] & [M_{cc}] \end{bmatrix}, \quad [C] = \begin{bmatrix} [C_{mm}] & [C_{mc}] \\ [C_{cm}] & [C_{cc}] \end{bmatrix}, \\ [K] &= \begin{bmatrix} [K_{mm}] & [K_{mc}] \\ [K_{cm}] & [K_{cc}] \end{bmatrix}, \quad \{F\} = \begin{Bmatrix} \{F_m\} \\ \{F_c\} \end{Bmatrix} \end{aligned} \quad (4)$$

where, m is the master DOF that defined at interference, and c is all nodes except master nodes.

Nodal displacement vector can be expressed by,

$$\{u\} = \begin{Bmatrix} \{u_m\} \\ \{u_c\} \end{Bmatrix} = [T] \begin{Bmatrix} \{u_m\} \\ \{y_\delta\} \end{Bmatrix} \quad (5)$$

where, $\{y_\delta\}$ is a truncated set of generalized modal coordinates. Transformation matrix used in this study is obtained by [12],

$$[T] = \begin{bmatrix} [I] & [0] \\ [G_{cm}] & [\Phi_c] \end{bmatrix} \quad (6)$$

where, $[\Phi_c]$ is the eigenvector obtained with interference nodes fixed.

In the reduced system, master DOFs serve to connect the CMS superelement with other elements or additional CMS superelements.

For the fixed-interface method, when the fixed-interface normal modes are mass-normalized, the resulting reduced stiffness, mass, and damping matrices, as well as the reduced load vector, take the following final form [13]:

$$[\hat{M}] = \begin{bmatrix} [M_{mm}] + [M_{mc}] [G_{cm}] + [G_{mc}] [[M_{cm}] + [M_{cc}] [G_{cm}]] & [[M_{mc}] + [G_{mc}] [M_{cc}] [[\Phi_c]] \\ [\Phi_c]^T [[M_{cm}] + [M_{cc}] [G_{cm}]] & [I] \end{bmatrix} \quad (7)$$

$$[\hat{C}] = \begin{bmatrix} [C_{mm}] + [C_{mc}][G_{cm}] + [G_{mc}][C_{cm}] & [[C_{mc}] + [G_{mc}][C_{cc}]] [\Phi_c] \\ [\Phi_c]^T [[C_{cm}] + [C_{cc}][G_{cm}]] & [\Phi_c]^T [C_{cc}] [\Phi_c] \end{bmatrix} \quad (8)$$

$$[\hat{K}] = \begin{bmatrix} [K_{mm}] + [K_{mc}][G_{cm}] & [0] \\ [0] & [\Lambda^2] \end{bmatrix} \quad (9)$$

$$\{\hat{F}\} = \begin{Bmatrix} \{F_m\} + [G_{mc}]\{F_c\} \\ [\Phi_c]^T \{F_c\} \end{Bmatrix} \quad (10)$$

where, $[G_{cm}] = -[K_{cc}]^{-1} [K_{cm}]$, $[G_{mc}] = [G_{cm}]^{-T}$ and, $[\Lambda^2]$ is a diagonal matrix consisting of the eigenvalues of the retained fixed-interface normal modes.

The displacements at the condensed or DOFs are obtained from Eq. (5). For the fixed-interface condition, they are computed using Craig-Bampton method as follows:

$$\{u_c\} = [G_{cm}] \{u_m\} + [\phi_c] \{y_\delta\} \quad (11)$$

In rotordynamic analysis, the dynamic equation in a stationary reference frame can be expressed as:

$$[M] \{\ddot{u}\} + ([G] + [C])\{\dot{u}\} + ([B] + [K])\{u\} = \{F\}, \quad (12)$$

where, $[G]$ is the gyroscopic matrix which plays a critical role in rotordynamic analysis as it depends on the rotational velocity and, $[B]$ is the rotating damping matrix.

When a structure rotates about an axis such as Z and a precession motion (a rotation about an axis perpendicular to Z Axis) is introduced, a reaction moment arises. This reaction, known as the gyroscopic moment, acts along an axis that is perpendicular to both the spin axis Z and the precession axis. The gyroscopic matrix $[G]$ represents this effect by coupling degrees of freedom in planes that are perpendicular to the spin axis in finite element modeling. Notably, this matrix is skew-symmetric, reflecting the nature of gyroscopic forces.

Whirling refers to the circular or elliptical motion exhibited by a rotating structure when it vibrates at its resonant frequency. This motion can occur in two distinct forms: forward whirl (FW), which occurs when the motion is in the same direction as the rotational velocity, and backward whirl (BW), which takes place when the motion opposes the direction of rotation. Both forms of whirling are critical in understanding the dynamic behavior of rotating systems, as they can influence stability and performance, particularly at resonance.

The critical speed refers to the rotational speed at which a structure's resonance frequency (or frequencies) coincides with the excitation frequency. This condition arises when the natural frequency of the system matches the frequency of the applied excitation, which can result from an unbalance that is synchronous with the rotational speed. To identify critical speeds, one effective approach is to conduct a Campbell diagram analysis. This involves calculating the intersection points between the frequency curves and the excitation line, which reveal the critical speeds of the system. This study focuses exclusively on the effect of unbalance forces synchronous with the rotational speed.

Methodology

In this study, the rotordynamic behavior of a turbopump, including the stator, was analyzed using a superelement approach. The rotor was modeled using 3D elements in ANSYS, and its dynamic characteristics were evaluated under the assumption of constant bearing stiffness. Subsequently, the rotor model was coupled with a superelement representing the entire stator. Additionally, the rotor was connected to the full stator model, which was also constructed using 3D elements in ANSYS, allowing for a comprehensive analysis of the system's dynamic performance.

Rotordynamic Model with Fixed Bearing Stiffness

A simplified model of the turbopump rotor, excluding the casing, was created using constant stiffness properties. Figure 1 illustrates a rotor configuration for the analysis, showing several rotating components arranged sequentially from the left end of the shaft: nut, front inducer, front impeller, front bearing and rear inducer, rear impeller, rear bearing and turbine. Additionally, the lumped mass model of the rotating non-axisymmetric components, such as blades of impeller, inducer and turbine are also represented in Figure 1. Since the objective of this study is to evaluate the impact of using a superelement on solution time and accuracy, the stiffness and damping effects of the seals are excluded from the scope of the analysis.

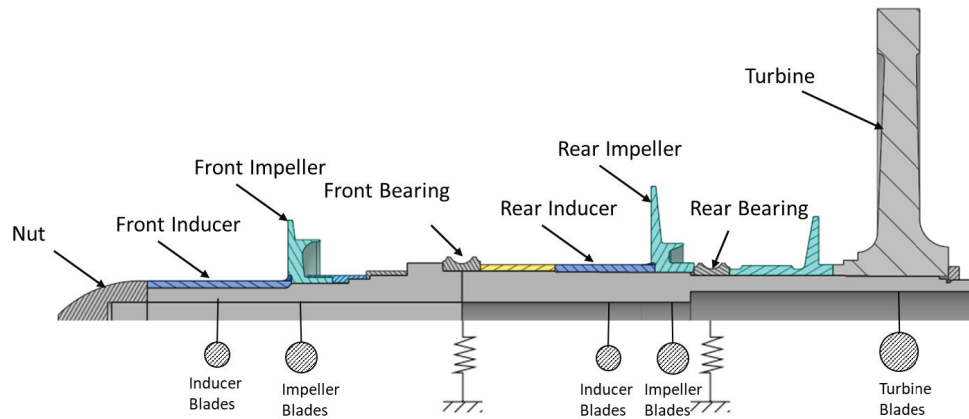


Fig. 1 The Turbopump Rotor Configuration.

The turbopump, powered by a turbine using hot, high-pressure gas, operates at a rotational speed of 23,500 RPM under steady-state conditions. It is assumed that both the front and rear bearings are isotropic with a stiffness of 10^8 N/mm. Inconel 718 superalloy, with a density of 8240 kg/m^3 , a Young's modulus of 200 GPa, and a Poisson's ratio of 0.3, was assigned to rotor components. As illustrated in Figure 2, the rotor model was discretized with 105,344 SOLID185 elements, which are first-order hexahedral elements in ANSYS. Bonded contacts were employed at all component interfaces to simulate interference. Additionally, COMBI214 elements were utilized to represent the bearings within the model.

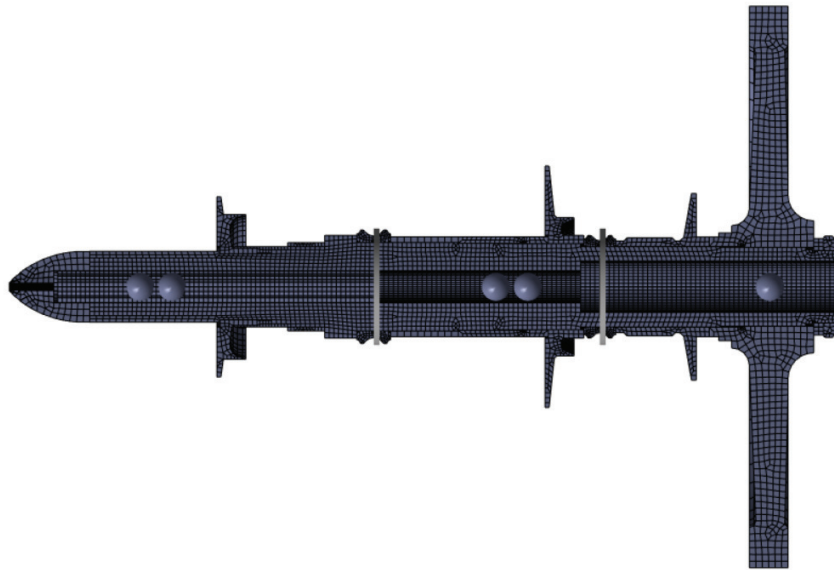


Fig. 2 Cross-sectional View of the 3D Finite Element Model of the Turbopump Rotor.

A complex eigenvalue analysis was conducted to generate the Campbell diagram over a range of rotor speeds (5,000-30,000 RPM), as shown in Figure 3. In this diagram, the 1X line denotes the rotor's operating speed. Figure 4 illustrates the first backward (1st BW) and first forward (1st FW) mode shapes of the rotor when operating at a speed of 5,000 RPM. These mode shapes provide insight into the rotor's dynamic behavior under these conditions. The overhang configuration of the turbine, along with its large diameter and mass, causes the first bending mode to predominantly involve shaft deflection on the turbine side, while the other rotor components remain largely unaffected and stationary.

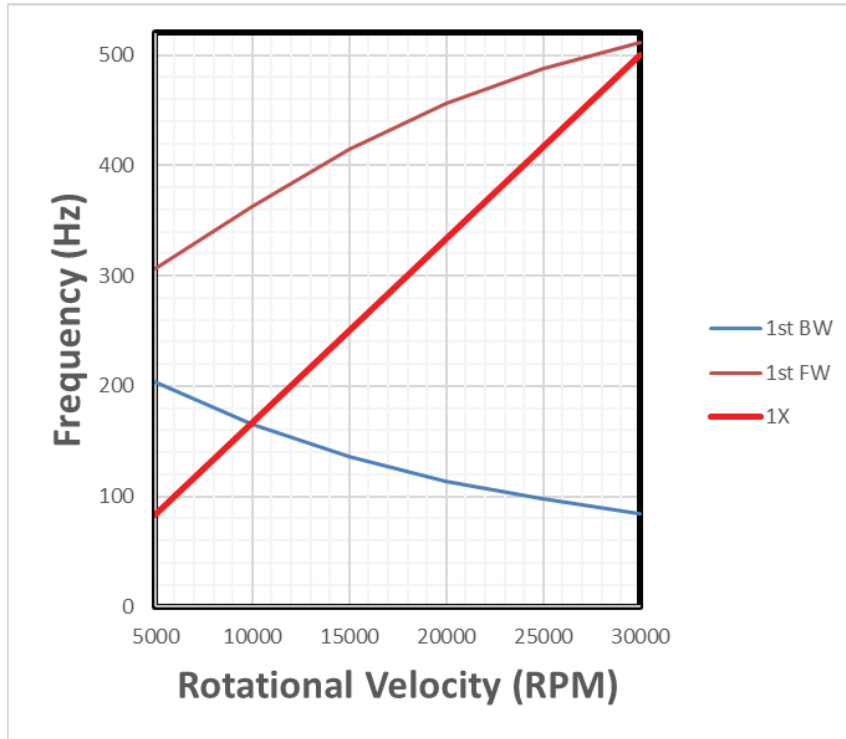


Fig. 3 Campbell Diagram of the Turbopump Rotor with Fixed Bearing Stiffness.

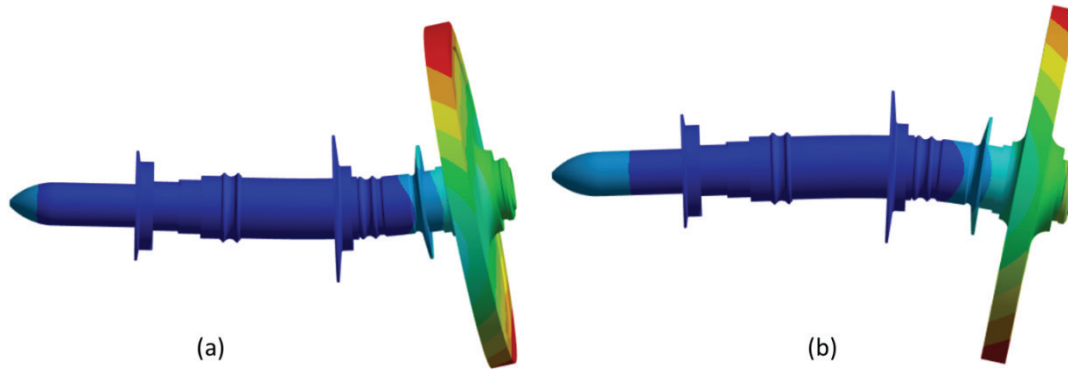


Fig. 4 Rotor Mode Shapes at 5,000 RPM: a) 1st Backward Whirl Bending (203.6 Hz), b) 1st Forward Whirl Bending (306.39 Hz).

Superelement Generation

The whole stator was meshed by using SOLID187 (tet10) elements except for bearing outer races where SOLID185 (hex8) elements were used. The whole model has 865,000 nodes and 481,000 elements. All the interfaces were modeled with Surf-to-Surf Always Bonded Contacts. Inconel 718, with a density of 8240 kg/m³, a Young's modulus of 200 GPa, and a Poisson's ratio of 0.3, was assigned to all components. The model was fixed by using the indicated locations illustrated in Figure 5.

A condensed model was created for the structural parts depicted in Figure 5. Fixity nodes and interface nodes of the bearing outer race were identified as master nodes. CMS algorithm was employed during condensation. Prior to this, modal analysis was performed to determine how many modes were required. It is aimed to capture approximately 95% of the total mass in all translational directions. The modal analysis indicated that the first 50 modes were necessary to achieve this. Sparse solver was used and expansion pass was disabled. With these options, the run time to generate a condensed model was approximately 7 sec with Distributed Solver using 10 processors.

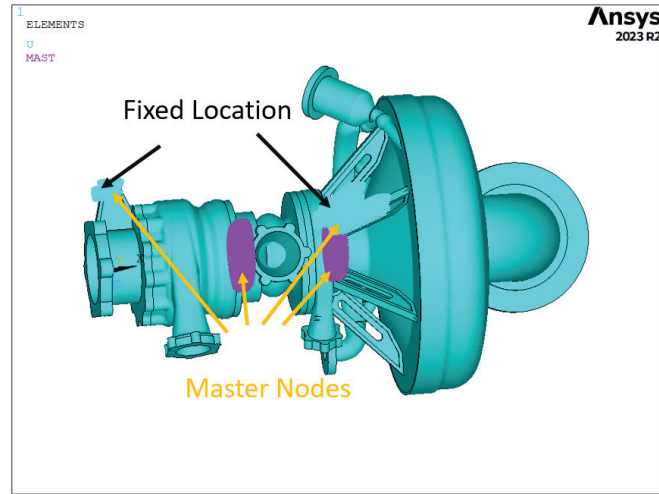


Fig. 5 Turbopump Stator Model for Component Mode Synthesis.

Rotordynamic Model including Stator Flexibility

In this section, the full 3D rotor model was coupled with the superelement representing the stator model. This coupling was facilitated by two bearings with constant stiffness, as shown in Figure 6. To analyze the dynamic behavior of this reduced model, a complex eigenvalue analysis was performed, generating the Campbell diagram for a range of rotor speeds from 5,000 to 30,000 RPM, as shown in Figure 7. Within this diagram, the 1X line represents the rotor's operating speed. Additionally, Figure 8 presents the first backward and first forward mode shapes of the reduced system at a speed of 5,000 RPM, providing valuable insights into the rotor's dynamic performance under these operating conditions.

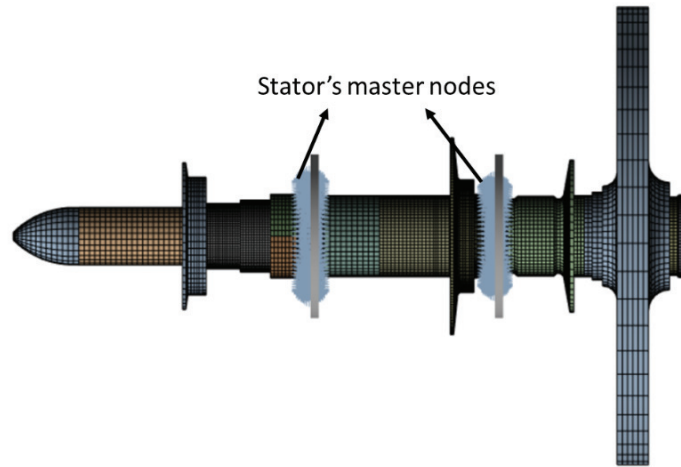


Fig. 6 Coupling between the Rotor and the Superelement (Reduced Model).

As illustrated in Figure 8, the inclusion of stator flexibility leads to a reduction in both the forward and backward whirl speeds. Specifically, the flexibility of the casing decreases the first forward whirl critical speed to 21,000 RPM, representing a reduction of 31% compared to the model that considers only the rotor. This highlights the significant impact that stator flexibility has on the dynamic behavior of the system. Furthermore, the flexibility of the stator introduces a critical speed within a range of approximately $\pm 15\%$ of the operating speed.

To validate the rotordynamic analysis of the reduced system, the complete turbopump model was analyzed by coupling the 3D rotor and stator models, which were prepared for the superelement extraction process, as shown in Figure 9. The rotor and stator were connected using bearing elements with a constant stiffness of 10^8 N/mm. The whole turbopump

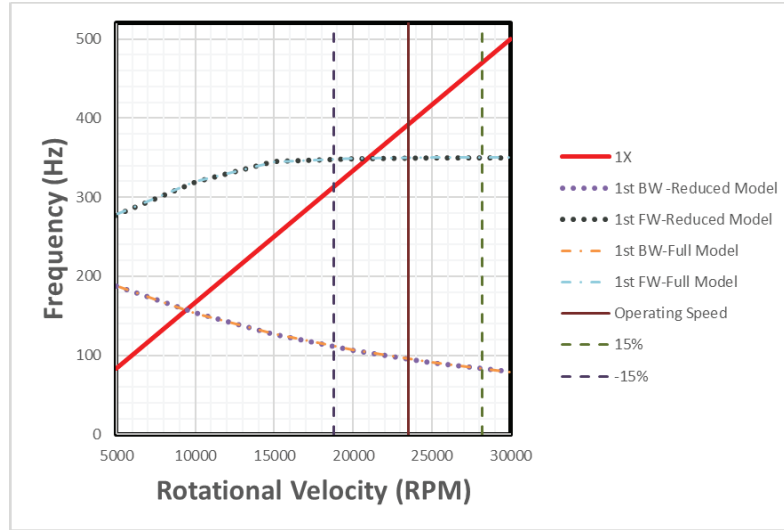


Fig. 7 Campbell Diagram of the Turbopump Rotor with Stator Configuration.

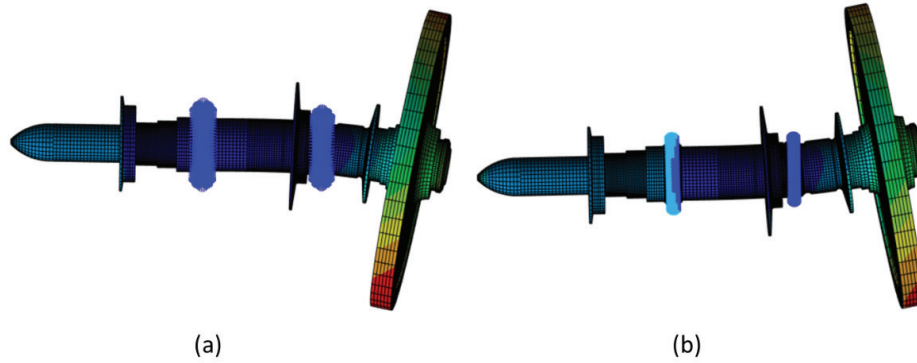


Fig. 8 Reduced Model Rotor Mode Shapes at 5000 RPM: a) First Backward Whirl Bending (187.64 Hz), b) First Forward Whirl Bending (277.85 Hz).

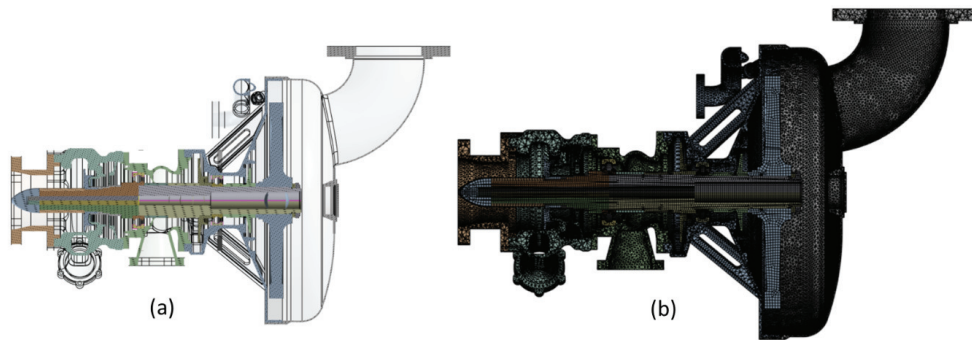


Fig. 9 Whole Turbopump Model a) Geometric Representation b) Finite Element Model Representation.

model consists of 1,001,832 nodes and 572,060 elements, as shown in Figure 9. The stator was predominantly meshed with SOLID187 elements, while the rotor was modeled using SOLID185 elements in ANSYS, ensuring an accurate representation of the system's dynamic behavior. A complex eigenvalue analysis was carried out to develop the Campbell diagram for rotor speeds ranging from 5,000 to 30,000 RPM, as shown in Figure 7. In this diagram, the 1X line indicates the rotor's operating speed, serving as a reference for evaluating resonance conditions. Furthermore, Figure 10 illustrates the first backward and forward mode shapes of the rotor at an operational speed of 5,000 RPM.

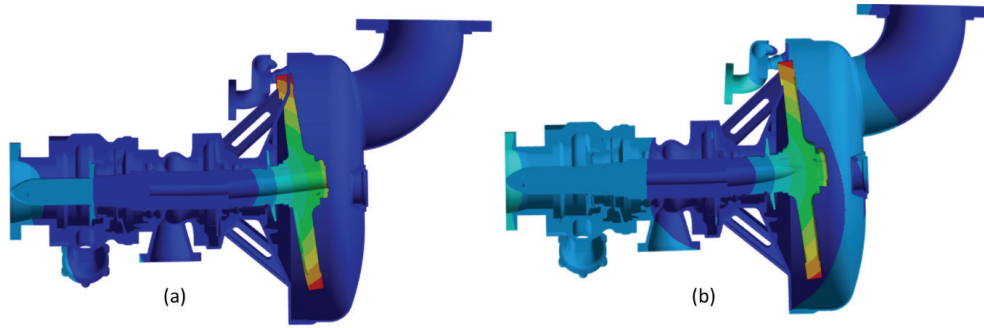


Fig. 10 Mode Shapes of the Whole Turbopump at 5,000 RPM a) First Backward Whirl Bending (187.64 Hz), b) First Forward Whirl Bending (277.85 Hz).

An examination of the Campbell diagram presented in Figure 11 reveals that the critical speeds for the whole turbopump model and the reduced turbopump model are nearly identical. When comparing critical speeds within the 5,000 to 30,000 RPM range, the discrepancy between the two models is less than 0.01%. However, when comparing these results to the rotor-only model, a significant difference of 31% in critical speeds is observed. Furthermore, the Campbell Diagram in Figure 11 indicates that, in the rotor-only model, no critical speed is present within $\pm 15\%$ of the operating speed. In contrast, the inclusion of stator flexibility results in a critical speed appearing within this range. These findings underscore the necessity of incorporating stator flexibility in turbopump rotordynamic analyses to ensure accurate results.

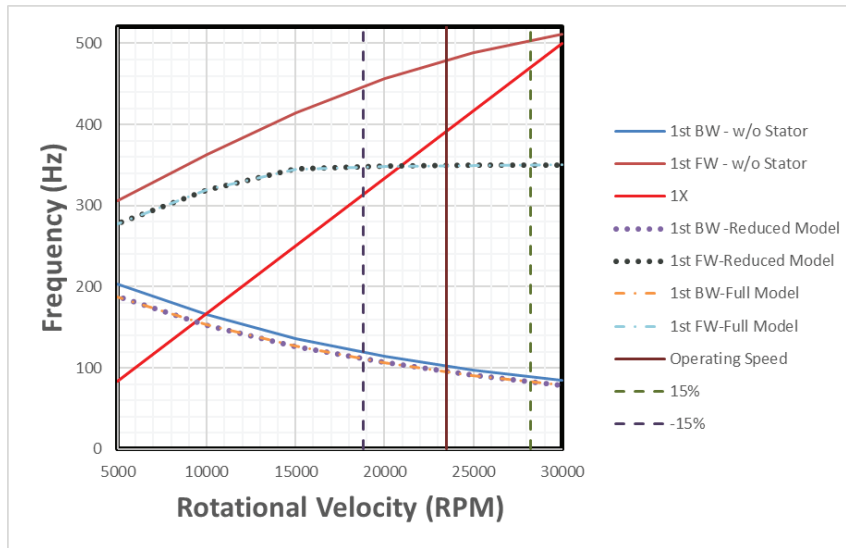


Fig. 11 Campbell Diagram of the Turbopump Rotor.

Wall time refers to the real-time duration required to complete a simulation, measured from start to finish. It encompasses the entire computational process, including overhead tasks such as input data loading, performing calculations, and outputting results. As per Table 1, although the whole turbopump model and reduced model yield almost the same result, the wall time for the reduced model is ten times faster than the whole turbopump model's wall time. Reduced model saves an enormous amount of time while maintaining accuracy. It is important to note that a superelement generation needs to be created only once and can be reused in any subsequent analyses.

Table 1 Wall Time Comparison for Campbell Diagram Generation

Model Name	Wall Time
Rotor without Stator	18 min
Reduced Model	26 min
Whole Turbopump Model	273 min

Conclusion

The use of the superelement approach in this study has proven to be highly effective in reducing the computational effort required for comprehensive rotordynamic analysis of turbopumps. The results of the reduced model closely align with those of the full 3D model, with a minimal discrepancy in critical speeds, thus validating the accuracy of the superelement method. The inclusion of stator flexibility was shown to have a significant impact on the system's dynamic behavior, particularly by lowering the critical speeds of both forward and backward whirl modes. This finding emphasizes the importance of considering stator flexibility in turbopump design to avoid potential resonance issues. The reduction in computation time by an order of magnitude demonstrates the practical advantage of the superelement method, making it an invaluable tool for the design and optimization of rotating machinery in aerospace applications.

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Chapter 3

Metamaterial-Based Vibration Mitigation for Enhanced Reliability of an Automotive Inverter: A Numerical Study

Sara Tincani, Claus Claeys, Elke Deckers, Nimish Pandiya, and Christian Dindorf

Abstract This paper investigates the deployment of locally resonant metamaterials on an electric drivetrain power electronics unit to attenuate structural vibrations on critical sub-components on the Printed Circuit Board (PCB) through simulation-driven investigations. The PCB comprises of a copper foil layer laminated with insulating materials, onto which various electronically functional components (such as transistors, diodes, and resistors) are soldered. Operational vehicle loads lead to critical vibration levels on the PCB plate at resonances, which are composed of both local (only PCB plate) and global (across the whole assembly) modes. Such loads are causes of sensor breakage on the PCB in terms of vibration reliability and lead to functional failures.

Traditionally, metamaterial solutions are mostly designed as periodic structures, applied locally and uniformly to the component requiring vibration reduction. However, such a direct, traditional application of repeated unit cells on the PCB is impractical due to operational and design constraints. This necessitates an alternative approach where the vibration mitigation system is applied to a different sub-component in the assembly in a non-uniform manner to shield the PCBs from critical zones. Hence, the feasibility of such an adaptation of metamaterial design to the spider-frame (a die-casted part that secures the PCB to the assembly) is examined.

The required mass increase and strategic placement of unit cells (UCs) are studied. Hereto different realizable resonators are compared. Furthermore, the results obtained with a realizable locally resonant metamaterial (LRM) concept are compared against idealized tuned vibration absorber (TVA). Vibration reduction is shown to be achieved via addition of metamaterials to the spider by means of a comparative frequency response function (FRF) analysis against the original design concept.

Keywords Locally Resonant Metamaterials, Tuned Vibration Absorbers, Reliability, Printed Circuit Board, E-mobility

Introduction

During their service life, electronic components in electric vehicles are subjected to vibrational loads that require in-depth analysis and testing to ensure their reliability during their operational lifetime [1]. The principal reason for damage occurring is the difference in bending characteristics between the Printed Circuit Board (PCB) and the electronic components soldered to it [2], [3]. Different studies have been conducted both experimentally and through simulation to consider the reliability

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of these components leading to proposed solutions for robust bonding between sensors and the PCB, surface treatment, or stiffening of the supporting structure [4], [5]. These solutions typically involve chemical treatments or time-consuming design modifications implemented for the PCB's sensors connections.

The work presented in this paper focuses on the possibility of exploiting novel, lightweight vibration control solutions to reduce the critical vibrational load in certain frequency ranges, which would be critical for the described components. In this regard, locally resonant metamaterials (LRMs) have been extensively studied in the past decade and have attracted (majorly) academic treatment and attention due to their ability to combine lightweight design and excellent NVH (noise, vibration, and harshness) behavior in targeted frequency ranges. These frequency ranges of interest, also called stop bands, are zones where no free wave propagation is allowed via targeted modification of the object's structural dynamics [6].

This is achieved mainly through two different approaches:

- Bragg scattering interference [7]: at frequencies where the Bragg condition is satisfied, stop bands may be created by ensuring low transmission of waves through the presence of zones with destructive interference between reflected and transmitted waves. This is the mechanism on which the creation of stop bands in photonic and phononic crystals is based.
- Resonance based approaches: due to Fano-type interference between wave propagating in the host structure and re-radiated waves by resonant additions to the host [8] [9], stop bands with vibrational attenuation are created. This type of material structures are typically referred to as LRMs [6].

While photonic crystals are significantly dependent on the length scale of periodicity, this is not the case with LRMs, which, due to their dependency on resonant phenomena, can enable stop bands by additions on a sub-wavelength scale. [9]. For these reasons, LRMs prove to be a promising solution for vibration reduction, especially in low-frequency, practical applications.

Various real-world scenarios of LRM implementations in the automotive sector for low-frequency airborne and structure-borne noise, and vibration reduction are present in literature [11] and are reviewed for this work. The vehicle cabin interior vibration attenuation by use of mass-spring systems is investigated through simulations by Wu et al. [12]. Effective reduction of structure-borne noise in the vehicle cabin and low frequency noise mitigation in tires are reported in the work of Sangiuliano et al. [13], [14] through numerical and experimental studies. Jung et al. [15] and Riess et al. [16] report reduction of air-borne noise by exploiting the implementation of LRM solutions on car body panels and doors respectively.

LRMs have been proved to be effective lightweight performance solution capable of improving NVH behavior over targeted frequency ranges, ensuring, if necessary, a compact design. Moreover, thanks to the possibility of using various production technologies such as additive manufacturing or injection molding [17], [18] and different materials, they can represent a valuable solution to vibrational problems for multiple industrial applications. Extending the concept described in the literature, a motivated investigation of LRMs on electronic components for reduction of vibrational load is warranted especially to ensure improved component reliability. In particular, the present simulation-based study investigates the potential of an LRM solution to reduce the vibrational load experienced on the PCB of an electric drive power electronics unit, especially when it is subject to a global mode of the structure. The most critical modes of the PCB are analyzed and various LRM solutions that can be implemented on the component, are proposed and compared via a numerical study.

The paper is outlined as follows: in Section 2 the problem is presented, and the component is described, both in terms of the typical NVH behavior of the structure and the definition of the frequency range of interest. The constraints to be respected in the design of the solution are highlighted. In Section 3 a single tuned vibration absorber (TVA), modeled as a mass-spring system, and a LRM concept, realized using idealized mass-spring systems, are compared. The results indicate the LRM solution as the best alternative. Section 4 presents two different LRM designs. Their dispersion curves are presented, and their performance is compared to an idealized TVA-case in order to determine the best performing design type. Thereafter, the optimal solution, that guarantees an improvement to the vibrational response of the component, is selected. In Section 5 the LRM solution chosen in the previous section is applied to the component, and the simulation on the entire structure discussed. Finally, in Section 6, the findings of the simulations are summarized along with an outlook for future work.

Problem Definition

The PCB as a part of the Power Electronic Unit (PEU) assembly (also referred to as inverter), exhibits both global modes (modes involving the entire structure) and local modes (where only the PCB and the sensors placed on it are affected) in the frequency range of interest. For the present treatment, a global mode of the structure is considered. For confidentiality reasons only the PCB mounted on the die-casted spider plate is shown (regarding these images, they represent a generic geometry, not specific to the product). For simulations, however, the complete PEU assembly is evaluated.

Fig. 1 shows the assembly composed of the spider frame and PCB, and the frame only. In particular, the latter is the component of interest for the implementation of the LRM solution since a direct application on the PCB is not practical. This is due to the complexity of the component itself and the infeasibility of making direct modifications to it. This work therefore considers the addition of resonators on the spider, tuned to act on a global out-of-plane mode of the (PCB + spider) sub-assembly where high displacements are seen to occur. Considering the complexity of the frame, which has various ribs and holes and is not characterized by a continuous surface, it is investigated how an efficient LRM solution would reduce the system response. To determine the feasibility of the solution, a (theoretical) LRM concept, modeled as a grid of spring-mass systems (TVAs), is compared with the implementation of a single TVA, and the different effects are compared (in the work presented, when referring to TVAs, it refers to idealized mass-spring systems).

Furthermore, two different design goals are set for the implementation of the LRM solution:

1. The total amount of mass added due to the resonators should not exceed around 15% of the assembly mass. The resonators should be producible by an efficient manufacturing process.
2. Considering the presence of the PCB screwed on the frame, the dimensions of the resonator are constrained by the space available between the two components.

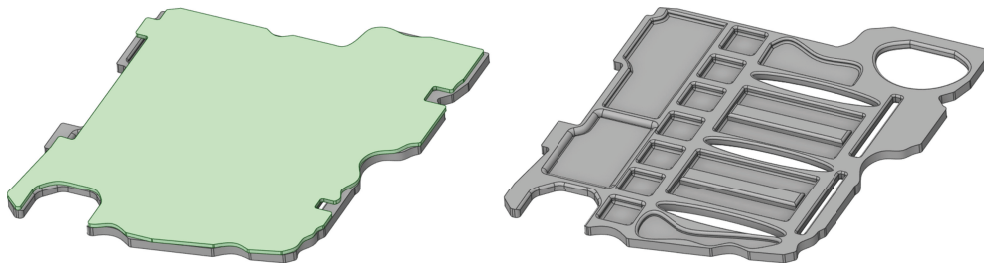


Fig. 1 Left: Sub-assembly: Frame and PCB. Right: Frame.

LRM Solution And Single Tuned Vibration Absorber: Comparison

In the initial phase of the study, both the LRM and the single-TVA were modeled using mass-spring systems. Ansys is used for all simulations described hereafter. The simplicity of the lumped-mass resonators permits a quick evaluation of the effectiveness of the proposed solutions and provide comparisons of their effectiveness on the structure. The lumped masses are modeled using MASS21 elements, while the springs are modeled using COMBIN14 elements. The ends of the spring elements are connected to the point mass and to the component in the desired and defined positions through a remote point. The total mass addition corresponds to 15% of the mass of the assembly. In all three considered configurations, the stiffness of the springs is adjusted so that the resonance frequency of each TVA is equal to the frequency at which the out-of-plane mode of the structure occurs: to obtain a stopband on the frequency range of interest it is necessary that the resonators introduce a localized resonant behavior in the host structure [3]. The exact frequency value is excluded in the present discussion due to confidentiality reasons and is replaced by f_{res} .

An important consideration regards the scale at which the resonant elements operate in relation to the wavelength of vibrations they are intended to mitigate: for a structure, to be an effective LRM solution, and hence exhibit stop band behavior, the size of the spacing between the resonant elements should be smaller than the wavelength of the waves they are going to affect [6].

The three cases are distinguished as follows:

1. In the first case, all the added mass is concentrated in a single TVA, which is placed in the central zone of the frame (also the zone subject to the maximum deformation at f_{res}). The single TVA does not directly relate to the theoretical LRM concept, as it focuses on localized mass addition rather than a periodic arrangement of elements.
2. In the second case, the added mass is equally divided among 25 TVAs. This case represents the maximum number of TVAs that can be realistically added, given constraints imposed by the geometry of the component, and have been also positioned where resonators could be placed in a real case scenario. The wavelength of interest is influenced by factors such as the material properties of the frame, the boundary conditions, and the operational environment, making a precise, universal definition of subwavelength challenging for this kind of application. However, given the physical

dimensions of the frame and the spacing between the TVAs, this concept is considered within the subwavelength regime for the range of frequencies considered.

3. In the third case, the added mass is equally divided among 50 TVAs. Placement of 50 TVAs is physically unrealistic as the added TVAs would cover the entire spider plate. Even so, this case is considered to evaluate the behavior and effect of a classic LRM concept. On average the spacing between the TVAs in this case can be similar as in case 2.

The three cases considered are shown in Fig. 2.

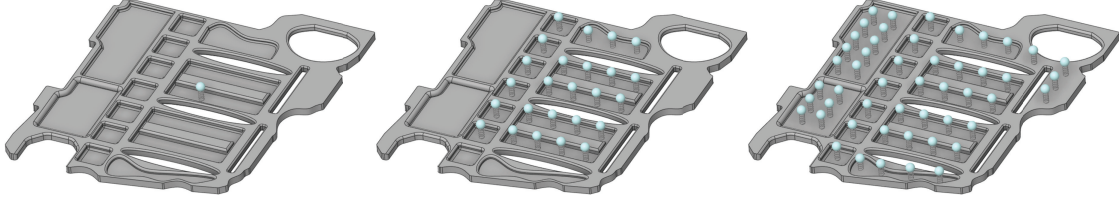


Fig. 2 Single TVA configuration. Center: 25 TVAs configuration. Right: 50 TVAs configuration.

For all three described configurations, a constant force with a magnitude of 1 N is used as the excitation of the complete PEU assembly. The system is fixed at its housing attachment points, and the profile of 1N from the lower edge of the frequency range, referred to as f_1 , to the upper edge, referred to as f_2 , is applied at those locations (as reported above, the entire inverter is not represented: on the housing, there are attachment points with which the inverter is fixed to other components, and in the simulations conducted, such points have been fixed and the force profile has been here applied). For the frame, a tetrahedral mesh made of TET10 elements is used, while for the PCB, a hexahedral mesh made of HEX20 elements is employed. Structural damping of 1% is applied to the entire inverter structure, while no structural damping is included in the TVAs. The forced response is computed via mode-based steady state simulation in Ansys.

It is worthwhile to note that the stop band generated by the resonators can be increased by the addition of damping, however the maximum attenuation achieved in this case is lower than the maximum attenuation obtained when damping is absent in line with findings in literature [6], [19]. In section 6, instead, where the real resonators have been applied to the structure, 5% structural damping has been included in the resonator's material properties.

Fig. 3 shows the averaged spectral displacement responses recorded on the upper surface of the PCB in the out-of-plane mode direction. The base configuration, i.e., without the presence of TVAs is also shown for reference.

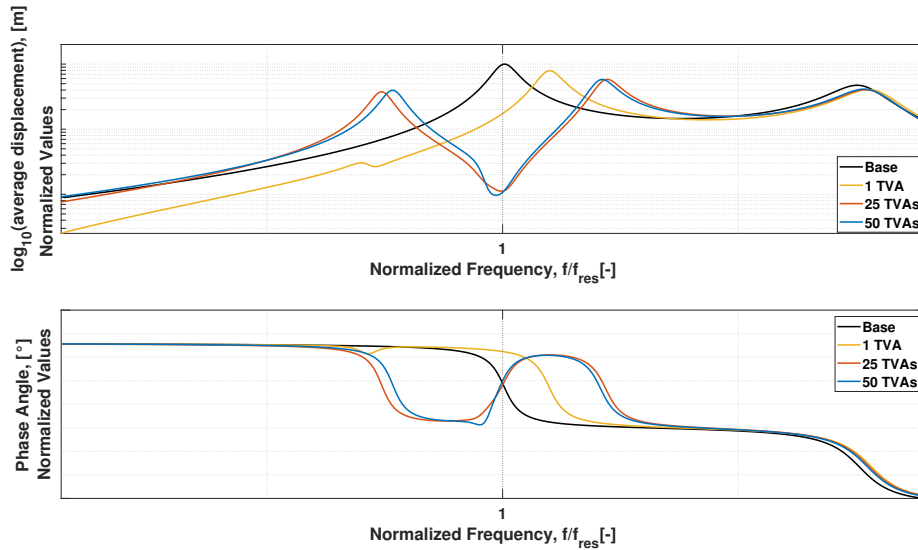


Fig. 3 The average of the directional displacement responses (amplitude and phase in normalized values) recorded on the upper face of the PCB. Frame without TVAs (Base), with 1, 25 and 50 TVAs configuration cases.

The displacement responses represented in amplitude and phase across a range of frequencies reveal different behavior among the configurations, even though the total added mass is consistent. This is expected since locations of each resonator acting on the targeted mode are different.

The single TVA configuration shows a shift in the peak frequency to a slightly higher value with a slight reduction in amplitude, indicative of a basic yet effective vibration mitigation strategy. In contrast, the configurations with 25 and 50 TVAs exhibit a more complex response, characterized by the emergence of dual peaks, one before and the other after the f_{res} frequency. This dual-peak phenomenon, also confirmed by the phase change, suggests an enhanced reduction effect, due to the distributed nature of the TVAs. Additionally, the amplitude of these peaks is reduced as compared to both the base case and the single TVA configuration, highlighting the superior efficiency of the LRM configurations. Interestingly, the similarity in response between the 25 and 50 TVAs configurations suggests a saturation effect, where additional TVAs beyond a certain point do not contribute significantly to further reduction enhancement. Optimal placement and quantity of TVAs can hence be marked as crucial factors in maximizing vibration mitigation. The boundary conditions of the frame, rigidly attached to the external part of the PEU assembly also likely contribute to this effect through emphasis on the predominant role of the central area in the system's vibrational response: due to the fixing of the frame on the PEU housing, it is indeed observed that the central area is more susceptible to deformation, while in the lateral parts, close to the attachment points, such deformation is almost nonexistent. In the configuration with 50 TVAs, therefore, those placed near the attachment points to the housing do not produce an effect comparable to those located in the central part, and it is for this reason that the two configurations exhibit similar behaviors. Furthermore, as previously discussed, the presence of structural damping in the TVAs is expected to reduce the prominence of the first and second peaks in the LRM configuration. This will result in less pronounced attenuation, leading to a broader stop band.

This preliminary study highlights the attenuation introduced by the LRM solution compared to the single TVA case. In the following sections, the Unit Cell modeling approach is explained, and the behavior of practically realizable resonators is studied, when applied to the component.

LRM Solution: Unit Cell Modeling And Comparison

Unit Cell (UC) modeling allows for the study of infinite periodic structures by considering a single part, called the Unit Cell, which represents the smallest non-repetitive part [20]. Starting from the single UC, dispersion curves are calculated, providing information about the presence of stop bands, frequency zones where no free wave propagation is possible [6]. Although this type of modeling is applied to the study of infinite periodic structures, it has been shown to be applicable to finite structures as well [17] and has been extensively used to simulate the behavior of a LRM concept on a real component [11], [13], [15], [16], [21].

In the present application case, considering the presence of a finite system and a particularly complex geometry that causes the presence of ribs and holes, a UC modelling approach is not fully representative. Nevertheless, it is used as a rapid and effective method to compare different types of resonators designs, allowing for the selection of the most suitable resonator type and avoiding computationally expensive simulations.

When applying resonant additions to the component, the conditions for stop band behavior are considered. The resonant cells are added to the component on a scale smaller than the structural wavelength to be influenced [6], the net sum of the forces on the hosting structure contributed by a resonator should be non-zero [10], and they should introduce a localized resonant behavior in the host structure [3], [22]. The UC modeling is based on the application of the Bloch-Floquet theorem, and the application of periodic boundary conditions allows for the simulation of the single UC using the Finite Element method (FEM) [23], [24]: for the inverse approach the wavenumbers are imposed and the problem is solved towards the frequencies, while for the direct approach real frequencies are imposed and the eigenvalue problem is solved to the wave propagation constants [20].

The Bloch's theorem considers the response of a single UC to define the response of a periodic system, which is described by the periodicity of the fundamental UC. This is made possible through the definition of an exponential term that defines both the amplitude and the phase change when a waves passes from one UC to the next [21], [23]. This exponential term utilizes the wave vector $\mathbf{k}$ to define the amplitude and the phase change, particularly considering two propagation directions $\mathbf{d}_x$ and $\mathbf{d}_y$. The propagation vector $\boldsymbol{\mu} = (\mu_x, \mu_y)$ defines the complex phase shift for a wave propagating along one of the two directions. Eq. (1) gives the relation between the wave vector, the propagation vector, and the direction of propagation:

$$\mu_i = \mathbf{k} \cdot \mathbf{d}_i \quad (1)$$

Therefore, considering that the forces and displacements on the single UC's boundaries can be scaled in the $\mathbf{d}_x$ and $\mathbf{d}_y$ directions using e^{μ_x} and e^{μ_y} , a matrix $\mathbf{R}$ (function of $\boldsymbol{\mu}$) is used to relate the generalized displacement $\mathbf{q}$ to a reduced set of displacement $\mathbf{q}^{\text{red}}$, eq. (2).

$$\mathbf{R}\mathbf{q}^{\text{red}} = \mathbf{q} \quad (2)$$

The same matrix $\mathbf{R}$ can also be used to rewrite the equation of motion, in a matrix form for a steady-state harmonic responses, in terms of the reduced set, eq. (3):

$$\mathbf{R}' (\mathbf{K} + j\omega \mathbf{C} - \omega^2 \mathbf{M}) \mathbf{R} \mathbf{q}^{\text{red}} = \mathbf{R}' \mathbf{f} \quad (3)$$

where $\mathbf{R}'$ is the conjugate transpose of $\mathbf{R}$. Assuming that no external forces are applied to the internal nodes of the UC and that $\mathbf{K}^{\text{red}} = \mathbf{R}' \mathbf{K} \mathbf{R}$, $\mathbf{C}^{\text{red}} = \mathbf{R}' \mathbf{C} \mathbf{R}$ and $\mathbf{M}^{\text{red}} = \mathbf{R}' \mathbf{M} \mathbf{R}$, the equation can be rewritten as in eq. (4) [26].

$$(\mathbf{K}^{\text{red}} + j\omega \mathbf{C}^{\text{red}} - \omega^2 \mathbf{M}^{\text{red}}) \mathbf{q}^{\text{red}} = \mathbf{0} \quad (4)$$

By solving the eigenvalue problem in eq. (4), the dispersion curves can be obtained. The three unknowns of the problem are: ω , μ_x , μ_y . As stated before, the presence of a direct $\mu(\omega)$ and inverse $\omega(\mu)$ approach derives from the different resolution that this equation has, as both ω and μ_x , μ_y can be complex.

- Inverse approach, $\omega(\mu)$: the wave vector constants (μ_x , μ_y) are imposed, and the problem is solved for ω . Free propagation wave is considered, and this type of approach is typically used for undamped UCs (with a damping matrix $\mathbf{C}$ equal to the zero matrix). The resulting frequencies can be real, imaginary or complex representing freely propagating, evanescent or decaying waves [20].
- Direct approach, $\mu(\omega)$: in this case, real ω values are imposed, and the problem is solved with respect to (μ_x , μ_y). A time-harmonic wave propagation is considered, and the approach is typically used for damped UCs.

For further information regarding this modeling technique and problem resolution using the Inverse and Direct approaches, the interested reader is referred to [6], [23], [24], [26], [27].

In the current study, the methodology of UC modeling through an inverse approach has been applied, including simulations of three distinct two-dimensional (2D) UCs without incorporating damping effects into the calculations. The UCs under examination are illustrated in Fig. 4, which comprises three variants. Fig. 4 presents on the left a UC conceptualized as a single TVA depicted as a mass-spring system and referred to as the TVA UC. On the central and right part, Fig. 4 displays UCs including practical resonator designs, labeled as the Cantilever Resonator UC and the Ring Resonator UC, respectively. The TVA UC configuration represents an idealized scenario, where the mass-spring system serves as a preliminary model, as detailed in section 3. The proper resonators designs are conceptualized for fabrication via Additive Manufacturing techniques: this technology allows to produce resonators bodies and when mass addition is needed, it can be positioned on top of the resonators. For the study presented, the Cantilever Resonator body has been modeled with polylactic acid-based filament (PLA) and the Ring Resonator with thermoplastic polyurethane (TPU). For the supplementary mass, steel has been considered. The following material constants have been employed, taken from commercial material datasheets, and reported in Table 1. For the subsequent creation of the samples, it is necessary to experimentally characterize the values for the indicated materials and to consider the substantial variations that can occur depending on the type of filament chosen or its orientation.

Table 1 Resonators' material properties.

Material	Young Modulus, E [Pa]	Poisson Ratio, ν [—]	Density, ρ [kg/m^3]
PLA	2865 MPa	0,35	1240 kg/m^3
TPU	60 MPa	0,4	1080 kg/m^3
Steel	193 GPa	0,3	8000 kg/m^3

In the three studied cases, the selected unit cell measures $0,075 \times 0,105 mm$ (lengths normalized with respect to the frame's longest dimension), allowing for the placement of 25 TVAs or resonators as established in preliminary simulations. This configuration also marks the limit for the number of resonators that can be placed on the frame due to the geometric constraints of the structure. The three types of resonators are designed to exert a force perpendicular to the structure: in fact, the out-of-plane waves aim to be affected by these types of resonators.

For the base of the three UCs, SHELL181 elements have been employed. The Cantilever Resonator and Ring Resonator bodies have been modeled using solid HEX20 elements. For the TVA UC, a MASS21 element has been used to simulate the mass component and a COMBIN14 element has been selected for the spring component. Detailed specifications of the models utilized are presented in Table 2.

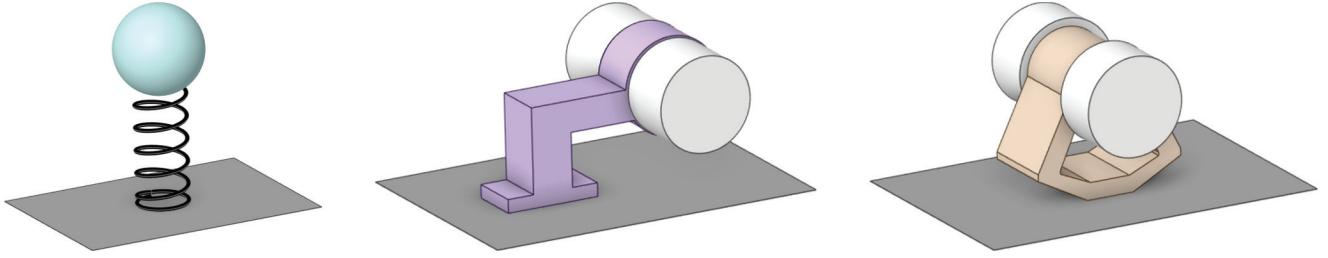


Fig. 4 Unit Cells. Left: TVA UC. Center: Cantilever Resonator UC. Right: Ring Resonator UC.

Table 2 Model specifications. UCs dimensions, mesh element types and mesh element number employed.

Type of UC	UC Length (mm)	Mesh elements (type)	Mesh elements (number)
TVA UC	0,075x0,105	Base: SHELL181 Mass: MASS21; Spring: COMBIN14	Base: 1800 Mass: 1; Spring: 1
Cantilever Resonator UC	0,075x0,105	Base: SHELL181 Body: HEX20	Base: 1607 Body: 1572
Ring Resonator UC	0,075x0,105	Base: SHELL181 Body: HEX20	Base: 1628 Body: 1649

In the case of the TVA UC, the mass-spring configuration contributes an additional 15% mass to the base UC and exhibits resonance at the frequency f_{res} . The analysis of dispersion curves, along the Irreducible Brillouin Contour (IBR) delineated as O, A, B, C, O and representing a transition through the points $(0,0)$, $(0,\pi)$, (π,π) , $(\pi,0)$, $(0,0)$, is shown in Fig. 5. The IBR is defined as the smallest region within the wave domain that contains all essential information: given the periodic nature of the structure under examination, its properties related to dispersion surfaces are also periodic and this periodicity justifies the reduction to the IBR for assessing the presence of stop bands [23]. Dispersion curves for the TVA UC are derived and depicted in Fig. 5. The area marked in light blue represents the stopband for the target frequency range and exhibits a bandwidth that will be addressed as $f_{\text{Band,TVA}}$. This region is defined by two boundaries corresponding to the out-of-plane flexural waves (indicated in orange). The TVA hence creates a stopband centered around f_{res} , which denotes the frequency corresponding to the out-of-plane modal response of the component.

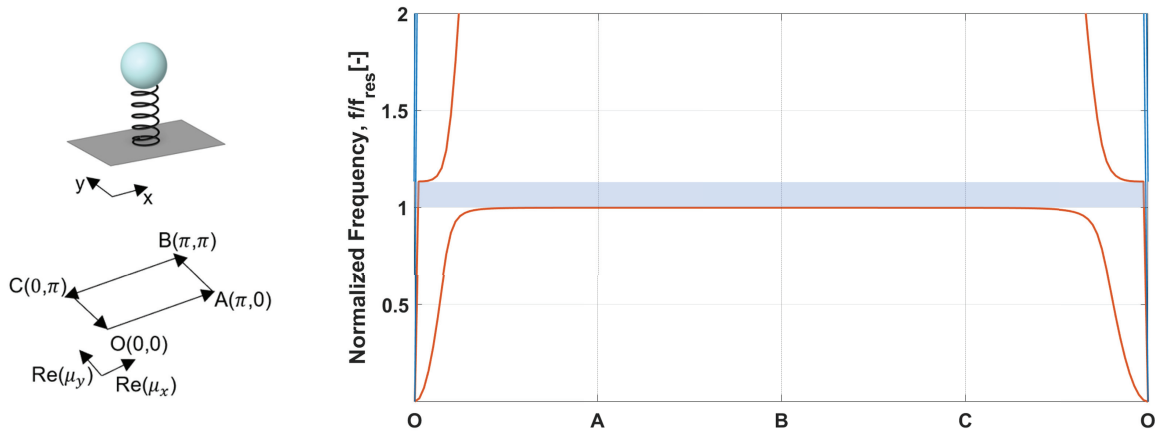


Fig. 5 Top left: Representation of the TVA UC. Bottom left: The IBR used in the computation. Right: Computed dispersion curves.

The fundamental UCs and their dispersion curves for the Cantilever Resonator UC, (Fig. 6), and the Ring Resonator UC, (Fig. 7), have been also analyzed. In these analyses, a mass equivalent to 15% of the UC base is added, and the resulting stop-band widths have been compared to those obtained for the TVA UC. The stop-band width for the Cantilever

Resonator UC is called $f_{\text{Band,Cant}}$ and its value is the 77% of $f_{\text{Band,TVA}}$, whereas for the Ring Resonator UC, it is named $f_{\text{Band, Ring}}$ and its value is the 94% of the $f_{\text{Band,TVA}}$. Evidently, latter is notably closer to the stop-band width achieved with an idealized TVA. Although the added mass is consistent across all three cases, the stop-band widths vary, particularly for the Cantilever Resonator. This difference is attributed to the significantly higher Effective Modal Mass (EMM) in the out-of-plane direction for the Ring Resonator compared to the Cantilever Resonator. This higher EMM results in the Ring Resonator approximating the idealized TVA more closely, as is also demonstrated in [28]. Specifically, the EMM in the out-of-plane direction for the Cantilever Resonator is 81% of that of the idealized TVA, while the relative EMM for the Ring Resonator is 97%. Consequently, the stop-band width for the Ring Resonator more closely aligns with that of the idealized TVA. In an idealized TVA, an increase in mass (analogous to the modal effective mass in the out-of-plane direction for real resonators) correlates with an increase in stop-band width, consistent with observations by Claeys et al. [6]. Therefore, the Ring Resonator is selected for further implementation and results on the component under test are discussed in the subsequent section of this study.

Also illustrated in Fig. 6 and Fig. 7 are the fixed base mode shapes of the two resonators. For the Ring Resonator, the mode shape depicted in Fig. 6 corresponds to the first resonance frequency of the system. An examination of the dispersion curves in Fig. 6, shows that the stop band, highlighted in light blue, originates from the first visible waveform on the diagram, which marks the lower limit of the stop band. This stop band is defined by two lines corresponding to the

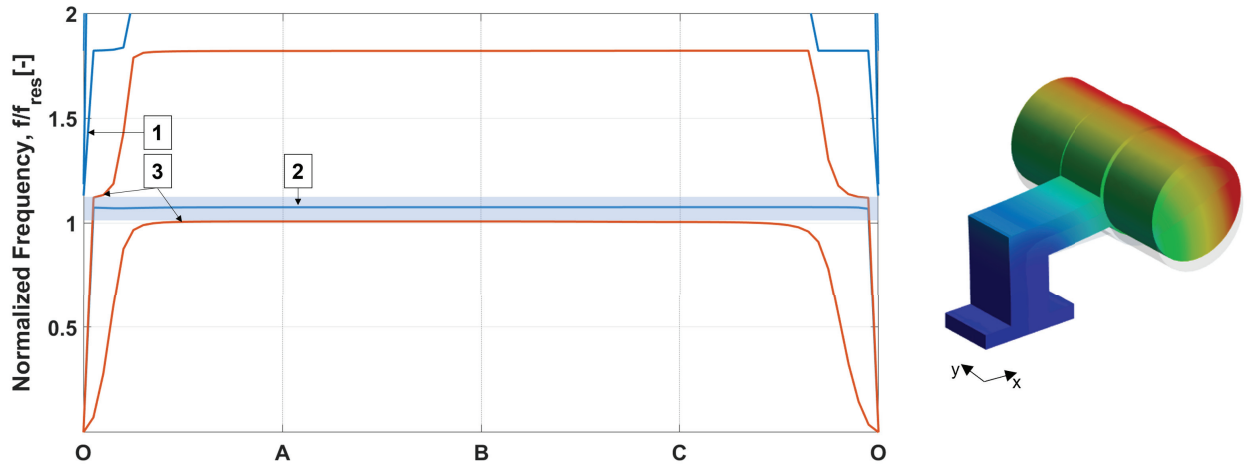


Fig. 6 Left: Cantilever Resonator UC Dispersion Curves. Right: Cantilever Resonator mode shape.

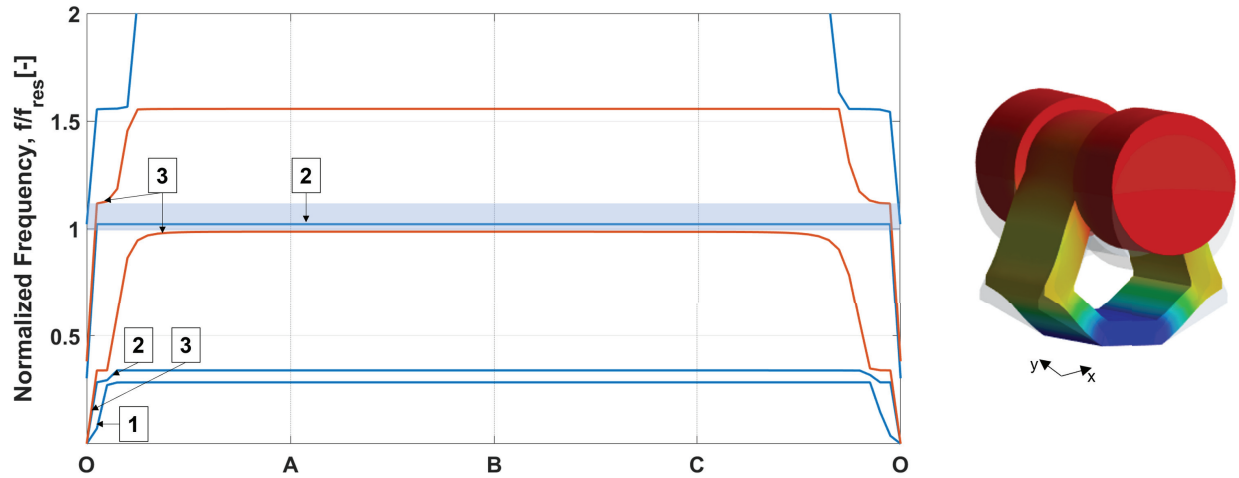


Fig. 7 Left: Ring Resonator UC Dispersion Curves. Right: Ring Resonator mode shape.

out-of-plane flexural waves, labeled in orange as “3”. The diagram also includes other types of waveforms (in blue), particularly longitudinal (labeled as “1”) and shear (“2”). However, these waveforms develop and impact the system in-plane rather than out-of-plane. To influence the global flexural out-of-plane mode of the component, it is necessary to implement the LRM concept such that the resonators generate a stop band to affect the same. Therefore, out-of-plane flexural waves created by the resonators are especially considered.

With regard to the Ring Resonator in Fig. 7, the obtained dispersion curves show a different shape, featuring a greater number of waveforms within the same frequency range. In this case, examining Fig. 7, “1” corresponds to longitudinal waves, “2” to shear waves, and “3” to bending waves (also in this case the out-of-plane flexural waves are indicated in orange and the other waves types in blue). Here too, the stop band occurs at the resonance frequency (f_{res}) for out-of-plane flexural waves, denoted by “3”. At lower frequencies (below $0.5f_{res}$), other curves and bandgaps are visible, which are not observed in the case of the Cantilever Resonator. For the Ring Resonator, the mode shape shown in Fig. 7 corresponds to its third resonance frequency. The first mode shape involves oscillation along the y-axis, and the second along the x-axis (axes are indicated in Fig. 7). The considered mode, tuned to f_{res} , involves a vertical mode along the z-axis, which results in a higher relative EMM compared to the Cantilever Resonator case. Thus, the various curves present below $0.5f_{res}$ are in-plane waveforms generated by the first two resonance frequencies of the Ring Resonator and in this case, no flexural wave gaps are obtained. The bandgap of interest for the out-of-plane flexural wave is instead obtained at f_{res} . In this scenario as well, the out-of-plane flexural waves depicted in the diagram are those capable of generating the stop band for the application of interest.

LRM Implementation On The Real Component

In the final section of this study, the implementation of the selected Ring Resonator is detailed. Specifically, 25 resonators, equivalent to the maximum number of Unit Cells (UCs) that can be positioned on the system, have been placed as illustrated in Fig. 8. This study does not present an optimization of the described system, as the focus is to evaluate the potential for vibration reduction using a LRM concept.

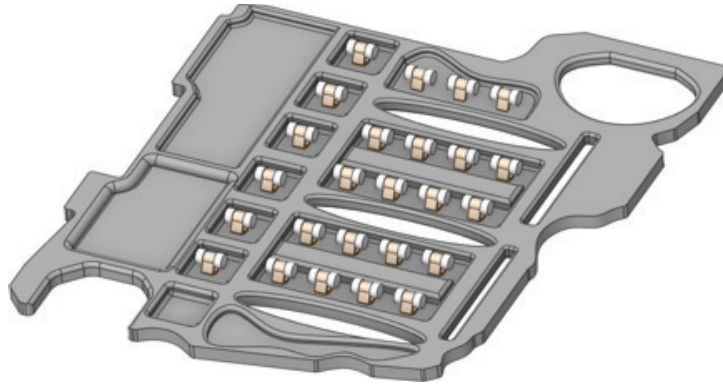


Fig. 8 Frame with LRM Ring Resonators implementation.

Similar to the TVAs discussed in Section 3, the entire system of the PEU has been simulated, and the Ring Resonators have been added to the structure, resulting in a 15% increase in the frame’s mass. The system has been secured at the housing attachment points, and a constant acceleration profile of 1m/s^2 has been applied at these locations as Base Excitation (to facilitate future experimental replication on a shaker). The results are presented as the average of the displacements on the upper face of the PCB. The Ring Resonators, selected as the optimal solution through UC modeling, are simulated (i) without including structural damping among the material properties of the resonators, and (ii) including a structural damping of 5%. Fig. 9 displays the results for the simulations conducted. As with the case presented in Section 3, the black curve represents the base system, the orange curve depicts the system’s response when the 25 Ring Resonators, without the inclusion of 5% structural damping, and the blue curve the response of the LRM solution with the 5% structural damping inclusion.

The outcomes are seen to be comparable and in agreement with the results from the simulation of the 25 TVAs. The amplitude of the peak at the resonant frequency (f_{res}) is reduced, and two distinct peaks are present before and after it, representing in general the in-phase and out of phase modes of the resonators. The presence of this dual-peak phenomenon is confirmed by the phase as well. For the damped resonator case, the reduction achieved is even larger as the two peaks

are present but significantly reduced in amplitude compared to the case without structural damping. As for the reduction achieved at f_{res} , the peak attenuation is higher than that of the case without damping. This is in accordance with references [6], [19].

Considering the results of the system's frequency response when TVAs are applied (ideal case, shown in Fig. 3) and the LRM solution (Fig. 9), it is possible to compare the width of the stopband, considered in this case as the frequency range for which the orange curve (in Fig. 3 representing the solution with 25 TVAs and in Fig. 8 the LRM solution without damping) is positioned below the black one (baseline case for both figures). The stopband obtained when the LRM concept is applied to the actual system turns out to be 92% of the solution that involves the 25 TVAs applied to the same system. It should be noted that, in the case of the simulation related only to the UCs and the computation of the dispersion curves, this percentage was around 94%. Therefore, the simulations conducted have demonstrated the feasibility of creating an LRM concept using designs that are practically manufacturable and whose behavior closely approximates the ideal case simulated in Section 3.

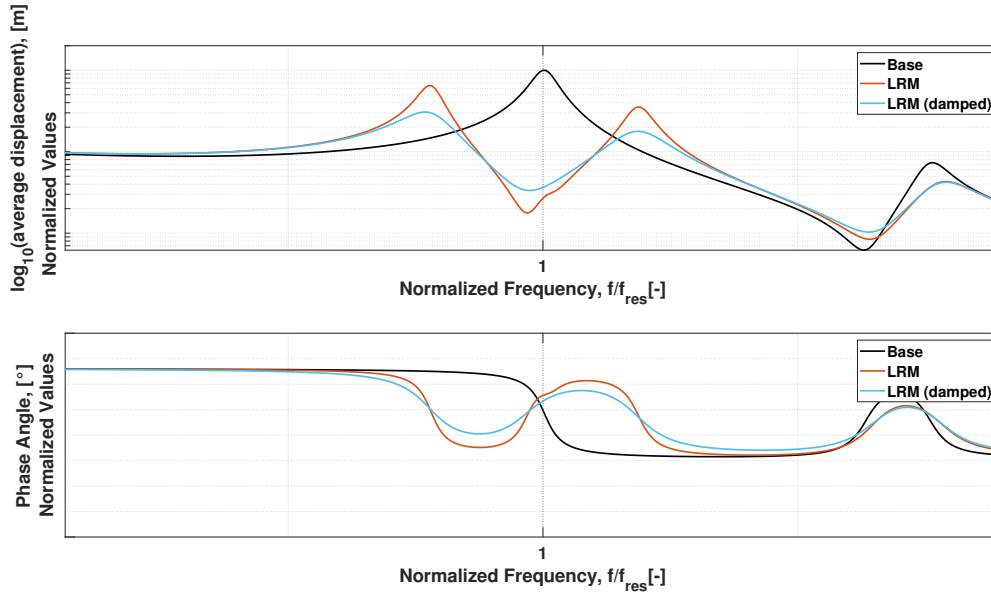


Fig. 9 The average of the directional displacement responses (amplitude and phase in normalized values) recorded on the upper face of the PCB. Frame without LRM concept (Base), LRM concept (LRM) and LRM concept with structural damping (LRM (damped)).

Conclusions

In the presented study, a solution involving LRM applicable to an electric drive PEU is investigated. The problem statement is defined in Section 2, such that an out-of-plane PCB mode, with a global assembly participation is targeted. Key challenges are the physical constraints due to the complex geometry of the component and the requirement to investigate a realizable solution.

The LRM solution is compared to the effectiveness of a single TVA exploring the optimal solution applicable to the component keeping additional mass consistent. This comparison establishes the LRM concept as the superior solution and although the single TVA has an impact on the system, the reduction in component response is less than that achieved with the LRM concept. A comparable reduction is achievable with a single TVA however, this approach is impractical due to the excessive increase in mass on the system at a single location. The proposed optimal resonator is achievable through additive manufacturing technology and capable of being positioned on the frame, while respecting the geometric constraints of the structure.

UC modeling is presented to evaluate initial designs, and different UCs with two types of realizable resonators are simulated and compared, also in relation to an idealized TVA. The Ring Resonator is selected as the optimal solution and its application to the component in a distributed sense is discussed. The results demonstrate its reproducibility in a real case using actual materials and manufacturing technologies.

The numerical study presented indicates that LRMs have potential for application in electronic devices to enhance their reliability and lays the groundwork for a possible experimental study of the considered system. Next steps include experimental characterization of the materials usable for stable production processes of the LRM solutions and experimental validation of the proposed solutions via shaker and vehicle testing for the electric drive PEU.

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Chapter 4

Test-vs-Simulation Correlation of Noise Emission in a Bent-Axis Pump

Bhaskar Banerjee, Michael Kwarta, Zuher Abdel Mallak, Marcus Wiklund Lång, Per-Ola Vallebrant, Adarsh Chaurasia, and Omer Kayani

Abstract This work presents a detailed methodology for simulating the noise emissions of a bent-axis pump using an integrated approach with Multibody Dynamics, Structural and Acoustic Finite/Boundary Element, and Psychoacoustic solvers leading to a Harmonic Acoustics simulation. By leveraging the capabilities of these simulation tools, we demonstrate the precise modeling of the bent-axis pump dynamics, and housing surface velocity to estimate the Equivalent Radiated Power (ERP) and its impact on noise level and quality at a specified distance. An initial correlation of the simulation model against real-world measurements, including microphone data, at the time of this abstract shows promise in the accuracy and reliability of the simulation process. The advantages of employing such a comprehensive model extend beyond technical precision, offering significant benefits in design optimization, manufacturing efficiency, and product development cycles. For pump manufacturers, this approach not only enhances product quality and compliance with noise regulations but also provides a faster go-to-market strategy. Furthermore, this work explores future avenues, including the integration of additional physics to refine the model's accuracy, positioning this methodology as a template for innovation in pump design and manufacturing.

Keywords NVH Workflow · Bent-Axis Pump · Test-vs-Simulation · Multibody Dynamics · Acoustics

Introduction

Accurate numerical models are crucial for the engineering industry for several reasons. These include:

Design prediction and optimization: Engineering simulations enable precise modeling of complex systems, allowing engineers to fine-tune designs and improve product performance [1].

Faster time-to-market: A comprehensive simulation approach accelerates the product development cycle, helping manufacturers bring products to market more quickly by reducing the need for iterative physical prototyping and testing [2].

Regulatory compliance and quality: Simulations ensure that products meet regulations (e.g., noise regulations) and maintain high-quality standards, reduce costs involved in certification testing, reducing potential manufacturing issues and enhancing overall product reliability [3].

The objective of this publication is to propose an effective NVH workflow for modeling the acoustic emissions of mechanical systems, with a focus on the bent-axis pump as a case study. This workflow integrates multiple physics-based solvers into a cohesive simulation strategy, providing accurate predictions of noise behavior and its psychoacoustic impact. Through this approach, we aim to present a robust methodology that can be extended to other complex machinery, offering insights into both the quantitative and qualitative aspects of noise emissions, while enhancing design optimization, product development cycles and compliance with noise regulations.

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Overview of the NVH Workflow

The NVH workflow comprises a structured approach divided into three key components: excitation, vibration response, and acoustics.[4] The process begins with excitation, which can stem from various sources such as electromagnetic analyses, dynamic outputs, or fluid pulsation. This excitation generates the necessary input for the subsequent step, where we calculate the vibration response of the enclosure (i.e. the physical structure or casing surrounding the system). This response is crucial, as it serves as the foundation for understanding how sound waves propagate through the surrounding air. Finally, this vibration response is fed into the acoustic domain to assess sound pressure levels at critical frequencies and to generate the far-field acoustic response spectrum. This comprehensive workflow allows for an in-depth analysis of sound power and sound pressure levels across a wide frequency range, providing valuable insights for optimizing NVH performance. One aspect of this workflow is that it accounts for excitations in the mid frequency range where the modal density of the structure is low enough to capture the vibro-acoustic response. For high frequencies ($>5\text{kHz}$), energy-based formulations such as SEA should be employed.

Excitation Stage

Ansys Motion Multibody Dynamics analysis (MBD) serves a pivotal role in the "excitation stage" of the NVH workflow of the bent-axis pump mechanism. This tool models the complex interactions and motion of various components within the pump, capturing the reaction forces acting at the interfaces between the mechanism and its housing, see Figure 1. By accurately simulating the kinetic behavior of the pump, MBD provides essential data on dynamic forces, which are critical inputs for subsequent stages of the workflow.

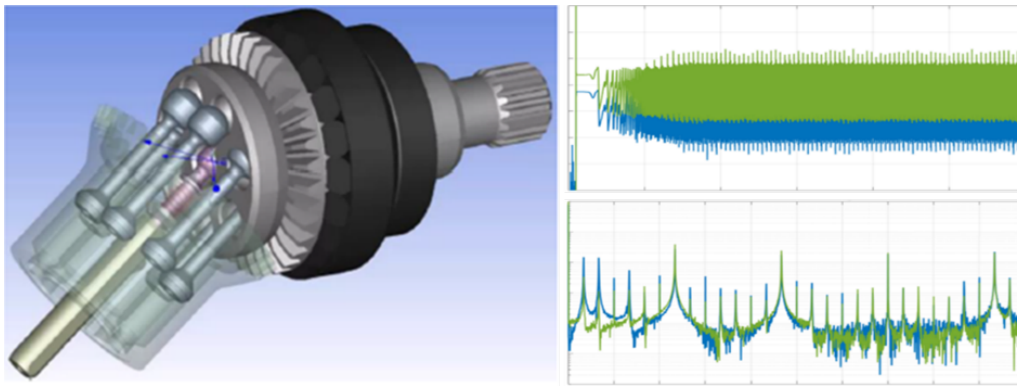


Fig. 1 Image of the bent-axis pump model (left) and a sample reaction force signal acting on interfaces between the mechanism and housing expressed in the time and frequency domains (left).

Vibration Response

The vibration response of the enclosure was calculated by performing a modal superposition using the modes of vibration and excitation forces to analyze the enclosure's behavior under dynamic conditions. The Modal Solver simplifies the enclosure's dynamic characteristics by expressing it as reduced model coordinates which efficiently captures its vibrational modes (see Figure 2). This reduced model, along with the reaction forces calculated during the excitation stage, serves as the input for the Harmonic Structural Solver. The Harmonic Solver then computes the velocity response of the enclosure's outer surface (see Figure 2), quantifying how the pump's operation affects the structure's vibrations. This critical output provides foundational data for the subsequent acoustic analysis.

Acoustics

In the Acoustics part of the NVH workflow, the velocity response of the enclosure's outer surface, obtained from the Harmonic Structural Solver, is utilized to assess sound emissions around the operating bent-axis pump. This data is input into acoustic solvers, which calculate the sound pressure levels (SPLs) at critical frequencies and evaluate the far-field acoustic

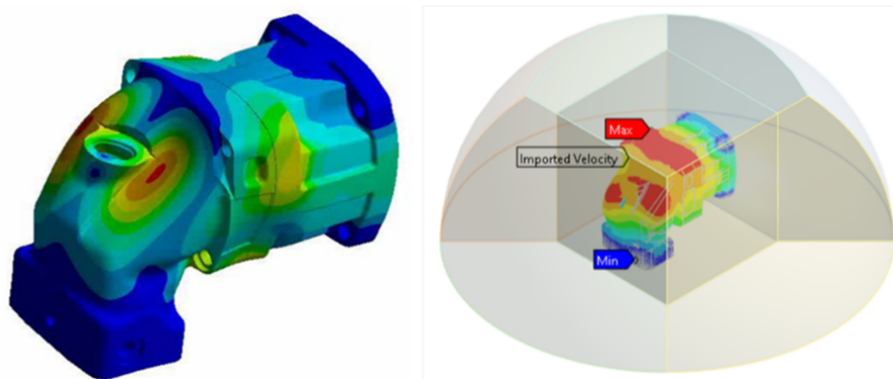


Fig. 2 One of the mode shapes computed by Ansys-Modal (left), and the velocity response of the enclosure's outer surface calculated by the Harmonic Structural solver and imported to the Acoustic solver (right).

response spectrum (as presented in Figure 3). By analyzing how the vibrations translate into sound, these solvers help quantify the noise impact of the pump on the environment, enabling engineers to identify potential areas for noise reduction and improve overall acoustic performance. This comprehensive analysis is essential for ensuring that the pump meets industry standards for noise, vibration, and harshness.

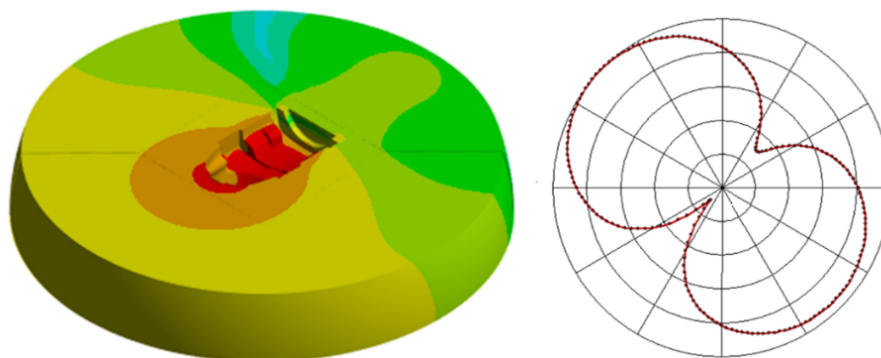


Fig. 3 Sample results from the acoustic solver. (Left) Sound pressure level at certain frequency and (right) the far-field directivity, which determines how focused the acoustic signal is.

Experimental Set-up

Experimental investigations were conducted in a semi-anechoic acoustic chamber at Parker Hannifin in Trollhättan, Sweden. Adjacent to this chamber is a separate room housing all the fixed equipment necessary to operate the pump of interest, including the drive motor, electric motor, and the pump shaft, as shown in Figure 4. The acoustic chamber contains the pump being tested, along with a microphone and amplifier (Brüel & Kjaer), as shown in Figure 4. Sound pressure data was acquired using the National Instruments system with a 50 kHz sampling rate.

The chamber is designed to be isolated from the surrounding environment, as illustrated in Figure 5. This separation helps minimize noise contributions from the factory and adjacent rooms. The acoustic chamber is lined with sound-absorbent materials that form a cone-like shape, as shown in Figure 5. Note that the floor does not have sound-absorbent materials; hence, it is a reflective surface (semi-anechoic room).

The room and measurements adhere to the International Standard ISO/DIS 3745, dated 1977-05-15. The microphone is placed 1 m away from the pump of interest. The facility can measure sound pressure with an accuracy of 1-3 dB for a frequency range of 125-10000 Hz. For the measurements and comparison against the NVH model, an F12-080 pump was used; F12 is a bent-axis fixed displacement pump (can operate as a motor) used for both open and closed-loop circuit applications (Figure 6).

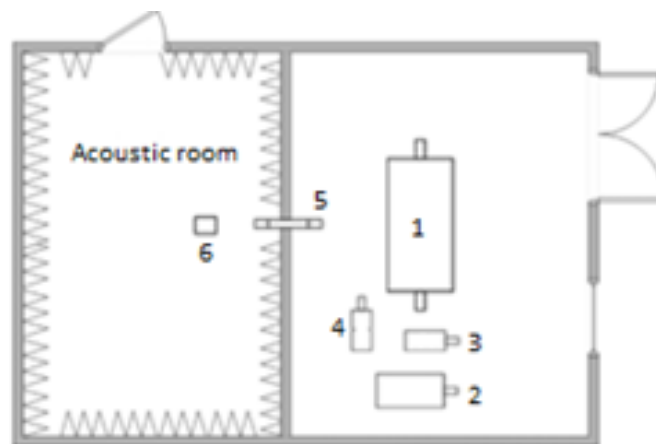


Fig. 4 Schematic view of facility to measure sound pressure. 1 – Drive motor; 2, 3, 4 – Electric motors with pumps; 5 – Shaft to acoustic room; 6 – Microphone and amplifier.

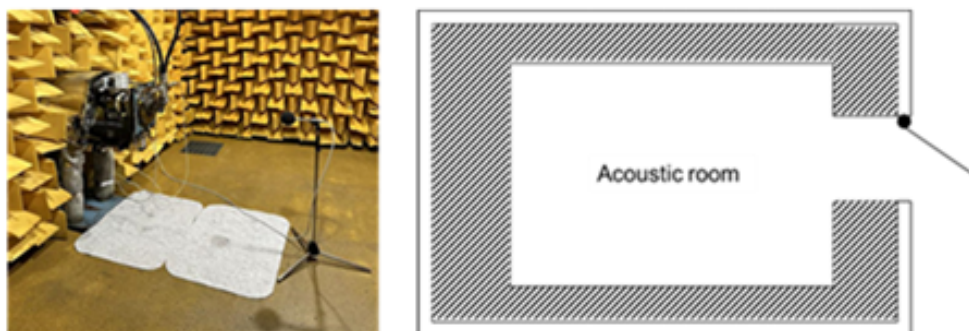


Fig. 5 (Left) Picture of test set-up and (right) Illustration of the acoustic room.

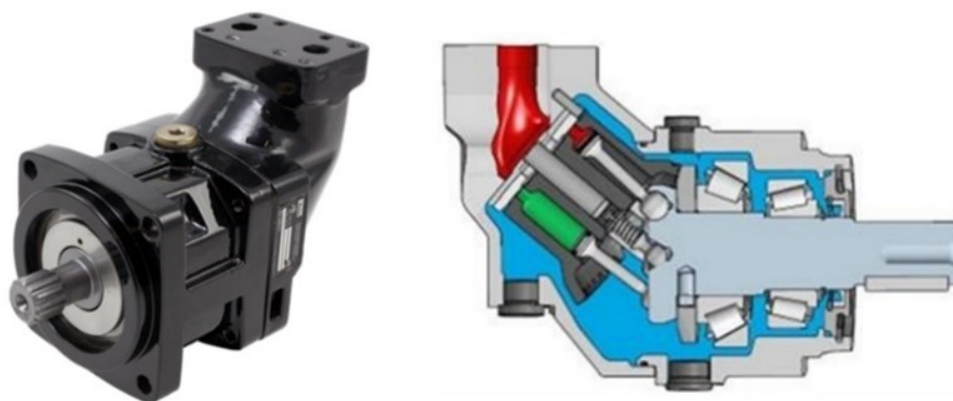


Fig. 6 Bent-Axis Fixed Displacement Pump - Series F12, Parker Hannifin.

A static and a speed sweep measurement were conducted. The static measurement involved recording sound pressure at several operating points; however, for the comparison, only data from operating points 2000 RPM and 200 Bar (outlet pressure) were used. The sound pressure was recorded with a sampling time of five seconds and a sampling rate of 50 kHz, and each operating point was measured twice. In the speed sweep campaign, measurements were carried out over approximately one minute, during which the outlet pressure remained constant (50 Bar and 200 Bar) while the speed varied from 100 to 2200 RPM and then back.

Test Data Interpretation

As previously covered in the experimental setup section, the acoustic chamber has a lower cut-off frequency of 125 Hz. The test cell also has ventilating fans and electronics that contribute to the overall acoustics of the chamber. An empty room run was conducted and the noise floor of the room was measured. It was determined that frequencies below 200 Hz have contributions from the chamber auxiliary components. The measured microphone data revealed there was a presence of both pump and motor orders present in the data. The test data used for this correlation was restricted only to the frequencies excited by the pump.

Preliminary Results Comparison and discussions

The boundary conditions of the test ensured that none of the initial excitation orders matched the resonant frequencies of the pump. Thus, the response obtained is purely forced for the frequency range of interest. The results for the major pump orders are compared. Initial results indicate that the difference between the test and simulation amplitudes for SPL are consistent across all frequency points, see Figure 7.

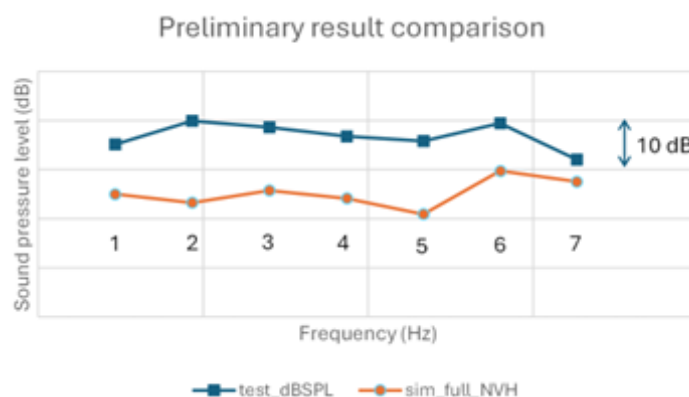


Fig. 7 Preliminary results comparison.

The higher frequency trends seen in the simulation model are in alignment with the test data obtained. For lower frequencies (Data point 1 and 2) the values follow opposite trends. The test cell floor as mentioned in the previous section is a reflective surface. At lower frequencies, the incident and reflected waves are in phase with each other since the wavelengths are large enough to constructively interfere with each other giving a higher amplitude. This phenomenon is not captured by the simulation model. At higher frequencies, the incident and reflected waves are out of phase with each other and hence do not amplify. Thus the behavior at high frequencies matches the simulation trends.

Future Work

Based on the results obtained from the preliminary correlation strategies to understand design sensitivity to acoustic response are conceptualized. Those are described below:

Impact of flexibility: The current simulation captures the forces and moments from the multi-body dynamics model when all inner parts are assumed to be rigid. This contributes to a conservative estimate of the dynamic interaction and thus the overall vibration response obtained at the enclosure. The impact on the acoustic response by considering crucial parts inside the mechanism as flexible is being studied.

Acoustic boundary conditions: As discussed in the preliminary results section, the impact of the reflective floor in the test cell amplifies the lower frequencies more in comparison to the higher frequencies. This can be studied by considering different acoustic mesh that captures the acoustic chamber conditions. A balance between acoustic domain size for accuracy vs computational resources available is important for this action.

Load and static pressure sensitivity: The current model setup considers the force interactions between individual components as well as an approximate force to capture the fluid pressure imparted on the fluid-structure interface. A load and pressure sensitivity analysis will help capture the impact of the load change on acoustic response change.

Fluid pressure fluctuations: The fluid pressure inside a positive displacement pump is a function of the rotational speed and the translational speed of the pistons. The fluid excitations will follow the same operating orders of excitations as the kinematic analysis. Due to the closed nature of the system, the fluid excitations can be used to study the vibrational response on the enclosure and the impact on the sound pressure level measured.

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Chapter 5

A Brief Review of Robotic Welding Technology for Structural Steel Applications

Ahmad Rababah, Feras Abla, Sahabeddin Rifai, Guilherme Pereira, Devin Huber, and Onur Avci

Abstract The increasing demand for faster and more economical construction of steel structures has positioned welding as a crucial element in both shop fabrication and site erection. This paper presents a brief summary of the application and feasibility of robotic technologies for automating the welding process. Welding is a cornerstone of modern steel construction, facilitating the connection of steel members for the creation of robust and durable structures. Due to the efficiency and significant contributions of stationary robotic welding systems and robotic welding arms, these technologies are expected to gain widespread popularity and become essential in the steel and metal industries. Well-aligned with the American Institute of Steel Construction's (AISC) "Need for Speed" initiative, robotic welding has the potential to bridge the gap between the industry's demand and the available qualified welders in the structural steel industry. This paper reviews the development of robotic and wall-climbing robot technologies, as well as the associated controlling tools such as sensors, cameras, GPS receivers, and simultaneous localization and mapping (SLAM) systems. These advancements indicate that robotics technologies are sufficiently advanced for implementation in structural steel applications at construction sites, with numerous autonomous robots having been designed and developed over the past decades.

Keywords Structural steel · Welding · Collaborative robots · Automated welding · Robotic welding

Introduction

The growing emphasis on reducing hazards and risks to individuals, enhancing productivity and efficiency, and optimizing both time and costs has led to the development and implementation of advanced technologies designed to assist with

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human tasks in manufacturing and industrial sectors. In the United States, and most other countries, the majority of structures, including bridges, high-rise buildings, and power and petrochemical plants, are typically constructed using structural steel. Recently, the steel shear wall system has been introduced as an effective alternative to conventional lateral load-resisting systems in the construction field. This system offers advantages such as a lighter structure, quicker construction, and superior quality compared to traditional reinforced concrete structures [1]. In addition, the implementation of continuous columns and interconnected column-beam systems is essential for maintaining structural integrity and ensuring optimal performance under various loading conditions and environmental circumstances. Components within a structure are often joined using bolts or welding techniques, including column splice welding and the welding of shear wall systems with steel frames. These methods effectively transfer vertical and horizontal loads between the connected elements. Welding, in particular, is a fundamental component in structural steel construction, enabling the creation of strong and durable structures. The automation of such an important component with mobile robots aligns with the American Institute of Steel Construction's (AISC) "Need for Speed" initiative, which emphasizes the importance of efficient and rapid construction methods.

In general, welding is a process of joining metals together using pressure or heat on the filler metal, in some methods, and workpiece to form the weld that can substitute the mechanical fastener connections. More than eighty different methods are used to achieve the welding [2]. Welding, brazing, and soldering are joining metal methods with different procedures. Brazing and soldering are characterized by melting the filler metals, but not the base metals as in the welding case. Neither brazing nor soldering is used for structural steel connections; as an alternative, welding is an appropriate option that offers the strongest joint to support the load. Typically, the welded section of a joint shows a greater strength than the base or original metal. However, imperfections that arise during or after the welding process, whether internal or external, can compromise the weld's performance and weaken the joints between metal components. Identifying and assessing these defects is crucial for ensuring quality control, thereby enhancing the weld's ability to perform its expected function effectively. Welding field-related workers need to comprehend the prevalent problems associated with welding in structures like causes, effects, and outcomes. Regulations and standards set by the American Welding Society (AWS) [3], are essential to guarantee the integrity and quality of weldment. Furthermore, the detection and inspection of defects are essential for addressing unacceptable defects that could affect the structural reliability of a welded structure. External defects, such as cracks, undercuts, overlaps, porosity, and spatters, are visible on the surface of the weld and can be detected by the human eye. Conversely, internal defects, which include incomplete penetration, inclusion, and lack of fusion, occur within the welding joint and are not visible to the naked eye. Detecting these internal defects requires specific methods including Non-destructive testing (NDT) [4], and destructive testing (DT). NDT can recognize the defects in welding by examining the workpiece without harming it or affecting its strength and reliability. Some of these tools and methods used in NDT are ultrasonic test (UT), radiography test (RT), electromagnetic test, eddy current test (ET), magnetic particle test (MT), acoustic emission (AE), dye penetrant (PT) and leak testing (LT). On the other hand, DT studies and tests the material or welded workpiece with some damages or destructions left on it, typically, such tests are conducted on small workpieces at the initial stages of the process, before welding and fabricating larger components. This approach enables an understanding of how the material will perform under various stress conditions.

Gas Metal Arc Welding (GMAW) and Gas Tungsten Arc Welding (GTAW) have traditionally been the most employed in robotic welding applications due to their compatibility with robotic systems [5]. However, developments in welding technology have expanded the range of methods engaged in robotic welding such as laser welding, plasma welding, metal inert gas welding, and spot resistance welding. A robotic welding system typically contains several critical components, including a mobile platform, manipulator, controller, welding torch, and an array of sensors. The mobile platform, or robotic base, serves as the primary structural frame of the system, supporting all other components and providing the capability to move, climb walls, and maneuver around obstacles. The manipulator is tasked with the precise movement and positioning of the welding torch, ensuring high-accuracy welding operations. Acting as the system's central intelligence, the controller processes input from the sensors and executes pre-programmed welding routines. This sophisticated organization of components not only enhances the precision and efficiency of welding tasks but also allows for real-time adjustments and optimization of welding parameters. In addition, the sensors play a crucial role in monitoring various parameters such as temperature, alignment, and weld quality, providing real-time feedback to the controller to ensure optimal performance and adjust parameters as needed [6].

Performing the welding tasks by using an automated system or robotic manipulator (arm), first developed by a Japanese company, Kawasaki in 1974 and used in the manufacturing of motorcycles has revolutionized the industry [7]. These automated systems offer numerous advantages, including enhanced productivity, improved quality, increased safety, and significant time and cost savings. Additionally, robotic welding systems can operate under extreme conditions, such as high temperatures and high pressures, which pose substantial risks to human workers, especially those working at high elevations in structural steel construction.

Robotic Welding

Robotic welding manipulators have achieved widespread adoption and have become the industry standard in manufacturing and related sectors. Initially designed as stationary-mounted robots, welding robots became highly popular in industries with active production lines and substantial welding tasks, such as metal fabrication, aerospace, and automotive manufacturing. These robots are programmed to perform repetitive tasks with high levels of accuracy and consistency, following predetermined routines that specify the direction, distance, velocity, acceleration, and deceleration of coordinated motion sequences [8]. This precision and reliability make robotic welding an ideal solution for high-volume production environments.

The implementation of robotic welding in the construction field offers significant advantages. One of the primary benefits is addressing the gap between the demand for skilled welders and the available workforce, a global issue highlighted by the American Welding Society (AWS). By deploying robots, the construction industry can mitigate this shortage, ensuring projects are completed efficiently and on schedule. Additionally, robotic welding enhances the quality and consistency of welds, reduces human error, and improves overall safety on construction sites. This technological advancement not only meets the immediate labor needs but also supports the industry's long-term goals of productivity and innovation. In addition, using innovative systems and technologies in the construction field contributes to the optimization of resources and reduction of waste, supporting the industry's commitment to sustainable development.

In the second stage of development, the stationary welding arm was advanced to create a more user-friendly, collaborative welding robot, commonly known as a "Cobot". In contrast to traditional welding arms, Cobots prioritize harmonious interaction between humans and robots, with a strong emphasis on safety and collaboration. These robots are equipped with advanced sensors, force and torque detection, and responsive performance features to ensure safe and effective human-robot interaction. The core idea behind this collaboration is to merge human creativity and decision-making abilities with cutting-edge technological systems. This interaction aims to advance a more innovative, efficient, and adaptable industrial environment, ultimately enhancing productivity and workplace safety [9]. The collaboration between Cobots and humans is facilitated through the application of machine-learning tools that control and train robots [10]. By leveraging machine learning and other decision-making algorithms, Cobots can learn from human actions, adapt to various tasks, and detect human moves which provides a safer environment to enable Cobots to work alongside human operators more effectively, enhancing precision and efficiency in welding operations. Generally, cobots are smaller in size compared to the traditional arms and designed for lighter repetitive welding tasks, more cost-effective, versatile, and simpler programming and reprogrammed for different tasks. Recently, several companies were able to develop collaborative welding robots such as the Cooper welding cobot which was designed by Lincoln Electric [11], the Copilot Cobot by Miller Electric [12], and the Cobot by ESAB [13]. The primary role of human welders collaborating with Cobots is to supervise the operational processes, ensure the quality of the welds, and make essential adjustments to the Cobots' programming to execute precise welding tasks [10].

Wall-Climbing Robots and Sensor Technologies

The current vision focuses on developing smaller autonomous robots capable of performing welding tasks in various locations and under different conditions. Over the past few decades, significant advancements have been made in creating wall-climbing robots, resulting in the emergence of hundreds of prototype systems. These innovations pave the way for designing specialized welding robots, capable of navigating and operating in challenging environments. For a welding robot, the payload is critical, alongside mobility, as the robot must be able to carry both the welding equipment and the manipulator. Considering that welding will be performed on ferromagnetic surfaces, and given the stability, smooth mobility, and lightweight nature of wheeled robots compared to legged robots, the use of wheeled robots is recommended and widely adopted for wall-climbing applications. Magnetic wheeled robots are particularly suitable for this purpose. These robots can adhere to and navigate vertical and inclined surfaces efficiently, making them ideal for welding tasks in such challenging environments. The adoption of magnetic wheeled robots in welding applications not only enhances operational stability and precision but also expands the scope of automated welding to include complex and hard-to-reach areas. In 1995, the Kansai Research Institute developed dual magnetic wheels to address the challenge of traveling on discontinuous surfaces and pipes with bending joints, passively without any external source of energy [14]. Electromagnetic wheels can mitigate some limitations associated with permanent magnetic or passive mechanisms, such as the ability to navigate sharp obstacles or transition between different or perpendicular surfaces. When using passive systems, the robot may be trapped in corners due to the equivalent of magnetic forces from both surfaces. To address this issue, an active magnetic wheel mechanism was developed, incorporating a controlled lifter-stabilizer that facilitates climbing and traversing obstacles such as perpendicular surfaces [14]. Eto and Asada designed a wall-climbing robot for welding which uses two free rocker-arm hovering

mechanisms that have magnetic spherical wheels. It can navigate obstacles that are 50 mm high and 90-degree corners [15]. Wu et al. developed a wall-climbing welding robot with non-contact permanent magnetic adhesion. Where the magnetic suckers are positioned beneath the axles with an adjustable gap, this design significantly enhanced the robot's performance [2]. In 2022 Zhang et al. improved the flexibility and stability of the robot by replacing the contacted traditional magnetic circuit on the wheels with a newly designed permanent magnetic chuck [16], which has been proven to increase the adsorption force by 9% compared to the traditional magnetic circuit [17]. A design of three divided body parts connected with flexible joints and a six-wheel robot was proposed to provide more motion flexibility and the ability to maneuver and travel through convex and concave surfaces, wall and ceiling climbing robot [18]. Franko et al. in 2020 designed a magnetic wall climbing robot, highly effective for turbine maintenance, a robot was provided with a LiDAR vision sensor for inspection and a fixed mainframe with a non-contact magnet to a climbed surface and rotating wheels to provide motion capability to the robot [1].

Moreover, various robots utilizing different technologies are currently available in the market. For instance, Sumitomo Heavy Industries, a Japan-based company, introduced a magnetic spherical wheeled climbing robot in 2022. This robot is capable of navigating ferromagnetic structures on vertical, horizontal, and curved surfaces. However, its traveling and climbing mechanisms are constrained by the material and shape of the surfaces it climbs [19]. In 2021, Gecko Robotics, a United States located company, successfully developed the latest advanced robot, the TOKA Flex, from the Toka series. This cabled robot is designed to inspect and gather data on ferromagnetic surfaces and was primarily created for application in industrial plants to perform non-destructive testing (NDT) inspections. This is achieved through the utilization of an ultrasonic transducer, which is specifically designed for inspecting pipes and tanks. Toka Flex can climb 75 ft in height and can climb pipes 6 in diameter by using its four magnetic wheels [20]. Furthermore, for any smooth non-ferromagnetic surfaces such as glass, plastic, and aluminum. The world's first non-magnetic climbing inspection robot was developed in New Zealand by Invert Robotics Company [21] which developed a series of climb robots that work for non-ferromagnetic surfaces by using multiple systems like sliding suction adhesion, adhesion pads, vacuum pumps, and magnetic systems.



Fig. 1 (a) Sumitomo Heavy Industries climbing robot [19]. (b) The TOKA Flex Robot [20].

To effectively operate welding robots, a comprehensive array of devices and equipment is essential, including an Inertial Measurement Unit (IMU), GPS receiver, and multiple cameras, mounted on the manipulator. Additionally, tactile sensors and encoders on each servo joint are crucial components of the robot's instrumentation. These devices facilitate real-time control, enabling precise and efficient operation. For precise robot localization, a simultaneous localization and mapping (SLAM) system can be employed, which integrates data from both inertial sensors and cameras. This approach significantly enhances the robot's ability to navigate and map its environment with precision [22]. SLAM systems allow a robot to plan a course through an unfamiliar environment while simultaneously identifying its location within that environment. To execute the SLAM process, robots must be equipped with a range of sensors to gather various measurements along their trajectories. The primary sensors utilized include cameras, LiDAR, Global Navigation Satellite System (GNSS), and IMU. When a SLAM algorithm mainly uses camera sensors, it is termed visual SLAM, whereas the use of laser scanners characterizes it as LiDAR SLAM [23]. Particularly, LiDAR SLAM enables robots to progressively determine their movement by matching scanned 3D points as they navigate. During their journey, robots estimate the distance traveled to determine their current position. Simultaneously, they generate a map of their environment by aligning these scanned points, typically utilizing the

Iterative Closest Point (ICP) algorithm to refine this alignment and enhance the accuracy of both the map and their positional estimates [24]. However, to achieve effective automated welding, it is essential to have an appropriate pre-trained program to ensure that the welding process can be monitored in real-time by various sensors. These sensors, integrated with software and neural networks, play an important role in controlling process parameters and predicting whether the final weld will be acceptable or defective. If the modeling predicts a potential defect during the welding process, it must intervene to adjust the process parameters and prevent any defects from occurring.

Overview of Mobile Robots in the Market

In recent decades, there has been a significant surge in the development and deployment of autonomous mobile robots. These robots have seen widespread application across various industries, including construction, agriculture, service, medical, and manufacturing sectors, among others. This growth is driven by developments in sensor technology equipped with onboard artificial intelligence and machine learning tools, which have enhanced the capabilities of these robots, enabling them to perform complex tasks with increasing efficiency and precision.

Mobile Welding Robots

In particular, the field of welding has seen numerous companies start the development of automated welding robots. Several countries have emerged as leaders in this endeavor, making significant progress in the advancement of robotic welding technology. The SIASUN company (based in China) has engineered a variety of mobile robots for various applications, including assembly, forklift operations, and material transfer. In 2024, SIASUN further expanded its robotic solutions by introducing a welding robot, including combined functionality in robotics, 3D vision, SLAM navigation, and artificial intelligence to develop the robot [25]. Moreover, for welding tasks, the O-Matic Corporation (based in China) has developed a mobile welding robot that integrates a mobile robotic platform featuring a 6-axis collaborative robot, alongside a welder and a water cooler. This sophisticated robot is capable of navigating construction sites with precision and executing welding tasks according to pre-programmed trajectories [26].

Additionally, Alpha Laser (based in the USA) company, was able to design the AL-ROCK Mobil, a highly advanced mobile robot equipped with a 4-kW laser, specifically designed for surface metal hardening and welding with wire or powder [27]. This system is fully mobile, as all essential components, including the laser, cooler, and robot controller, are integrated within the robot mobile frame. The Construction Innovation and Technology Fund (CITF) (based in China) has introduced the Welbot, an on-site mobile welding robot featuring advanced control software [28]. This innovative system is capable of auto-generating weld paths, performing 3D welding weaves, and managing complex, non-linear welding trajectories. Additionally, in Germany, the Intelligent Mobile Systems Lab (IMSL) is engaged in a project scheduled for completion in 2024, aimed at developing an omnidirectional manipulator for a mobile robot welding system. This innovative setup consists of an omnidirectional mobile platform and a robot arm that collaboratively performs continuous movements. The system is designed for various tasks such as spray painting, welding, and validating parts [29].

Mobile Robots in Diverse Sectors

In the construction field, a wide range of robots have been developed to perform various specialized tasks. These include robots designed for painting, drilling, welding, and cutting, as well as those intended for management and supervising purposes. Dusty Robotics, based in California, has developed the FieldPrinter 2, a state-of-the-art autonomous layout robot designed to revolutionize on-site performance, efficiency, and productivity [30]. This next-generation innovation features a compact design and enhanced capabilities, signifying a substantial advancement in construction automation technology. Okibo has proposed a mobile multi-purpose autonomous robot adapted for construction sites, an autonomous finishing robot, including wall plastering tasks, such as stucco, EIFS, concrete, primer, and adhesives [31]. In 2020, Hilti introduced its first robot, Jaibot, designed for semi-autonomous mobile ceiling-drilling operations. This cordless system offers ease of use, achieves precise indoor localization, and conducts dust-controlled drilling activities [32]. Moreover, the "CAMERA" project (Construction And Manufacturing Enabled by a Mobile Robotic Arm) represents a collaborative effort between ABB, HAL Robotics, Skanska, and InnoTech UK in developing this robotic system. The robot consists of three distinct components: a mobile base, a scissor lift, and a robotic arm manipulator designed to maneuver tools. Tool paths are generated from a CAD model, while a localization system guides the robot to ensure precise execution of tasks, as it was engineered to perform a range of construction activities [33].

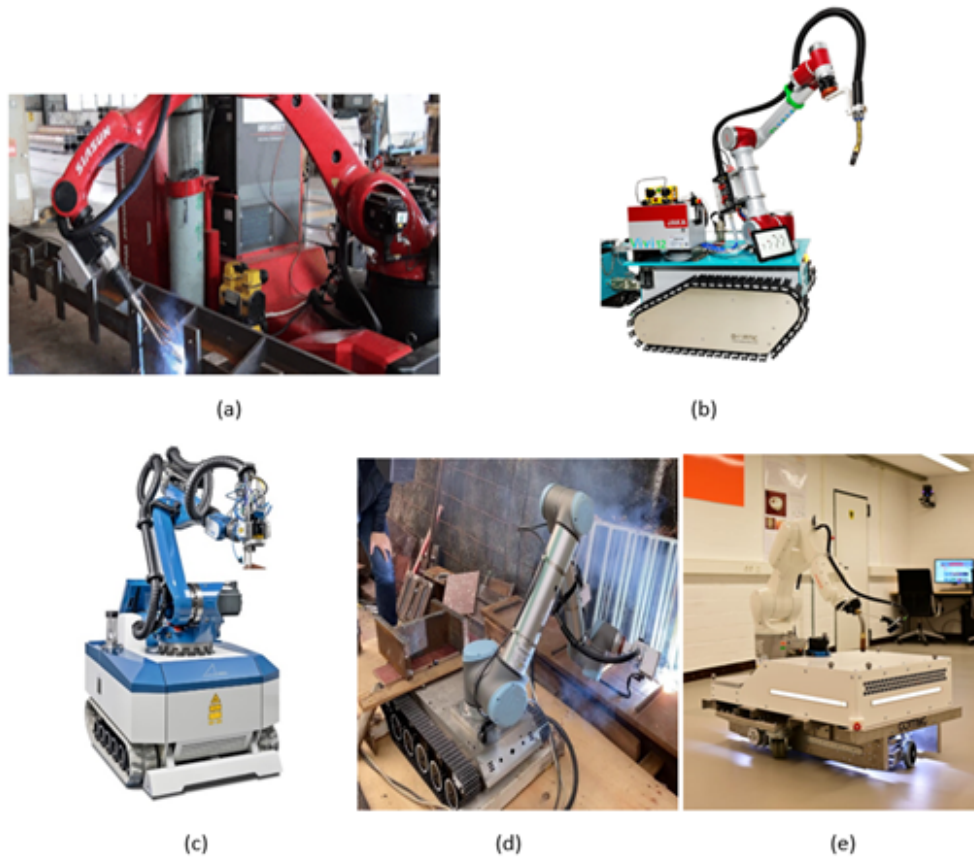


Fig. 2 (a) SIASUN robot [25]. (b) O-Matic robot [26]. (c) AL-ROCK robot [27]. (d) CITF Wellbot robot [28]. (e) IMSL Project [29].

LiDAR-Equipped Mobile Robots

As discussed in Section 3, the implementation of advanced sensors and cameras is essential for the effective control and operation of autonomous robots. Doxel's LiDAR-equipped robots, for instance, capture comprehensive visual and spatial data of construction sites, providing detailed reports to project teams [34]. Similarly, the University of Edinburgh's TrimBot2020 utilizes a camera, stereo 3D vision technology, and LiDAR for autonomous pruning of roses and bushes, enabling the robot to recognize and distinguish between plant parts and ensure safe and efficient operation [35].

Furthermore, the Technical University of Darmstadt's Team Hector has developed a variety of robotic systems for search, rescue, and emergency purposes. These robots demonstrate proficiency in generating real-time 3D environmental maps, facilitated by a rotating Velodyne VLP-16 LiDAR for creating detailed 3D point clouds, complemented by an omnidirectional camera and an RGBD camera [36]. By combining these sensors and technologies, these autonomous robots can effectively navigate and operate in diverse environments, providing valuable data and services to their respective fields.

Challenges and Discussion

Despite the shortage of skilled welders, as reported by the American Welding Society (AWS), there are significant concerns about automating welding jobs with robots. Critics argue that robots will replace human welders, thus reducing job opportunities. However, it is essential to clarify that even with the deployment of welding robots, these machines cannot perform the entire task autonomously. Skilled experts are still needed to oversee, control, and make critical decisions during the welding process. Therefore, the role of human welders will evolve rather than disappear, with a greater emphasis on supervisory and decision-making responsibilities. This collaboration between human expertise and robotic efficiency can lead to enhanced

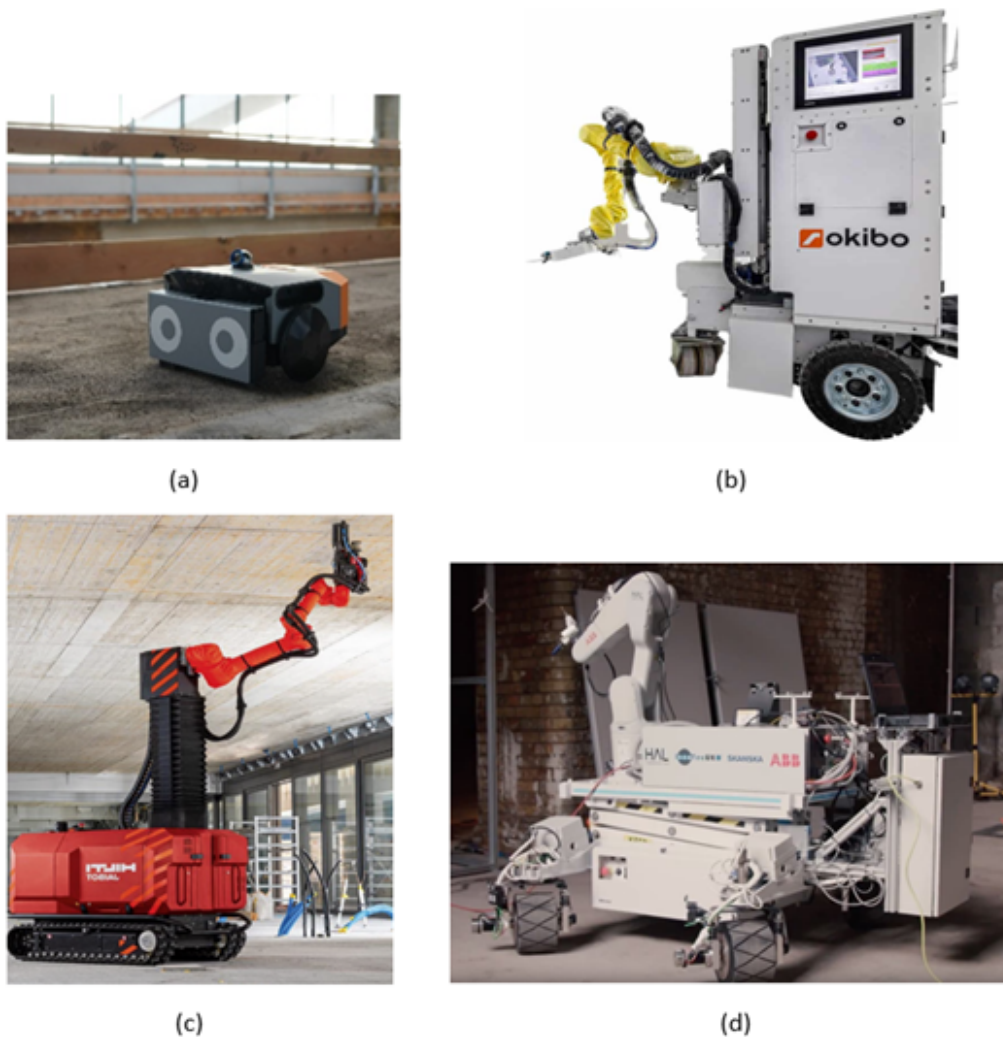


Fig. 3 (a) Dusty FieldPrinter2 robot [30]. (b) Okibo finishing robot [31]. (c) Hilti drilling robot [32]. (d) CAMERA robot [33].

productivity, job quality, and improved worker safety in the welding industry in addition to addressing the labor shortage issue. Moreover, it offers mutual benefits, enabling companies to meet production demands more effectively and providing skilled welders with opportunities to focus on higher-level tasks such as supervision and decision-making, rather than manual welding tasks.

Furthermore, the power supply for mobile robots is a crucial consideration in their design. Typically, most of the wall-climbing robots have been powered either by linking them to tethers or by using built-in batteries. Tethers also help with safety, however, restrict the robot's accessibility and the height of surfaces it can climb. Conversely, batteries also pose limitations due to their limited operation time. Currently, there are no small-sized batteries that offer extended operation times; larger or heavier batteries increase the robot's weight, leading to additional challenges like the weight of the robot. This is a critical factor, as it affects the robot's payload capacity, which must accommodate essential tools such as the torch, manipulator, and welding equipment. Furthermore, the robot's ability to climb surfaces and its stability depends on its weight and the strength of its climbing mechanism, which might utilize magnetic or electromagnetic wheels. The strength and characteristics of the surfaces the robot climbs are also valuable considerations. Balancing these factors is key to designing an effective and efficient mobile robot capable of performing complex tasks in various environments. Self-powered or power-harvesting solutions can be a significant step toward solving the power supply problem for mobile robots, offering increased flexibility and mobility. By harnessing energy from the environment or through innovative self-charging mechanisms, these solutions can extend the operational time of robots without the need for heavy or large batteries in addition to enabling robots to navigate and operate more freely, without the limitations imposed by tethers or frequent recharging requirements.

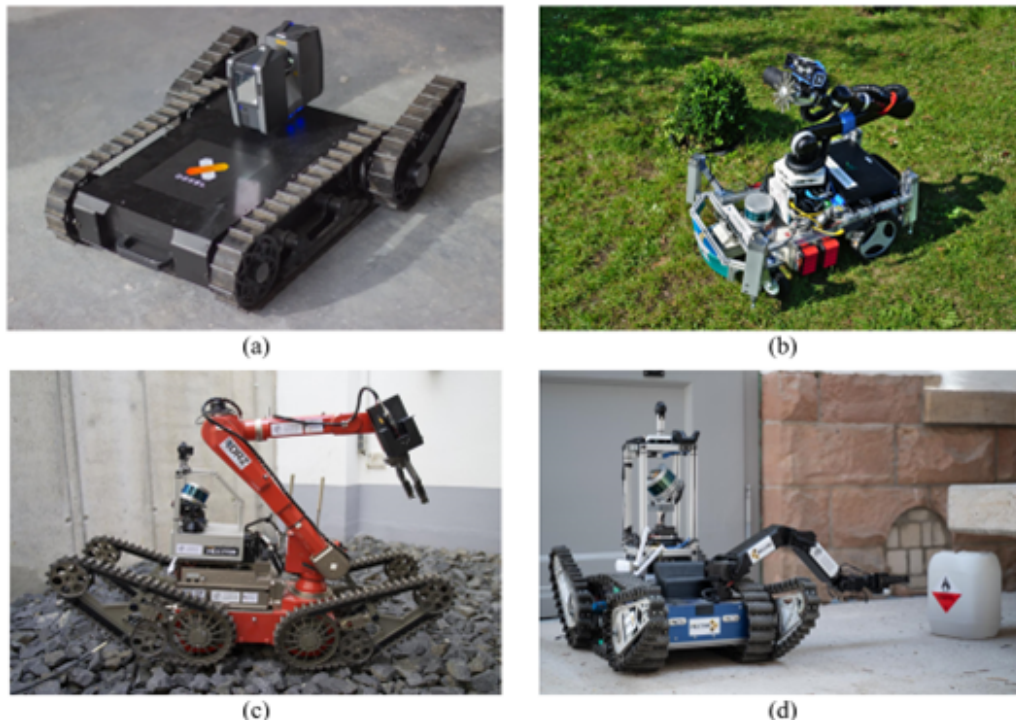


Fig. 4 (a) Doxel's Robot [34]. (b) TrimBot2020 Robot [35]. (c) Hector DRZ Telemax [36]. (d) Hector Asterix [36].

Conclusions

This study offers a thorough review of recent advancements in robotic design across various fields, emphasizing the latest developments in automating welding tasks and innovations in robotic welding. It also examines progress in sensor technologies, such as enhanced vision systems, localization techniques, and tactile sensors. These advancements have greatly enhanced robots' ability to perceive and interact with their environments, enabling precise navigation, accurate task execution, and improving reliability in autonomous welding operations. Furthermore, this review explores mobility techniques tailored to welding robots, focusing on wall-climbing methodologies and related innovations. Wheeled robots utilizing magnetic and electromagnetic systems have emerged as suitable approaches for welding tasks, offering the necessary stability and maneuverability on vertical surfaces, although they may face challenges when crossing beam intersections. However, with the continuous advancements in robotic technology and design, it is expected that the complexities of beam intersections will be effectively addressed in future iterations, enhancing both the adaptability and efficiency of robotic welding solutions.

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Chapter 6

Bridging Minds and AI for Enhanced Autonomous Driving



Roveri Nicola, Silvia Milana, Antonio Culla, Gianluca Pepe, and Antonio Carcaterra

Abstract This paper studies some aspects of a larger a ground-breaking research initiative aiming to advance autonomous driving through the integration of neurotechnology. This multidisciplinary approach combines mechanical engineering with cutting-edge monitoring technologies to develop autonomous driving algorithms that closely mimic human behaviour and reactions. By equipping drivers with state-of-the-art sensors to monitor brain activity, eye movements, heart rate, and skin conductivity, comprehensive data on driver responses is gathered in realistic driving scenarios.

The primary objective is to bridge the gap between human understanding of driving and the capabilities of autonomous systems. Through the detailed analysis of the collected data, the project aims to identify patterns and reactions that can be incorporated into autonomous driving algorithms, enhancing their intuitiveness, safety, and alignment with human behaviour. This project addresses significant gaps in the current literature and opens new avenues for the development of more effective autonomous driving systems. With this purpose, the paper studies the opportunity to integrate information from brain signals, extracted from a dataset, into the control system of the vehicle guidance system.

Keywords Neurotechnology · Brain activity · Autonomous driving · Optimal control theory · Dynamics

Introduction

In this article, an adaptive control strategy that integrates neurofeedback into vehicle guidance systems is proposed. The objective is to evaluate the efficacy of a brain-assisted control system by simulating a simplified two-dimensional obstacle avoidance problem. The vehicle is modeled as a point in a plane, tasked with moving from a starting point A to a target point B while avoiding obstacles [1, 2]. Obstacles are represented using a potential field model, where attractive forces guide the vehicle toward the goal and repulsive forces push it away from obstacles.

In recent years, the integration of brain signals into autonomous driving systems has generated increasing interest within the scientific community. Several studies have explored the use of Electroencephalography (EEG) to enhance the safety and efficiency of autonomous vehicles. Another noteworthy approach was presented in [3], where the driver's brain waves were used to improve emergency braking by combining EEG signals with information from environmental sensors. Additionally, in [4], a semi-autonomous control system was developed in which the vehicle can be driven through mental commands detected by EEG.

Other studies have investigated human-vehicle collaboration through hybrid brain-computer interfaces (BCIs) and computer vision systems. In [5], a collaborative driving system was proposed where decisions on the path are entrusted to the driver via BCI, while the computer vision system keeps the vehicle on the road and avoids collisions. Furthermore, in [6], the use of EEG was studied to monitor brain activity related to errors during driving, both in a simulator and in a real car, highlighting the potential of EEG in detecting mental states critical for road safety. Finally, in [7], the applicability of a dry EEG system in passive BCI for autonomous driving was evaluated, demonstrating the feasibility of using non-invasive and user-friendly EEG devices in real driving contexts.

Despite these advancements, challenges remain in effectively integrating brain signals into vehicle control systems. This work distinguishes itself by utilizing neurophysiological signals to modulate in real time the parameters of the vehicle's adaptive control, rather than merely monitoring the driver's state or issuing discrete commands. This deep integration

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between neurotechnology and control engineering represents a step forward toward the development of more intuitive and safer autonomous vehicles.

For this study, the MPDB dataset [4] was considered. Physiological signals measured through EEG, Electrocardiography (ECG), Electromyography (EMG), Galvanic Skin Response (GSR), and eye movement data in a six-degree-of-freedom (6DOF) driving simulator were correlated with driving behavior categorized into five groups: Smooth Driving (SD), Acceleration Driving (AD), Deceleration Driving (DD), Lane Changing Driving (LCD), and Turning Driving (TD) [8]. The trajectories obtained from a classical Linear Quadratic Regulator (LQR) control system are compared with the solution provided by an adaptive version of the LQR [9, 10] where the control gains are modulated based on the vehicle's proximity to obstacles.

Background

The obstacle avoidance problem is formulated using a potential field approach, which defines a composite potential $U(\underline{q})$ for the vehicle at position $\underline{q} = (x, y)^T$ in a two-dimensional space. The potential is a combination of an attractive field guiding the vehicle toward the goal $\underline{q}_{goal}$ and a repulsive field pushing the vehicle away from obstacles.

The total potential at position is defined as:

$$U(\underline{q}) = U_{att}(\underline{q}) + U_{rep}(\underline{q}) \quad (1)$$

where U_{att} is the attractive potential toward the target, and U_{rep} is the repulsive potential generated by obstacles.

The force acting on the vehicle at the position $\underline{q}$ is due to the gradient of the total potential defined before:

$$\mathbf{F}(\underline{q}) = -\nabla U_{att}(\underline{q}) - \nabla U_{rep}(\underline{q}) \quad (2)$$

where ∇ is: $\nabla(\bullet) = (\partial(\bullet)/\partial x, \partial(\bullet)/\partial y)^T$.

Let be $\underline{q}_{goal}$ the target point, the attractive force is modelled as:

$$\underline{F}_{att}(\underline{q}) = -k_{att}(\underline{q} - \underline{q}_{goal}) \quad (3)$$

where k_{att} is a scalar that acts as a weight by defining the strength of attraction and has to be set adequately.

The repulsive force is active only when the vehicle is within a certain distance $\rho(\underline{q})$ from an obstacle. The repulsive force is defined as:

$$\underline{F}_{rep}(\underline{q}) = \begin{cases} k_{rep} \left(\frac{1}{\rho(\underline{q})} - \frac{1}{\rho_0} \right) \frac{\underline{q} - \underline{q}_{obs}}{\rho(\underline{q})^3} & \text{if } \rho(\underline{q}) < \rho_0 \\ 0 & \text{if } \rho(\underline{q}) > \rho_0 \end{cases} \quad (4)$$

where $\underline{q}_{obs}$ is the point of the obstacle with minimum distance, *i.e.* the nearest obstacle, $\rho(\underline{q}) = |\underline{q} - \underline{q}_{obs}|$ is the distance from an obstacle, ρ_0 is distance threshold, k_{rep} is a weight factor and has to be set adequately.

The resultant force $\underline{F}(\underline{q})$ acting on the vehicle is given by: $\underline{F}(\underline{q}) = \underline{F}_{att}(\underline{q}) + \underline{F}_{rep}(\underline{q})$. This force governs the direction and magnitude of the vehicle's movement in the plane.

The system state is represented by $\underline{z}(t) = (\underline{\dot{q}}, \underline{q})^T$. The state dynamics is described by the linear system:

$$\dot{\underline{z}}(t) = \mathbf{A} \underline{z}(t) + \mathbf{B} \underline{u}(t) + \underline{n}(t) \quad (5)$$

where $\underline{u}(t)$ is the control input, directly related to $\underline{F}$, and $\underline{n}(t)$ is an external noise.

The objective of control is to impose on a system a desired behaviour. The studied type of control is the feedback control: that is, one must determine the law that binds $\underline{u}(t)$ with the state vector $\underline{z}(t)$. The theory of optimal control has as its objective the minimisation-maximisation of some performance criterion. To realise optimal control a variational approach consisting in the minimisation of a functional J is performed.

In general J is defined as an integral that extends throughout the observation time of the phenomenon, T , the whose integrand is the Lagrangian function $\mathcal{L}(\underline{z}, \underline{u}, \underline{n})$ which depends on the state, the control and the external noise.

In the classical Linear Quadratic Regulator (LQR) framework, and the objective function J is: $\underline{z}^T \mathbf{Q} \underline{z} + \underline{u}^T \mathbf{R} \underline{u}$. The term $\underline{z}^T \mathbf{Q} \underline{z}$ measures the benefit, the smaller it is, the better the performance; the term $\underline{u}^T \mathbf{R} \underline{u}$ measures the control cost to obtain the benefit. Therefore, the problem can be written as follows:

$$\begin{cases} \min J = \int_0^T (\underline{z}^T \mathbf{Q} \underline{z} + \underline{u}^T \mathbf{R} \underline{u}) dt \\ \dot{\underline{z}} = \mathbf{A} \underline{z} + \mathbf{B} \underline{u} + \underline{n}, \quad \underline{z}(0) = \underline{z}_0 \end{cases} \quad (6)$$

By using the Lagrange multipliers the following relationship is obtained:

$$\tilde{J} = \int_0^T [\underline{z}^T \mathbf{Q} \underline{z} + \underline{u}^T \mathbf{R} \underline{u} + \lambda^T (\dot{\underline{z}} - \mathbf{A} \underline{z} - \mathbf{B} \underline{u} - \underline{n})] dt \quad (7)$$

A solution strategy using feedback control allows to write:

$$\begin{cases} -\dot{\mathbf{K}} + \mathbf{A}^T \mathbf{K} + \mathbf{K} \mathbf{A} + \mathbf{K} \mathbf{B} \mathbf{R}^{-1} \mathbf{B}^T \mathbf{K} - \mathbf{Q} = 0, & \mathbf{K}(0) = 0 \\ -\dot{\underline{p}} + \mathbf{A}^T \underline{p} + \mathbf{K} \mathbf{B} \mathbf{R}^{-1} \mathbf{B}^T \underline{p} + \mathbf{K} \underline{n} = 0, & \underline{p}(0) = 0 \end{cases} \quad (8)$$

The first equation is the Riccati's equation and $\mathbf{K}$, solution of the Riccati's equation, is the gain matrix.

In this formulation, the matrices $\mathbf{Q}$ and $\mathbf{R}$ are constant, meaning that the controller uniformly balances trajectory tracking (reaching the goal) and control effort throughout the entire task.

Analysis

Numerical simulations are performed to compare the performance of the standard LQR controller with constant $\mathbf{Q}(\underline{z})$ and $\mathbf{R}(\underline{z})$ matrices against an adaptive LQR controller with state-dependent matrices. The scenarios involve a vehicle navigating through a field of obstacles to reach a predefined target. Metrics such as time to target, control effort, and proximity to obstacles are analyzed to assess the effectiveness of the adaptive controller.

To simulate the effect of brain signals, we introduce a state-dependent LQR controller. The idea is that brain signals modulate the vehicle's behaviour based on its proximity to obstacles. As the vehicle approaches an obstacle, the driver becomes more alert, and the control system prioritizes obstacle avoidance. Conversely, when the vehicle is far from obstacles, the system focuses on target tracking. In this adaptive LQR, the matrices $\mathbf{Q}$ and $\mathbf{R}$ become functions of the distance to the nearest obstacle $\rho(\underline{q})$. Matrix $\mathbf{Q}(\rho)$ penalizes the vehicle deviation from the desired state more heavily when near obstacles:

$$\mathbf{Q}(\rho) = \mathbf{Q}_0 + \Delta \mathbf{Q} \sigma(\rho) \quad (9)$$

where $\mathbf{Q}_0$ is the nominal weight matrix, $\Delta \mathbf{Q}$ is the additional penalty when near an obstacle, $\sigma(\rho)$ is a *sigmoid* function of the distance ρ to the nearest obstacle.

Matrix $\mathbf{R}(\underline{q})$ reduces the penalization on control effort near obstacles to allow for more aggressive corrections:

$$\mathbf{R}(\rho) = \mathbf{R}_0 + \Delta \mathbf{R} (1 - \sigma(\rho)) \quad (10)$$

where $\mathbf{R}_0$ is the nominal control cost, $\Delta \mathbf{R}$ is the reduction in control cost near obstacles.

The control input is still of the form $\underline{u}(\underline{z}, t) = \mathbf{R}^{-1} \mathbf{B}^T (\mathbf{K}(\underline{z}, t) \underline{z})$, but the gain matrix $\mathbf{K}(\underline{z}, t)$ now depends on the state-dependent matrices $\mathbf{Q}(\rho)$ and $\mathbf{R}(\rho)$, calculated in real-time by solving the Riccati's equation. As the vehicle approaches an obstacle, the controller automatically shifts its behavior, focusing more on avoiding the obstacle and less on reaching the goal. This dynamic adjustment simulates the heightened alertness of a human driver in critical situations.

Physiological signals and driver behavior

The brain signals are extracted by the MPDB dataset. This one gives a set of physiological signals correlated to corresponding driver's reaction. The environment where the experiment is performed is a 6DOF driving simulator and the signals are measured by EEG, ECG, EMG, GSR and eye track [8]. The experiment is performed showing a set of events to the driver and registering his reaction. The event and the driving behavior are correlated as shown in the following correspondence table (1).

Table 1 Corresponding table [8]

Driving behavior	Corresponding events
Smooth driving	Normal straight line driving
Acceleration	Overtaking Congestion relief
Deceleration	The front emergency brake The slide front cut-in Pedestrian crossing
Lane-change	Static obstacle on right Static obstacle on left
Turning	Left-turn sign Right-turn sign

Conclusion

This paper introduces a novel adaptive LQR control strategy that integrates simulated brain responses into a vehicle's obstacle avoidance behaviour. By dynamically adjusting the weight matrices $\mathbf{Q}$ and $\mathbf{R}$ based on the vehicle's proximity to obstacles, the controller mimics the human driver's heightened attention in critical situations. Numerical simulations demonstrate the potential of this approach in improving both safety and navigation efficiency.

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Chapter 7

Rotordynamic Influence of Balance Pistons with Combined Axial and Radial Flow

Maximilian M.G. Kuhr, Zhengzhong Gao, and Peter F. Pelz

Abstract The dynamic behaviour of rotating machinery, such as compressors or turbo pumps, is largely influenced by the dynamic characteristics of narrow fluid-filled gaps, namely media- or oil-lubricated journal bearings, annular seals, thrust bearings and balance pistons. In addition to the induced forces and moments on the rotor due to the hydrodynamic effect, the specific machine elements are subjected to substantial pressure differences, resulting in considerable axial, radial, or combined axial and radial flow components, as seen for example in balance pistons. While the rotordynamic force characteristic of narrow gaps, such as conventional journal bearings or annular seals, and its rotordynamic impact is well-documented, the rotordynamic influence of machine elements inducing forces and moments - like long seals and bearings, balance pistons, or thrust balancers - is less understood. To better comprehend the dynamics of such narrow gaps, the revision of an existing test rig at the Chair of Fluid Systems at the Technische Universität Darmstadt is presented. During the revision, a comprehensive rotordynamic analysis was carried out using Finite Element Method (FEM) simulations. The objective was to identify safe operating conditions concerning the natural frequencies and stability of the rotor, and to assess the influence of the fluid-filled annuli on the overall rotordynamic characteristic of the test rig. In particular, the influence of different degrees of modelling of the rotordynamic influence of the fluid-filled gaps was investigated. Specifically, the influence of the added mass, as well as the additional stiffness and damping resulting from either the induced forces or moment due to the translational and angular degrees of freedom, was examined. It is shown that in contrast to the classical assumption of solely assuming induced forces, the additional tilt and moment coefficients highly influence the stability of the overall system as well as the specific natural frequencies of certain vibrations modes.

Keywords Rotordynamics · Narrow gaps · FEM

Introduction

In the context of contemporary turbomachinery design and development, there is a notable emphasis on the pursuit of enhanced operational capabilities, elevated rotational speeds, and the reduction of overall machinery weight. These developments are driven by the demand for more efficient and compact machines that can operate in a wider range of conditions. However, these innovations present specific technical challenges, particularly in relation to the emergence of complex vibrational behaviour, which is difficult to predict and control. These vibrations are even more dangerous when the rotor operates near critical speeds, where resonance conditions amplify oscillations. Given these risks, a comprehensive understanding of rotordynamic behaviour is essential. It plays a vital role not only in reducing vibration-induced noise but also in preventing sudden and catastrophic failures of the rotating components, which could result in costly downtime and repairs.

Addressing these challenges in the design phase is even more important, as it allows engineers to predict potential problems before the machinery is manufactured and put into operation. One of the principal factors influencing the rotordynamic behaviour of contemporary turbomachinery is the effect of induced forces and moments on narrow, fluid-filled annuli, such as those found in journal bearings, annular seals and balance pistons. In these components, motion-dependent forces and moments are induced on the rotor, either solely or as a combination of the hydrodynamic [1] or Lomakin effect [2], due to substantial pressure differences across the annulus. It is this pressure differential that gives rise to significant flow components observed in the pertinent machine elements, being either axial or radial, for instance in annular seals or thrust collars, or a combination of both, e.g. in balance pistons.

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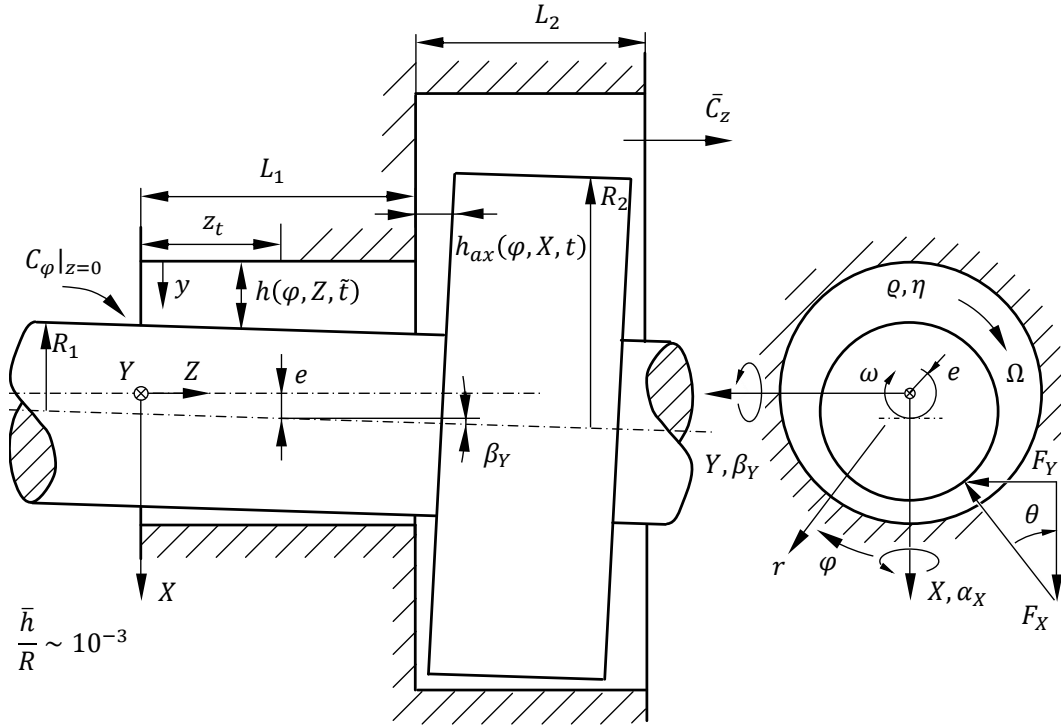


Fig. 1 Generic balance piston geometry.

While the force and moment characteristic of journal bearings or annular seals, with either a major axial or radial flow component is well-documented [3–8], the force and moment characteristic of machine elements with combined axial and radial flow such as balance pistons is largely unexplored due to the limited availability of reliable experimental data for validation purposes, cf. [9]. Furthermore, in examining the impact of annuli on overall machine dynamics, it is common practice to consider only the force characteristics due to lateral motions that are proportional to static and velocity-dependent movements of the rotor, i.e. the consideration of the annuli's stiffness and damping characteristics. However, when the complete equation of motion is considered, including the stiffnesses K , damping C and inertia M coefficients from the five degrees of freedom $X, Y, Z, \alpha_X, \beta_Y$,

$$\begin{aligned}
 - \begin{bmatrix} F_X \\ F_Y \\ F_Z \\ M_X \\ M_Y \end{bmatrix} &= \begin{bmatrix} K_{XX} & K_{XY} & K_{XZ} & K_{X\alpha} & K_{X\beta} \\ K_{YX} & K_{YY} & K_{YZ} & K_{Y\alpha} & K_{Y\beta} \\ K_{ZX} & K_{ZY} & K_{ZZ} & K_{Z\alpha} & K_{Z\beta} \\ K_{\alpha X} & K_{\alpha Y} & K_{\alpha Z} & K_{\alpha\alpha} & K_{\alpha\beta} \\ K_{\beta X} & K_{\beta Y} & K_{\beta Z} & K_{\beta\alpha} & K_{\beta\beta} \end{bmatrix} \begin{bmatrix} X \\ Y \\ Z \\ \alpha_X \\ \beta_Y \end{bmatrix} + \begin{bmatrix} C_{XX} & C_{XY} & C_{XZ} & C_{X\alpha} & C_{X\beta} \\ C_{YX} & C_{YY} & C_{YZ} & C_{Y\alpha} & C_{Y\beta} \\ C_{ZX} & C_{ZY} & C_{ZZ} & C_{Z\alpha} & C_{Z\beta} \\ C_{\alpha X} & C_{\alpha Y} & C_{\alpha Z} & C_{\alpha\alpha} & C_{\alpha\beta} \\ C_{\beta X} & C_{\beta Y} & C_{\beta Z} & C_{\beta\alpha} & C_{\beta\beta} \end{bmatrix} \begin{bmatrix} \dot{X} \\ \dot{Y} \\ \dot{Z} \\ \dot{\alpha}_X \\ \dot{\beta}_Y \end{bmatrix} + \\
 &+ \begin{bmatrix} M_{XX} & M_{XY} & M_{XZ} & M_{X\alpha} & M_{X\beta} \\ M_{YX} & M_{YY} & M_{YZ} & M_{Y\alpha} & M_{Y\beta} \\ M_{ZX} & M_{ZY} & M_{ZZ} & M_{Z\alpha} & M_{Z\beta} \\ M_{\alpha X} & M_{\alpha Y} & M_{\alpha Z} & M_{\alpha\alpha} & M_{\alpha\beta} \\ M_{\beta X} & M_{\beta Y} & M_{\beta Z} & M_{\beta\alpha} & M_{\beta\beta} \end{bmatrix} \begin{bmatrix} \ddot{X} \\ \ddot{Y} \\ \ddot{Z} \\ \ddot{\alpha}_X \\ \ddot{\beta}_Y \end{bmatrix}, \quad (1)
 \end{aligned}$$

it becomes evident that the overall impact of the annuli may be more significant than initially expected when modelled, cf. [10]. It can be reasonably deduced that the modelling of the rotordynamic influence of the fluid-filled gaps is significant, particularly around critical rotation speeds. Within equation 1 F_X, F_Y, F_Z and M_X, M_Y are the induced forces and moments of the annulus acting on the rotor. The translational motion, velocity and acceleration of the rotor is given by $X, \dot{X}, \ddot{X}, Y, \dot{Y}, \ddot{Y}, Z, \dot{Z}, \ddot{Z}$ whereas $\alpha_X, \dot{\alpha}_X, \ddot{\alpha}_X$ and $\beta_Y, \dot{\beta}_Y, \ddot{\beta}_Y$ denotes the angular motion, velocity and acceleration of the rotor around the X and Y axis. The dependence on the Z coordinate is only pertinent in the context of a balance piston, as considered here. In the case of purely axial or purely radial flow through the gap, the equation of motion reduces accordingly.

Equation 1 can be written in an alternative form with the definition of the individual sub-matrices based on the four different combinations: (I) induced forces on the rotor from translational motion, (II) induced forces on the rotor from angular motion, (III) induced moments on the rotor from translational motion and (IV) induced moments on the rotor from angular motion

$$-\begin{bmatrix} F_X \\ F_Y \\ F_Z \\ M_X \\ M_Y \end{bmatrix} = \begin{bmatrix} \mathbf{K}_I & \mathbf{K}_{II} \\ \mathbf{K}_{III} & \mathbf{K}_{IV} \end{bmatrix} \begin{bmatrix} X \\ Y \\ Z \\ \alpha_X \\ \beta_Y \end{bmatrix} + \begin{bmatrix} \mathbf{C}_I & \mathbf{C}_{II} \\ \mathbf{C}_{III} & \mathbf{C}_{IV} \end{bmatrix} \begin{bmatrix} \dot{X} \\ \dot{Y} \\ \dot{Z} \\ \dot{\alpha}_X \\ \dot{\beta}_Y \end{bmatrix} + \begin{bmatrix} \mathbf{M}_I & \mathbf{M}_{II} \\ \mathbf{M}_{III} & \mathbf{M}_{IV} \end{bmatrix} \begin{bmatrix} \ddot{X} \\ \ddot{Y} \\ \ddot{Z} \\ \ddot{\alpha}_X \\ \ddot{\beta}_Y \end{bmatrix}. \quad (2)$$

Typically, the rotordynamic coefficients depend on three different characteristics of the annulus, cf. figure 1: (i) the geometry of the annular gap, i.e. the gap lengths L_i , the shaft radii R_i , the mean gap clearances $\bar{h}_i$ and the gap functions $h = h(\varphi, Z, t)$ respectively $h_{ax} = h_{ax}(\varphi, X, t)$ with the circumferential and axial coordinate φ, Z and the radial coordinate X ; (ii) the operating conditions, i.e. the eccentric and angular position of the shaft e, α_X, β_Y , the distance of the centre of rotation z_T from the annulus entrance, the mean axial velocity $\bar{C}_z$, the angular velocity of the shaft Ω and the pre-swirl velocity at the annulus inlet $C_\varphi|_{z=0}$; (iii) the lubricant characteristics, i.e. the dynamic viscosity η and the fluid density ϱ .

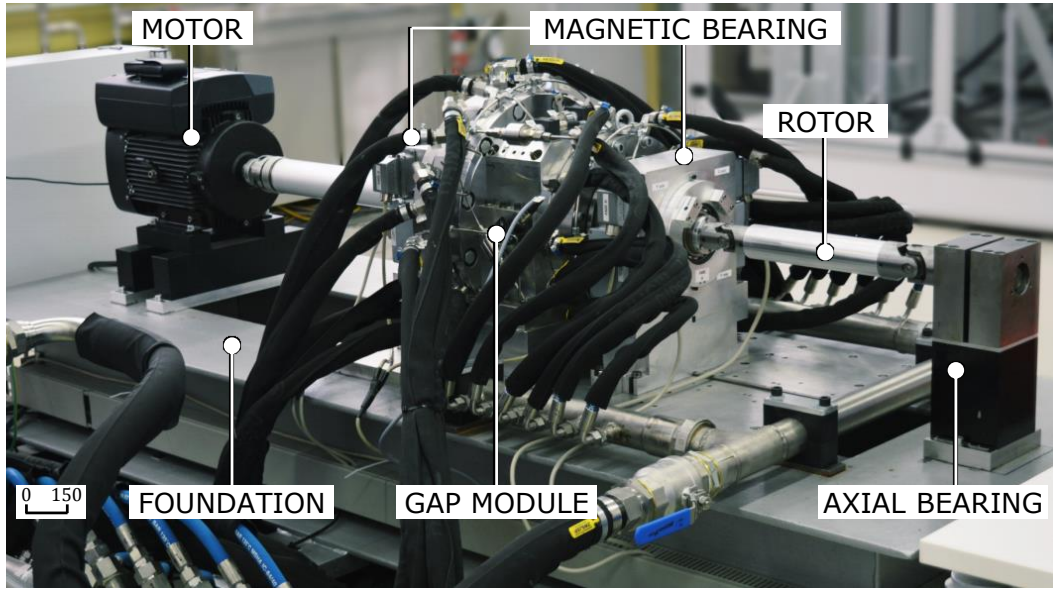
To address the issue of a lack of experimental validation data, a revised test rig [7] developed by the Chair of Fluid Systems at the Technische Universität Darmstadt for the determination of the static and dynamic characteristics of annular gaps with combined axial and radial flow components is described in the following section. Due to the construction of a new rotor, a rotordynamic analysis is conducted on the test rig employing the Finite Element Method with the use of ANSYS Mechanical. During the analysis, not only is the rotor investigated in terms of eigenfrequencies and stability margin, but also the degrees of modelling the influence of the balance piston is examined.

Revised Annular Seal and Bearing Test Rig

The upper part of figure 2 shows the formerly presented test rig by Kuhr et al. & Kuhr [7, 11]. As illustrated, the main components of the test rig are the foundation, the motor, the axial bearing, the gap module as well as the two active magnetic bearings (AMB) to support and excite the rotor. Compared to conventional bearings such as ball or journal bearings used by several authors, cf. Childs [3, 12], Kanemori & Iwatsubo [13, 14] and Moreland et al. [15], magnetic bearings have the advantage of bearing the rotor completely contactless and thus frictionless. Furthermore, the AMBs constitute an intrinsic force and displacement measuring system, with the capability to excite the rotor radially at defined amplitudes and frequencies. This renders them particularly suited to the identification of the dynamic characteristics of rotating machinery and components such as annular seals and journal bearings. The test rig is capable of measuring both concentric and off centred, i.e. eccentric, as well as misaligned rotor positions in an eccentricity and misalignment range of $\varepsilon := e/\bar{h} = 0..0.9$, $\alpha_x, \beta_y = 0.15^\circ$. By using a ten-stage centrifugal pump to supply the test rig, the axial pressure difference across the annulus Δp can be varied continuously in the range of $\Delta p = 0..20$ bar, whereas the pre-swirl at the seal entrance can be controlled in the range of $C_\varphi|_{z=0} = 0..1.7$. For further information on the capabilities, calibration and uncertainty of the test rig, please refer to the work of Kuhr [7] and Kuhr et al. [11].

The lower part of figure 2 illustrates the revised test rig for the identification of balance pistons, thrust collars and axial hydrodynamic bearings. In addition to larger and more powerful magnetic bearings, the axial bearing, the gap module and the rotor underwent significant modifications and redesign. Given the absence of an axial excitation of the rotor in previous designs, the axial bearing was created with the specific purpose of exciting the rotor in the axial direction. This is achieved through the use of a piezoelectric actuator with a possible displacement of 0.2 mm in combination with a movable axial bearing supported by linear guideways. The induced axial forces exerted by the balance piston are measured through the use of a piezoelectric force transducer with a measurement range of ± 14 kN and an absolute uncertainty of $\delta_F < 70$ N. At the same time, the induced moments of the annulus are supported by the magnetic bearings.

Figure 3 shows a technical drawing of the main part of the test rig, i.e. the gap module. The fluid path through the gap module is indicated by the black arrows. The module consists of 4 components: (i) the inlet to guide the flow and to measure the pre-swirl as well as the static pressure at the entrance of the annulus; (ii) the gap module; (iii) the outlet and (iv) the mechanical seals. The inlet is specifically designed to generate pre-swirled flows in front of the annulus. It is well known that the pressure field inside the annulus is influenced by the pre-swirl $C_\varphi|_{z=0}$. To investigate and quantify the influence on the static and dynamic characteristics, the process fluid, i.e. water, is tangentially fed into the inlet. By dividing the flow into two parts, the gap and the bypass volume flow, the circumferential velocity component in front of the annulus can be



REVISED GAP FLOW TEST RIG

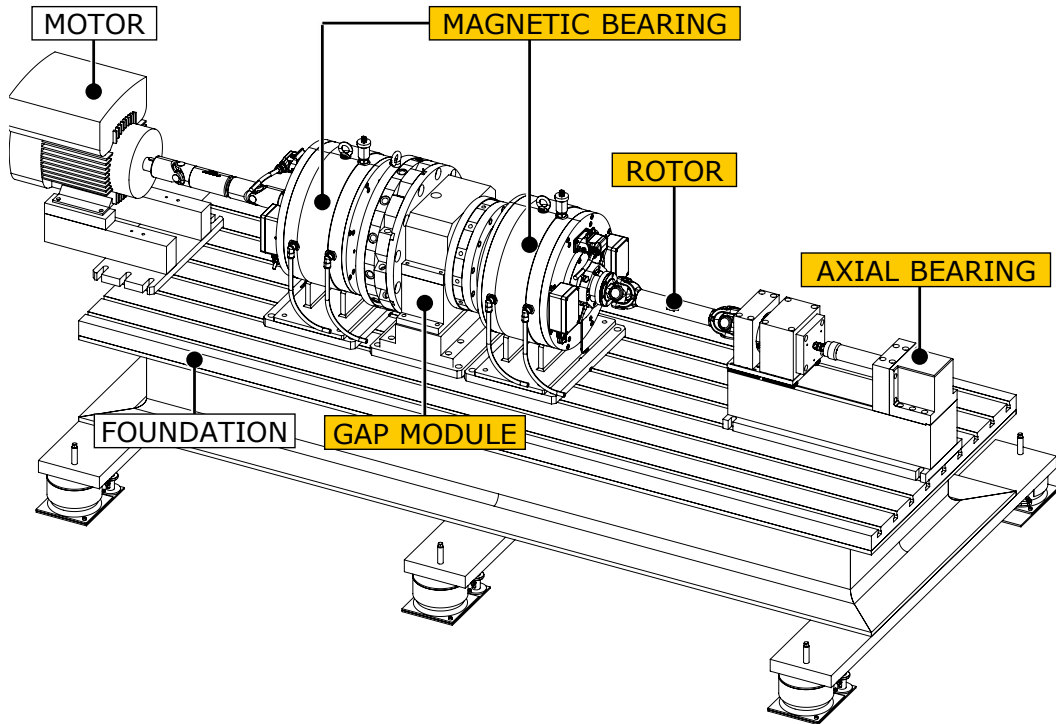


Fig. 2 Upper: Gap flow test rig at the Chair of Fluid Systems [7]; Lower: Revised test rig for the identification of annular gas with combined axial and radial flow component. The highlighted descriptions indicate major changed in the setup.

varied continuously. A Pitot tube connected to a pressure transmitter ($\delta_{\Delta p} < 0.01$ bar) is used to measure the circumferential velocity component. The test rig is capable of generating a pre-swirl in the range of $C_\varphi|_{z=0} = 0 \dots 1.7$. In order to ascertain the position of the rotor within the balance piston, the position of the shaft is measured at three distinct planes. (i) at the entrance, (ii) at the exit of the lubrication gap and (iii) at the midpoint of the thrust collar. To achieve this, two eddy current sensors are employed, each with a user-defined measuring range of 1 mm and an absolute uncertainty of $\delta_{X,GAP} < \pm 2.4 \mu\text{m}$.

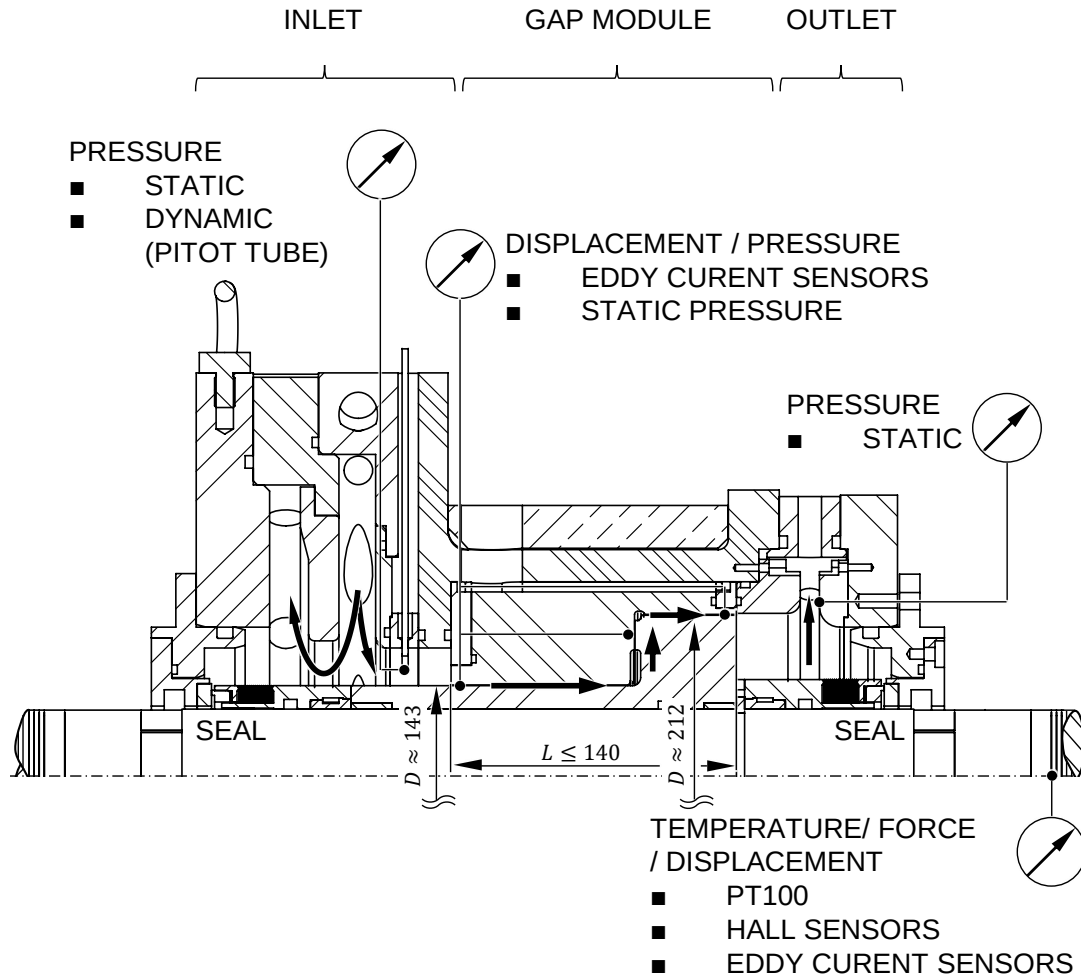


Fig. 3 Revised gap module of the annular seal and bearing test rig at the Chair of Fluid Systems. The black arrows indicate the flow path through the annulus inside the test rig.

These are arranged at an angle of 90° at the inlet and outlet of the piston. The axial position is then measured at four circumferentially distributed locations using identical eddy current sensors. Due to the modular design of the test rig, it is possible to vary the length of the piston as well as the clearance by changing the stator inlay and the rotor diameter within the gap module. The length can be varied in a range of $10 \text{ mm} \leq L \leq 140 \text{ mm}$, whereas the gap clearance varies between 0.1 mm and 0.7 mm . The supply pressure is measured at the inlet of the gap module by using a pressure transmitter connected to four wall pressure taps equally spaced around the annulus with an uncertainty of $\delta_{p,z=0} < \pm 0.033 \text{ bar}$. The static pressure before and after the thrust collar is determined using pressure transmitters connected to a single wall pressure tap, each, whereas the pressure difference across the annulus is measured by a differential pressure sensor with an absolute uncertainty of $\delta_{\Delta p} < \pm 0.04 \text{ bar}$. The test rig is designed to investigate pressure differences of up to 20 bar . The volume flow through the gap is measured using an electromagnetic flowmeter with an absolute uncertainty of $\delta_{\dot{Q},\text{GAP}} < \pm 0.04 Q_{\text{GAP}}$. To avoid cavitation within the annulus, the test rig can be pressurised up to 15 bar . The test rig is operated using water at a constant temperature of 35°C and is fed by a 10 staged 55 kW centrifugal pump.

Finally, figure 4 presents an isometric sketch of the revised rotor. The rotor is constructed from steel type 1.1730 and stainless-steel type 1.4305, comprising two cardanic shafts with preloaded needle bearings and the main rotor with the annular gap specimen and the rotating part of the mechanical face seals. The cardanic shafts have been selected on the basis of a carefully considered design. One shaft incorporates a slider mechanism, which serves to compensate for the axial excitations. In contrast, the other shaft is devoid of this feature. This design choice is crucial, as the induced axial forces are transmitted through the cardanic shaft without a slider into the aforementioned axial bearing, which is connected to the force transducer and the piezoelectric actuator. Additionally to the components of the rotor, figure 4 indicates the modelling of

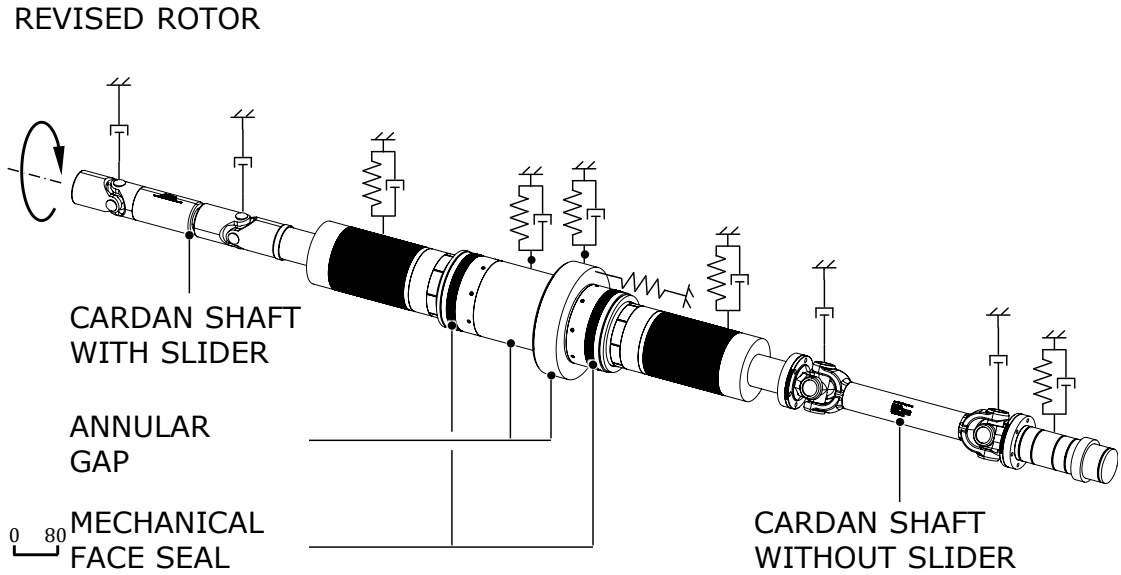


Fig. 4 Revised rotor of the annular seal and bearing test rig at the Chair of Fluid Systems. The sketched mechanical elements indicate the position of additional modelled stiffness, damping, and inertia coefficients used in the calculation.

the cardanic shaft joints, magnetic bearing, ball bearing and balance piston for the rotordynamic analysis. Here, the cardanic shaft joints are represented by assuming solely damping characteristics, whereas the magnetic bearings and ball bearings in the revised axial bearing are modelled using stiffness and damping coefficients. The annular gap of the balance piston is modelled by the aforementioned rotordynamic stiffness, damping and inertia coefficients, cf. equation 1. The material damping is chosen accordingly to the Rayleigh damping to be proportional to the mass and stiffness matrix, cf. Markert [16]

Results and Discussion

Numerical Setup

The rotordynamic analysis, in terms of predicting natural frequencies and stability margin, is carried out using a Finite Element Method (FEM) calculation using ANSYS Mechanical 2022 R2. Adjacent components with no relative motion are meshed and very fine structures with limited influence on the overall behaviour such as screws, holes and chamfers are neglected. The rotor was discretised using SOLID186 and SOLID187 elements, with element sizes ranging from 3 to 6 mm. Most of the components, such as those with cylindrical shapes, are meshed with hexagonal SOLID186 elements, while the components with more complicated geometries, such as the crossed cylinders in the universal joint, are meshed with SOLID187 elements, a high-order 3D element. The FEM simulation consists of approximately 550,000 nodes and 130,000 elements in total. COMBIN40 elements are used to model the deep groove ball bearings and the magnetic bearings. The COMBIN40 element is a combination of a spring and a damper in parallel and connects two nodes, one of which can be fixed to the ground. In the simulation, the outer ring of the bearing is assumed to be fixed to the ground, while the inner ring of the bearing is meshed with several nodes. These nodes are connected to one of the nodes of the COMBIN40 element using an MPC184 element, which indicates the multipoint constraint and models the kinematic constraints between the nodes. The MPC184 element is also used to model the coupling and translational joint. The inertia part of the dynamic coefficients is modelled by MASS21 elements, which are concentrated mass component elements with a single node. However, the MASS21 element cannot model the cross-coupled terms in the inertia matrices. As a result, only the diagonal terms of the inertia matrices are considered in the following calculation. The dynamic behaviour of the annular gap, i.e. the balance piston, is modelled using MATRIX27 elements. The values are obtained by an in-house calculation tool, the Clearance-Averaged Pressure Model for gaps with purely axial flow, see [6]. The influence of the thrust collar is obtained by CFD analysis. Only the direct stiffness K_{ZZ} is considered. Furthermore, the MATRIX27 element is only capable of modelling skew-symmetric inertia matrices. Therefore, the inertia matrices are only considered with the dominant diagonal terms. The complete equation of motion is reduced accordingly

$$\begin{aligned}
-\begin{bmatrix} F_X \\ F_Y \\ F_Z \\ M_X \\ M_Y \end{bmatrix} &= \begin{bmatrix} K_{XX} & K_{XY} & 0 & K_{X\alpha} & K_{X\beta} \\ K_{YX} & K_{YY} & 0 & K_{Y\alpha} & K_{Y\beta} \\ 0 & 0 & K_{ZZ} & 0 & 0 \\ K_{\alpha X} & K_{\alpha Y} & 0 & K_{\alpha\alpha} & K_{\alpha\beta} \\ K_{\beta X} & K_{\beta Y} & 0 & K_{\beta\alpha} & K_{\beta\beta} \end{bmatrix} \begin{bmatrix} X \\ Y \\ Z \\ \alpha_X \\ \beta_Y \end{bmatrix} + \begin{bmatrix} C_{XX} & C_{XY} & 0 & C_{X\alpha} & C_{X\beta} \\ C_{YX} & C_{YY} & 0 & C_{Y\alpha} & C_{Y\beta} \\ 0 & 0 & 0 & 0 & 0 \\ C_{\alpha X} & C_{\alpha Y} & 0 & C_{\alpha\alpha} & C_{\alpha\beta} \\ C_{\beta X} & C_{\beta Y} & 0 & C_{\beta\alpha} & C_{\beta\beta} \end{bmatrix} \begin{bmatrix} \dot{X} \\ \dot{Y} \\ \dot{Z} \\ \dot{\alpha}_X \\ \dot{\beta}_Y \end{bmatrix} \\
&\quad + \begin{bmatrix} M_{XX} & & & & \\ & M_{YY} & & & \\ & & 0 & & \\ & & & M_{\alpha\alpha} & \\ & & & & M_{\beta\beta} \end{bmatrix} \begin{bmatrix} \ddot{X} \\ \ddot{Y} \\ \ddot{Z} \\ \ddot{\alpha}_X \\ \ddot{\beta}_Y \end{bmatrix}, \quad (3)
\end{aligned}$$

$$\begin{aligned}
-\begin{bmatrix} F_X \\ F_Y \\ F_Z \\ M_X \\ M_Y \end{bmatrix} &= \begin{bmatrix} K_I & 0 & K_{II} \\ 0 & K_{ZZ} & 0 \\ K_{III} & 0 & K_{IV} \end{bmatrix} \begin{bmatrix} X \\ Y \\ Z \\ \alpha_X \\ \beta_Y \end{bmatrix} + \begin{bmatrix} C_I & 0 & C_{II} \\ 0 & 0 & 0 \\ C_{III} & 0 & C_{IV} \end{bmatrix} \begin{bmatrix} \dot{X} \\ \dot{Y} \\ \dot{Z} \\ \dot{\alpha}_X \\ \dot{\beta}_Y \end{bmatrix} + \begin{bmatrix} M_I & & \\ & 0 & \\ & & M_{IV} \end{bmatrix} \begin{bmatrix} \ddot{X} \\ \ddot{Y} \\ \ddot{Z} \\ \ddot{\alpha}_X \\ \ddot{\beta}_Y \end{bmatrix}. \quad (4)
\end{aligned}$$

Campbell Diagram

Influence of Different Levels of Modelling the Dynamics of the Annulus

The eigenfrequency analysis is carried out in the operating range of the test rig. Here, the rotor speed can be varied from a minimum of $f = 6.3$ to a maximum of 50 Hz, whereas the pressure difference across the balance piston is assumed to be constant at a value of $\Delta p = 10$ bar. The balance piston itself consists of three separate gaps, (i) a first annular gap with a relative length of $L_1/R_1 = 1.26$ and a relative clearance of $\bar{h}_1/R_1 = 1.7\%$, (ii) the thrust collar with an assumed gap height of $\bar{h}_{ax} = 0.1$ mm and (iii) a second annular gap with a relative length of $L_1/R_1 = 0.47$ and a relative clearance of $\bar{h}_1/R_1 = 2.6\%$. This results in circumferential Reynolds numbers of the annuli of $Re_{\varphi,1} := (\Omega R_1 \bar{h}_1)/\nu \approx 750..5889$ and $Re_{\varphi,2} := (\Omega R_2 \bar{h}_2)/\nu \approx 1800..12000$, indicating both laminar and turbulent flow conditions over the rotor speed range. Six different degrees of modelling the influence of the balance piston is investigated:

- Only the rotor system and its foundations are modelled, i.e. the rotor is operated without the balance piston.
- The rotor system and the stiffness and damping coefficients due to translational motion, i.e. K_I and C_I , are considered.
- The rotor system as well as the stiffness, damping and inertia coefficients due to translational motion, i.e. K_I , C_I and M_I , are considered.
- The rotor system as well as the stiffness, damping and inertia coefficients of the forces due to translational and angular motion, i.e. $K_{I..II}$, $C_{I..II}$ and $M_{I..IV}$, are considered.
- The rotor system and the stiffness, damping and inertia coefficients of the forces and moments due to translational and angular motion, i.e. $K_{I..IV}$, $C_{I..IV}$ and $M_{I..IV}$, are considered.
- The rotor system and the full reduced matrix, equation 3, i.e. the stiffness, damping and inertia coefficients of the forces and moments due to translational and angular motion, as well as the direct axial stiffness K_{ZZ} , are taken into account.

Figure 5 depicts the Campbell diagrams of the six different degrees of modelling the influence of the balance piston, which illustrate the natural eigenfrequencies of the rotor system of the rotor speed for varying degrees of modelling. The black lines represent the rotor speed and twice the rotor speed, while the coloured lines correspond to the different eigenmodes of the rotor system. An intersection between the black and coloured lines indicates a critical speed of the system. In the figures, the first three bending modes, as well as the first torsional and first axial mode, are displayed. The solid lines represent a forward whirling motion of the rotor, whereas the dashed lines indicate a backward whirl. Distinction between forward and backward whirl is evident only for the different bending modes. Additionally, the critical speeds of the system are given in the figures. Here, the value is an approximate evaluation based on the forward whirl motion of the rotor.

Focussing first on the rotor without the liquid-filled balance piston (figure 5a), it becomes evident, that the most critical eigenmode is the second bending mode (blue line) as well as the first torsional mode (green) of the system. Both modes occur initially above twice the rotational frequency of the rotor. At a critical frequency of approximately 22 Hz and 44 Hz, the second bending mode coincides with twice the speed of the rotor as well as its actual speed, making it a critical point. The first axial mode as well as the third bending mode always lies above twice the speed of the rotor.

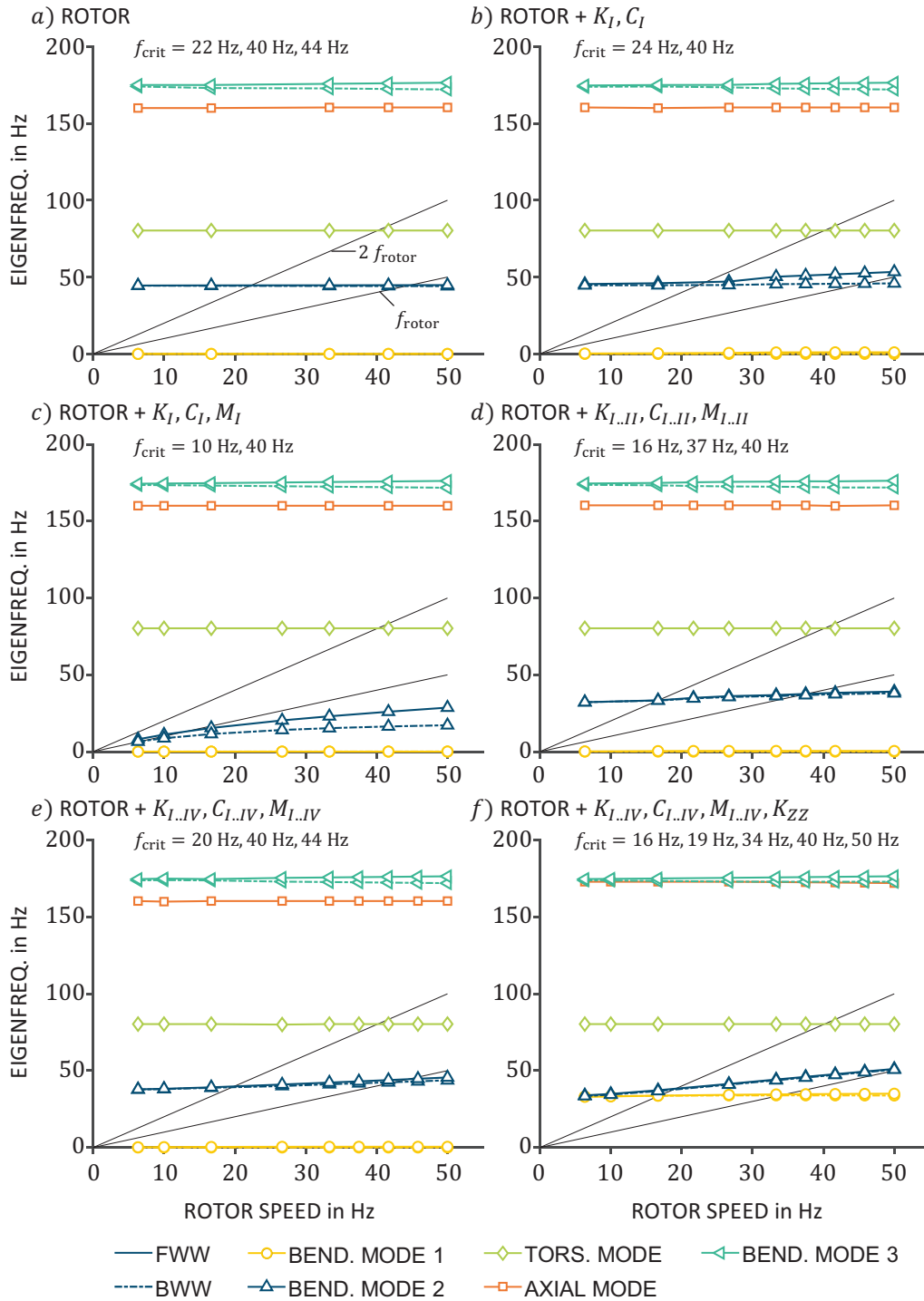


Fig. 5 Campbell diagrams of the rotor system. The different figures a) to f) indicate different degrees of modelling of the rotordynamic influence of the fluid-filled gaps, ranging from a) the rotor without the annulus, to f) the rotor with a balance piston inducing both forces and moments as well an additional axial stiffness onto the rotor

Focussing now on the impact of the annular gap and later the overall balance piston. Figure 5b depicts the Campbell diagram when the rotor system and the stiffness and damping coefficients due to translational motion, i.e. K_I and C_I , are considered. This level of modelling is typically employed for classical journal bearings with laminar flow conditions, where the influence of flow inertia is negligible. It is shown that there is only a slight change in the eigenfrequencies of the overall

rotor - gap system. The first bending mode is almost unaltered, whereas the eigenfrequencies of the second bending mode experience a slight increase towards higher rotor speeds. Due to this increase, the forwards whirling motion of the second bending mode is no longer a critical frequency at a spinning speed of 44 Hz. Furthermore, the frequency difference between the forward and backward whirl motion is more pronounced at rotor speed exceeding 25 Hz. The introduction of the annulus has no impact on the first torsional mode as well as on the first axial mode and the third bending mode. By introducing the annular gaps, the rotor balance piston system exhibits three critical speeds over the operation range of the test rig.

Focussing now on the impact of the additional inertia coefficients M_I of the balance piston, figure 5c a strong decrease of the eigenfrequencies of the second bending mode. This is reasonable since adding additional mass terms, like the added mass induced by the annulus, lowers the eigenfrequencies of the oscillating system. This is of peculiar interest, as the reduction in frequency is proportional to the increased mass. It is important to recognise that the masses induced by the annular gap, especially with an eccentric rotor, can be an order of magnitude higher than the mass of the rotor itself. In the case shown here, the added mass of the balance piston is approximately equal to the mass of the solid rotor. Nevertheless, it is essential to consider the added mass when assessing the natural frequencies of the overall system.

The aforementioned trend of nearly unrecognisable changes in the Campbell diagram is also apparent when considering not only the stiffness, damping and inertia resulting from forces due to translational movements but also the coefficients of forces resulting from angular motions, i.e. K_{II} , C_{II} and M_{II} , cf. figure 5d. Nevertheless, a more detailed examination reveals a notable discrepancy. On the one hand, the system now exhibits three critical frequencies when the coefficients K_{II} , C_{II} and M_{II} are considered. This was first the case when only considering the rotor, cf. figure 5a. On the other hand, the resonance frequency of the second bending mode (blue) exhibits a slight shift towards lower frequencies. This phenomenon can be attributed to the fact that the second bending mode represents an S-shaped movement, i.e. a tumbling movement within the annular gap. It is therefore to be expected that this motion will be influenced by the angular stiffness, damping and inertia of the gap. Regarding the first torsional and axial mode as well as the third bending mode, no significant changes can be seen.

Finally, figure 5e and 5f show the impact of the induced moments of the annulus on the eigenfrequencies of the systems. Similar to the figures above, the visible changes in figure 5e are small. However, when taking all rotordynamic coefficients into account, the first and second bending mode are altered as well as the axial mode. Furthermore, the number of critical speeds of the system increased to five critical speeds. However, the underlying reasons for the significant alterations observed in the initial bending mode remain to be fully elucidated.

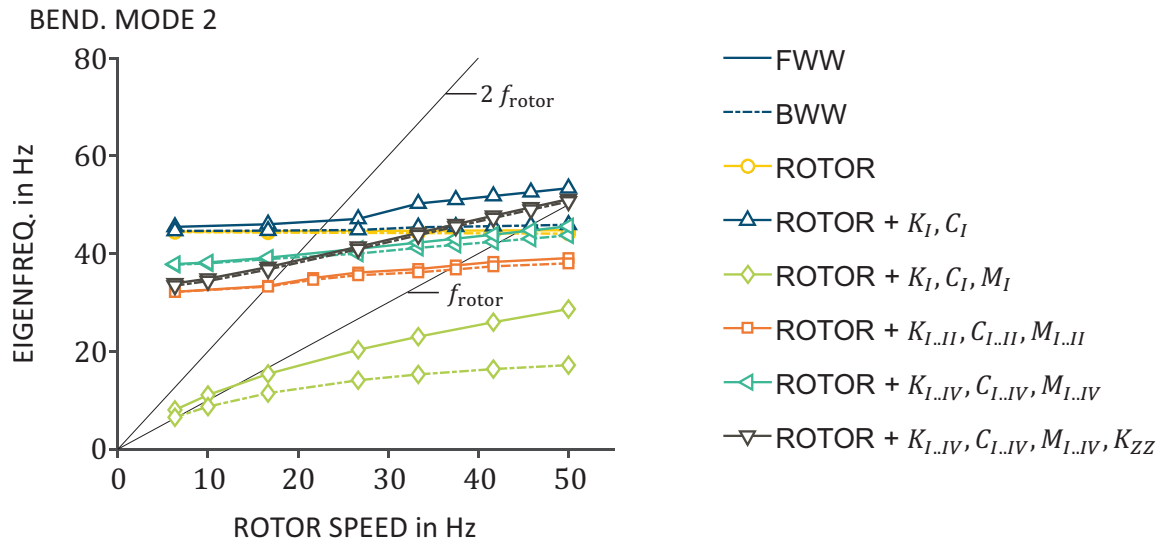


Fig. 6 Detailed investigation of the first and second bending mode regarding the influence of the different degrees of modelling of the rotordynamic influence of the balance piston.

In order to emphasise the impact of the different degrees of modelling the rotordynamic impact of the balance piston on the test rig rotor, the second bending mode is displayed in detail in figure 6. In contrast to figure 5, the different coloured lines represent the different degrees of modelling the impact of the annulus. The second bending mode of the rotor exerts a significant impact on the selection of stiffnesses, damping, and inertia coefficients. As the level of detail in the dynamic characteristic of the balance piston increases, the resonance frequency of the second bending mode increases at first, but

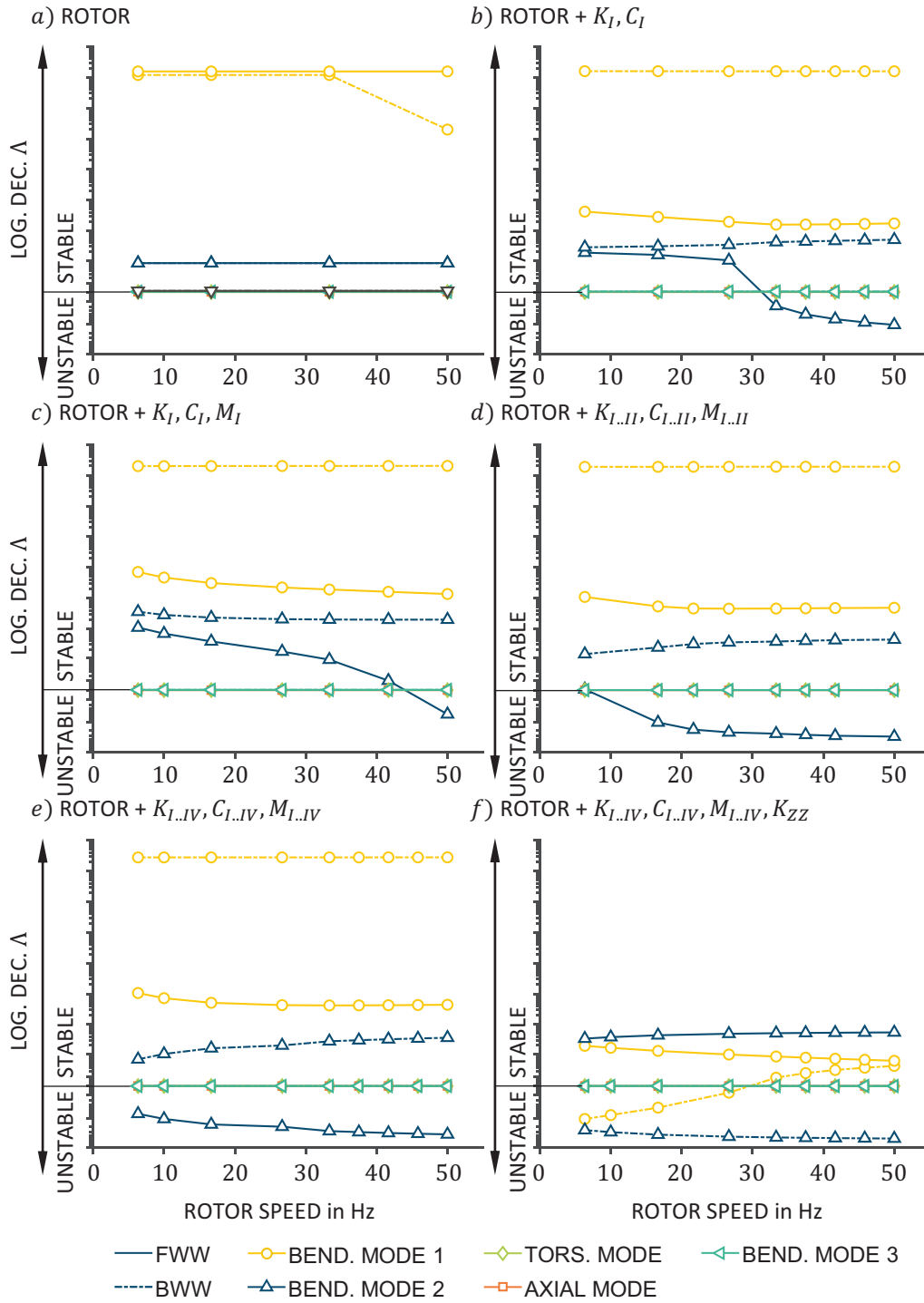


Fig. 7 Stability diagram of the rotor system. The different figures a) to f) indicate different degrees of modelling of the rotordynamic influence of the fluid-filled gaps, ranging from a) the rotor without the annulus, to f) the rotor with a balance piston inducing both forces and moments as well an additional axial stiffness onto the rotor.

decreases significantly when adding the added mass to the calculations. Subsequently, the eigenfrequencies increase with the detail in modelling. Interestingly, when taking all rotordynamic influences into account, the second bending mode exhibits a characteristic similar to the system when only taking the stiffness and damping coefficients due to the translational motion into account.

Stability Analysis

To provide a comprehensive analysis, the stability of the system is evaluated using the logarithmic decrement. A positive logarithmic decrement indicates a stable eigenvalue and, consequently, a stable system. Conversely, a negative logarithmic decrement signifies an unstable eigenvalue, which in turn denotes an unstable system. The logarithmic decrement can be interpreted physically by considering the time course of a decaying oscillation in a weakly damped system. The ratio of two consecutive oscillation amplitudes, $q(t)$ and $q(t + \Delta t)$, which are offset by the time Δt , is

$$\frac{q(t)}{q(t + \Delta t)} = \exp(\Lambda). \quad (5)$$

The logarithmic decrement Λ is defined as the exponent of the ratio of two successive oscillations. If the exponent is negative, the amplitudes will increase over time. Conversely, if the exponent is positive, then the oscillations decay over time. Therefore, a negative logarithmic decrement is indicative of unstable system behaviour, whereas a positive logarithmic decrement signifies a stable state of the system. It should be noted that the logarithmic decrement is only defined for weakly damped systems.

Figure 7 depicts the stability diagrams of the six different degrees of modelling the influence of the balance piston. In contrast to the Campbell diagram, the black line represents the stability threshold at which the system modes become unstable. As with the analysis of the critical frequencies, the stability diagrams demonstrate a significant impact of the degree of modelling on the dynamic characteristics of the balance piston. The introduction of the annulus results in a modification of the overall rotor system, cf. figure 7a. The forward whirl motion of the second bending mode becomes unstable, whereas the first bending mode is stable over the pictured rotor speeds. Interestingly, the second bending mode is also the one influenced the most when increasing the degree of modelling the rotordynamic influence of the balance piston. It is also worth noticing that the characteristic changes when taking the axial stiffness of the balance piston K_{ZZ} into account, cf. figure 7f. Here, the forward whirl motions of the second bending mode becomes stable.

Interestingly, and in contrast to the Campbell diagrams in the previous chapter, clear differences in the stability of the system can be observed as the level of detail of the modelling increases. For example, figure 7d shows that adding the stiffness, damping and inertia coefficients of the forces induced by the angular motion, K_{II} , C_{II} and M_{II} , destabilises the forward whirl motion of the second bending mode, even at small rotor frequencies. This becomes even more pronounced when taking the induced moments due to the translational and angular motion into account, cf. figure 7e. However, at the finest level of modelling, the system seems to be stable for all the investigated forward whirl motion. The results of the stability analysis reinforce the conclusion of the Campbell diagrams that the extent to which annular gap flows are modelled, namely the degree to which coefficients representing forces and moments due to translational and angular motion are included, has a considerable impact on the overall dynamic behaviour of the system, particularly regarding the stability margin of critical frequencies.

Summary & Conclusion

The performance of rotating machinery, such as compressors or turbo pumps, is significantly affected by the dynamic characteristics of narrow fluid-filled gaps. These include media- or oil-lubricated journal bearings, annular seals, thrust bearings and balance pistons. In particular, the dynamic characteristic of annuli with combined axial and radial flow components such as balance pistons is not yet fully understood. Therefore, a revised annular seal and bearing test rig is resented. The revised rotor is subjected to rotordynamic analysis, with a particular focus on eigenfrequencies and system stability. Furthermore, the impact of the level of modelling the rotordynamic characteristic of balance pistons is examined. In particular, we are examining the impact of added mass, as well as the effects of additional stiffness and damping resulting from induced forces or moments due to translational and angular degrees of freedom. The introduction of stiffness, damping, and inertia coefficients related to both translational and angular motions has a significant impact on the system's natural frequencies and critical speeds, particularly affecting mainly the second bending mode. As the level of detail in the modelling increases, the resonance frequencies of this mode shift, notably exhibiting both an increase as well a decrease in the eigenfrequencies. This highlights the necessity of considering not only the translational effects, but also the angular forces and moments induced by the balance piston and annular gaps in rotordynamic analyses. The findings indicate that conventional approaches that neglect these additional degrees of freedom could result in underestimations of critical speeds and unexpected large vibration amplitudes, particularly in turbomachinery systems operating near critical speeds.

CRedit Authorship Contribution Statement

Maximilian M.G. Kuhr: Conceptualization, Methodology, Analysis, Validation, Investigation, Writing, Visualization, Supervision, Project Administration.
 Zhengzhong Gao: Software, Analysis, Validation, Investigation.
 Peter F. Pelz: Funding.

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Chapter 8

Investigating the Propulsive Efficiency of Bio-Inspired Fish-like Elastic Caudal Fin through Dynamic Analysis and Experimental Validation

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Abstract The present work focuses on the dynamic analysis of bio-inspired fish-like robotic fins in order to evaluate their propulsive efficiency. This study relates the effect of the main factors on which this efficiency depends, particularly materials and stiffness in a typical biomimetic range. Numerical and experimental modal analysis techniques are employed to examine the vibrational properties of different fins. This analysis is essential to characterize the elastic materials and understand how the respective stiffness levels affect the dynamic behavior of the fin. A numerical analysis is then conducted using an analytical model that simulates the coupled dynamics between the fluid and the elastic fin. This model takes into account the two-ways interaction between the hydrodynamic forces and elastic deformations of the fin, allowing a detailed evaluation of propulsive efficiency parameters. Key performance indicators, such as elastic deformation, thrust, and efficiency, are calculated and analyzed in relation to changes in materials and stiffness. Hydrodynamic tank experiments are ongoing to directly measure the performance of the robotic fins, comparing the experimental data with the predictions of the analytical model. This validation phase is critical to confirm the accuracy of the model and to further refine the predictions regarding propulsive efficiency. The results of this paper provide valuable insights for the future design of underwater robots, with potential applications in areas such as marine exploration, environmental monitoring, and bioinspired robotics.

Keywords Fish-like · Bionspired robot · FSI model · Modal analysis · Biomimetic propulsion

Introduction

The use of biomimetic thrusters in the marine environment offers significant advantages for marine transportation, particularly through reduced noise compared to conventional thrusters and the ability to provide a wide range of propulsive efficiency. Although the applicability of such systems is currently limited to small marine vehicles, such as AUVs (Autonomous Underwater Vehicles) or ROVs (Remotely Operated Vehicles), interest in their performance is growing. Numerous studies in the literature point out that these thrusters are often designed on a small scale, with dimensions on the order of a centimeter, mainly for educational purposes or laboratory testing [1, 2]. The objective of this study is to overcome these limitations by proposing a systematic analysis on elastic propulsion system intended for larger robotic prototypes, with dimensions of the order of a meter. This allows them to cope with more complex operating conditions, such as higher cruising speeds and greater operational depths, than the micro-robots currently in use. Specifically, the paper presents an analysis of the propulsive behavior of different types of elastic thrusters designed to move an AUV with fish-inspired morphology. The analysis is based both on numerical results and on a preliminary experimental campaign, with the aim of identifying the most suitable materials and the relative frequencies peculiar to the systems under consideration.

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Mathematical Evaluation of Bio-Propulsion Efficiency

The proposed two-way model represents the cross section of the oscillatory rectangular caudal fin. This involves coupling the one-dimensional Euler-Bernoulli plate equation with potential flow theory. The hydroelastic treatment is developed entirely in the time domain, to be interfaced with control logics [3, 4], and from this has been derived an integro-differential final equation of motion with the presence of convolutional memory effects terms [5]. The model is developed and validated in [6, 7], where all coupling terms (mass, stiffness, added damping and memory effects) are developed in closed form through complex analysis, by making the model semi-analytical [8]. The coupled equation of motion is here formulated in nondimensional form, choosing the half-length $L/2$ and the flapping period $1/f_\alpha$ as the characteristic length and time scales, respectively so, the equation of motion for the fin displacement w is

$$2R \frac{\partial^2 w}{\partial t^2} + S_d \frac{8\pi^2}{3\sigma^2} \frac{\partial^2 w}{\partial x^2} = \tilde{F}_f(x, t) \quad (1)$$

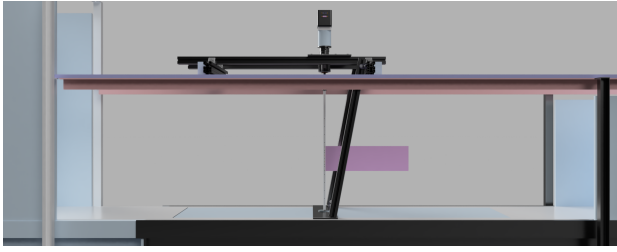
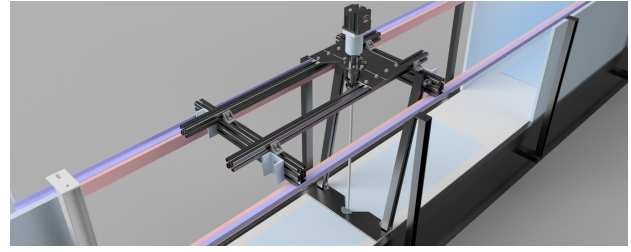
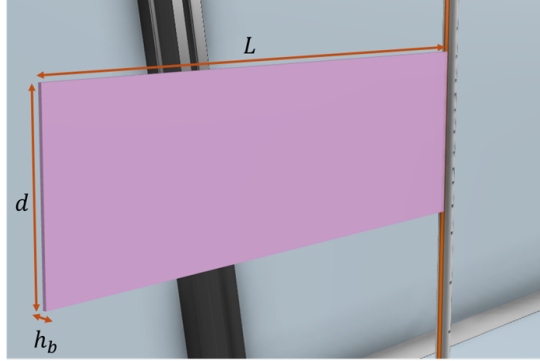
where the dimensionless hydrodynamic load is given by $\tilde{F}_f(x, t) = \frac{4F_f(x, t)}{\rho d L^2 f_\alpha^2}$ with ρ the fluid density and d the fin width, the reduced frequency $\sigma = \frac{\pi L f_\alpha}{U}$ with U the inflow speed, the stiffness $S_d = \frac{E h_b^3}{\rho U^2 L^3}$ with E the Young modulus and h_b the fin thickness and the inertia ratio $R = \frac{\rho_b h_b}{\rho L}$ with ρ_b the material density. Using the modal approach $w(x, t) = \sum_{i=1}^N \phi_i(x) q_i(t)$ where N is the modes number and $\phi_i(x)$ is the fixed-free beam eigenfunction, equation (1) can be decoupled and solved by imposing the initial conditions $q_i(t=0) = q_{i0}$ (see [6] for further details). Once the solution $w(x, t)$ is found from equations (1), one can compute the propulsive parameters fundamental to evaluate the efficiency of the system, such as: the thrust force $T(t) = -d \int_{-L/2}^{L/2} \Delta P(x, t) [\alpha(t) + w'(x, t)] dx$, the input power expended $P(t) = d \int_{-L/2}^{L/2} \Delta P(x, t) [\dot{h}(t) + (L/2 + x)\dot{\alpha}(t) + \dot{w}(x, t)] dx$ and so the Froude efficiency $\eta = \frac{\overline{TU}}{\overline{P}}$, where the overline represents the mean value. Another important parameter is the peak-to-peak amplitude of the flexible panels' trailing edge $\chi = 2 \max \left[h(t) + L\alpha(t) + \frac{w(L/2, t)}{1 - \alpha(t)^2/2} \right]$, which is an output of the system that depends on the balance between inertial, elastic and external fluid forces. Through this, it is then possible to calculate the Strouhal number, $St = f\chi/U$, used to evaluate the optimality of the biomimetic propulsion. The global optimal efficiency for the two-dimensional elastic plate then occurs when two conditions are simultaneously satisfied: (i) the oscillation frequency of the panel produces an amplitude of motion that yields a Strouhal number in the optimal range $0.25 < St < 0.35$ and (ii) this frequency of motion is tuned to the structural resonant frequency of the panel. The propulsive thrust parameter in dimensionless form became $C_T = \frac{\overline{T}}{\frac{1}{2}\rho U^2 d L}$ and will be reported in the design maps as function of the inertia and stiffness dimensionless parameters, S_d and R for a specific operating condition (i.e. U, f_α, A_α).

Experimental Setup

An experimental set-up has been designed to measure the propulsive force exerted by the fins, (Figures 1-2). It features a closed-loop circulation channel with a width of 536 mm and a water free surface height of 650 mm. A support structure is installed to house the actuators, transmission, and propeller, isolating the system from channel vibrations that could interfere with the experiment and data extraction. The pumps control the free stream velocity up to 1 m/s. The actuation system consists of a brushless SmartMotor (1.45 Nm continuous torque, 3.39 Nm peak, encoder resolution 8.000 counts/rev, support for DeviceNet, PROFIBUS, and CANopen communication protocols) connected to a planetary gearbox and a 6-axis ME load cell for measuring propeller performance (50 N, 5 Nm). An elastic joint minimizes misalignment effects between the gearbox and load cell. The drive shaft connects to the channel's bottom via a flanged bushing, with fins secured by hexagon socket set screws to prevent damage and ensure proper placement.

Characterization of Propulsion System's Material Properties

After preliminary numerical simulations, the optimal geometric configuration for the propulsion of a fish-like AUV with a length of about 1.5 m, designed to operate in a cruising speed range from 0.1 to 0.9 m/s, is selected. The propulsion system, represented by the fin, has a rectangular geometry, as shown in (Figure 3 - a), with dimensions equal to $L = 510$ mm, $d = 150$ mm, $h_b = 1, 2, 3$ mm. The materials chosen for the fin include Aluminium (AL), for the metallic configuration, and plastic materials as HDPE and PMMA, selected to cover a range of stiffness in line with that of biomimetic systems, i.e. $R \ll 1$ and $10^{-3} < S_d < 10^3$. Figure 3 - b) shows an example of real fin (AL 3 mm).

**Fig. 1** Experimental setup in hydraulic channel**Fig. 2** Rotational handling system

a)



b)

Fig. 3 Rectangular fin configurations a) rendering and dimension b) real fin (AL 3 mm)

To identify the mechanical properties of fin materials, a preliminary density measurement was made by weighing each fin, the results of which are shown in Table 1. Next, an experimental analysis was conducted to determine the first natural frequency under dry conditions for each fin with a thickness of 3 mm in free-free condition. The obtained experimental frequencies were compared with those computed by modal analysis conducted by the finite element method (FEM) using Ansys software. The results showed excellent agreement between the experimental and simulated frequencies, with errors of less than 2 % (Table 2).

Table 1 Material mechanical properties of 3 mm fins

Material	$\rho_s [kg/m^3]$	ν	$E [GPa]$
AL	2633.62	0.33	65.89
PMMA	1191.60	0.42	2.35
HDPE	933.42	0.42	1.96

Knowing all the geometric characteristics, it was possible to calculate the Young's modulus for each material analyzed, using the appropriate relationship $E = \frac{\rho_b A_b}{I} \left(\frac{2f_1 \pi L^2}{\lambda_1^2} \right)^2$, with the inertia modulus $I = \frac{dh_b^3}{12}$ and λ_1 the first free-free beam wavenumber. The results are shown in Table 1 where the standard value for the Poisson's coefficient ν was considered.

Experimental Identification of Dry and WET Fin's Frequency

The experimental setup involves clamping a plate inside a rod. The rod mimics an elastic support rather than a perfect clamp, decreasing the constraint stiffness and fin's natural frequencies and so necessitating a realistic modeling approach. To accurately determine the elasticity of real constraint, an experimental modal analysis was conducted to identify natural frequencies of the fin in air, using an accelerometer to measure the response of the structure when it is struck at specific grid points. These results are shown in Table 3 first column. Then, the fins were modeled with Ansys software, applying a 1-cm elastic support to the clamp side, with a shell element discretization of 5 mm. The tuning of the modeled stiffness

Table 2 First natural frequency in free-free condition of 3 mm fins: Experimental vs FEM

Material	AL	PMMA	HDPE
Experimental results [Hz]	59.3	22.2	17.2
Numerical results [Hz]	60.101	22.36	17.2
Percentage error [%]	-1.344	-0.716	0.397

in the FEM was then done until good agreement was obtained in the comparison between the numerical FEM analysis and the experimental one. From this comparison the elastic constraint identified was a distributed stiffness of 1010 N/m^3 , which provides good agreement between the experimental and numerical frequencies, all reported in Table 3.

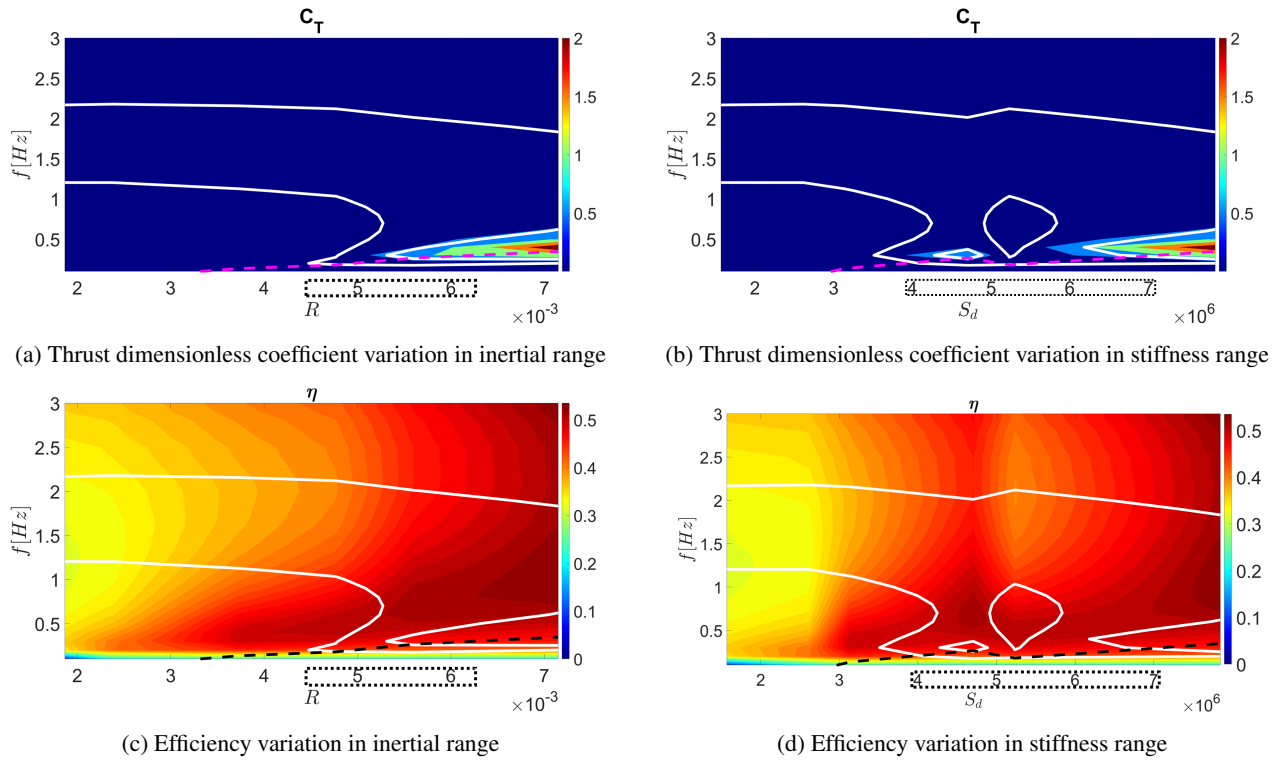
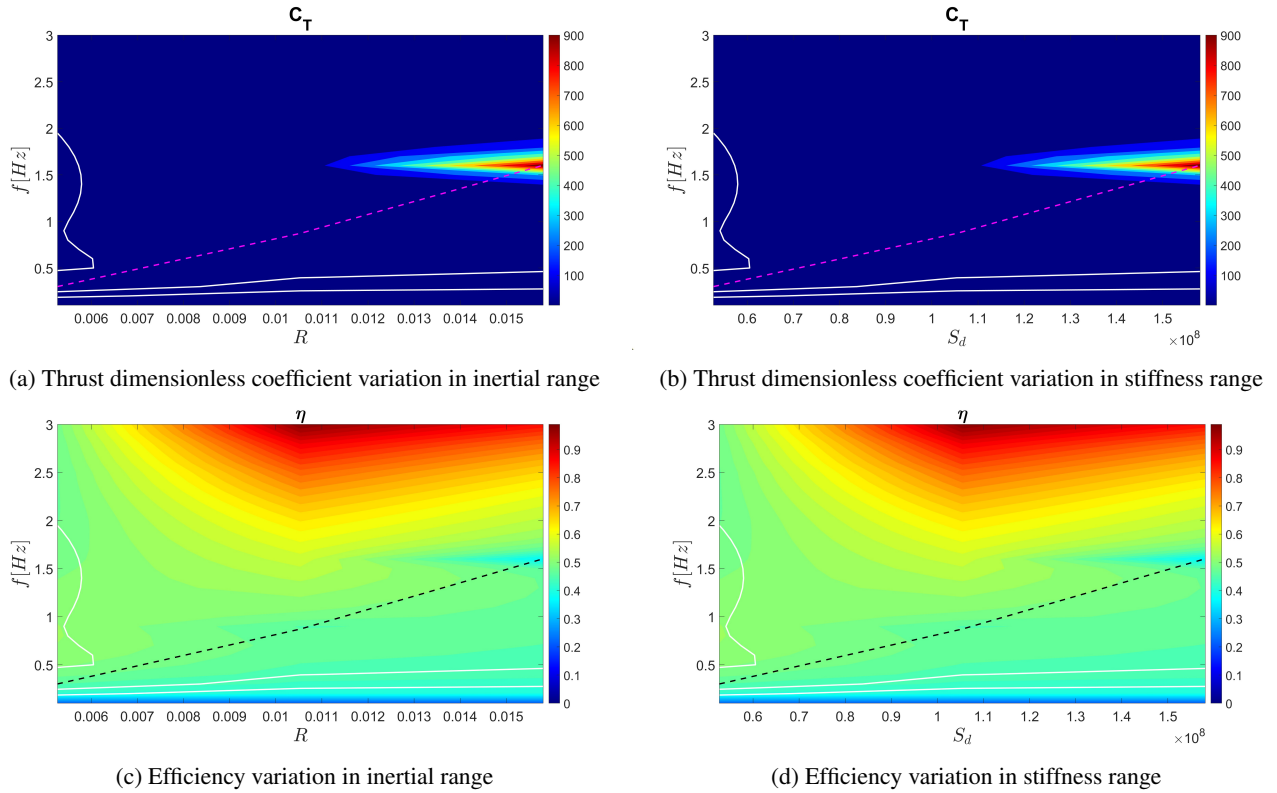
Table 3 Dry and wet fins natural frequencies in clamped configuration

Plate Thickness	Dry 1 st Frequency [Hz]		Wet 1 st Frequency [Hz]
	Exp	FEM	FEM
HDPE			
1 mm	-	0.96	0.134
2 mm	1.65	1.90	0.357
3 mm	2.46	2.80	0.576
PMMA			
1 mm	1.12	1.10	0.175
2 mm	2.07	2.16	0.469
3 mm	2.85	3.15	0.766
AL			
1 mm	3.16	3.17	0.749
2 mm	5.32	5.40	1.768
3 mm	6.14	6.14	2.33

To get the desired wet frequencies, fundamental parameter to evaluate the propulsion efficiency, a FEM analysis was carried on in Nastran software adding both the elastic support and the water body around the plate. The result can be seen in Table 3 whose values will be compared in the future with those measured experimentally.

Numerical and Experimental Results

After having characterized the experimental system, the dynamic simulations of all fins under analysis were conducted when the flow velocity is $U = 0.1 \text{ m/s}$, the maximum amplitude of the imposed pitching motion is $A_\alpha = 5^\circ$, in a range of oscillation frequency under 3 Hz chosen so as to ensure the visualization of the first wet natural frequency. Figures 4a and 4b show, for the analyzed operating conditions, the maps of the propulsive thrust parameter, as a function of frequency and of the inertia and stiffness dimensionless parameters, S_d and R , respectively. Figures 4c and 4d show the corresponding maps for efficiency. On each map the conditions for which an optimal Strohual number is obtained (solid white lines) and those for which the first resonance of the system is reached (dashed lines) are shown. This way of reporting numerical simulations allows one to quickly visualize the geometric conditions (combinations of S_d and R) such that optimal biomimetic propulsion is achieved, i.e when the resonance condition met the optimal Strohual condition (intersection of two lines). The maps in Figure 4 show the results for the plastic configurations, while those in Figure 5 of the metallic one. Under the same operating conditions, it can be seen that the rigid configurations will never guarantee propulsion optimality and for equal pitching condition the elastic configurations are still characterized by higher propulsive efficiency. The optimal range of S_d and R identified corresponds to the configurations PMMA 2 mm and HDPE 3 mm. The differences between the two optimal configurations are minimal: the 2 mm PMMA corresponds to $\eta = 0.44$ and a $C_T = 0.22$, while the 3 mm HDPE $\eta = 0.42$ and $C_T = 0.27$. Finally, 3 mm HDPE was considered as the best configuration because even with just a slightly lower yield it provides a larger C_T thrust gain. The experimental campaign was then started by analyzing this configuration at the experimentally identified wet resonance $f_\alpha = 0.2 \text{ Hz}$. It is important to underline that the experimental tests are preliminary and will require more attention on the individual setup subsystems, the error committed and the effect of the free surface.

**Fig. 4** Efficiency variation in stiffness range**Fig. 5** Rigid fins: propulsive map in 0-3 Hz range

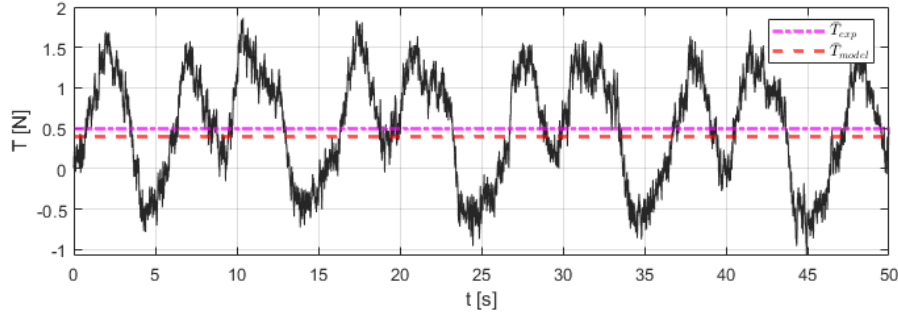


Fig. 6 Experimental evaluation of Thrust in time for the best configuration

From the measured thrust trend (Figure 6) an average value of approximately $\bar{T} \approx 0.5N$ (pink line) is estimated with a relative $C_T = 1.18$. Under the same conditions, the mathematical model identifies the first wet frequency at about $0.3Hz$. This discrepancy caused by the limitations of the developed approximate model, however, ensures good agreement in terms of qualitative comparison for propulsive force since $\bar{T} \approx 0.4N$ (red line) giving $C_T = 1.06$. The result shows a good starting point for the experimental results and a necessary correction of the model in use in order to estimate appropriate values for the coupling terms (added mass, hydrodynamic damping) that are critical for the wet frequency estimation.

Final Remarks

This article presents an analysis of the propulsive efficiency of elastic fins used as innovative propulsion systems for large fish-like robots. A semi-analytical coupled model, previously presented and validated by the authors, is utilized as a design tool to help choose the best fin configuration based on the conditions in which the robot will operate. A preliminary experimental campaign was conducted to identify the elastic materials properties, their natural frequencies through modal analysis, and frequency analysis to characterize the various subsystems involved in the experimental setup. This was designed in a hydraulic channel to simulate the oscillation of the fin in water. These experimental tests, although preliminary, are fundamental for an initial numerical assessment of the propulsive efficiency, which will be subsequently compared with the actual experimental results. The experimental campaign is currently in its initial stage and will be completed in the near future, considering a wider range of operating conditions, materials, and fin geometries.

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Chapter 9

Bandgap Design of Metamaterial Structures by Varying Local Resonator Properties

Hannes Wöhler and Sebastian Tatzko

Abstract The concept of metamaterials can be used in structural dynamics to reduce vibrations. By attaching local resonators of given eigenfrequency to a host structure, a so-called bandgap is created which indicates significant vibration reduction in a broad frequency range. Although the resonators are basically represented by spring-mass systems the bandgap phenomenon goes beyond the well-known tuned mass damper behavior. This is due to the fact that the resonators are positioned at different locations of the host structure. It is well known that by increasing the mass of the resonators, the bandgap width also increases. However, adding weight is not always a viable option when it comes to lightweight structures for example.

In this study we investigate new ways to increase the bandgap width without adding mass to the metamaterial structure. We consider a set of resonators for which we change individual stiffness values and investigate different arrangements of the eigenfrequency distribution of the resonators while keeping the total mass constant. The resonators influence the host structure dynamics and can be seen as a band-pass filter for mechanical waves which offers various possible applications in the field of vibration mitigation.

Keywords Metamaterial · Bandgap · Local resonators · Detuning · Mechanical filter

Introduction

In the field of structural dynamics, the use of the bandgap effect of metamaterials to attenuate vibrations is a current topic in research and development. Metamaterials consisting of a periodic e.g. geometric pattern generally offer a so-called phononic bandgap [1], which is in principle located in the higher frequency range. The origin of this effect can be traced back to research on phononic crystals [2]. Another approach is to add local resonators in a periodic pattern to a host structure, where traditionally resonators tuned to identical frequencies are distributed across the structure. This also creates a bandgap, which in this case is based on the so-called resonator effect. The big advantage for structural dynamics is that the usual bandgap is in a much lower frequency range. It is determined by the arrangement and properties of the resonators [3–5].

Recent research has extended this approach by exploring variations in resonator tuning to achieve multiple [6] and wider bandgaps [7, 8]. One such concept is a series of differently tuned resonators to cover broader frequency ranges. This study builds on by proposing systematic detuning patterns of resonators attached to a segment of a host structure. The main goal is to increase the bandgap width by employing different configurations of resonator detuning.

To investigate the effectiveness of these configurations, we use a basic lumped-mass model that serves as a simplified framework for analyzing different metamaterial arrangements. The study examines several detuning patterns, including alternating and linearly in-/decreasing triangular formations, each with three levels of detuning. In addition, we compare the calculated bandgap widths of these finite lumped-mass models with those obtained from the Bloch's theorem model, which uses a unit cell with complex boundary conditions to calculate a dispersion diagram [9, 10].

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Modeling of Metamaterial Structure

This work is based on a metamaterial model consisting of a lumped-mass system, which is described via mass matrix M , damping matrix C and stiffness matrix K . In this structure, the resonator effect is utilized to achieve a bandgap considering the displacement amplitudes from one end to the other. Figure 1 shows that seven resonators are coupled to the central part of the host structure which is an oscillator chain with 15 degrees of freedom (DOFs). Each host mass is coupled to the environment via a linear spring. The whole metamaterial structure is base excited via stiffness at the first DOF x_{exc} .

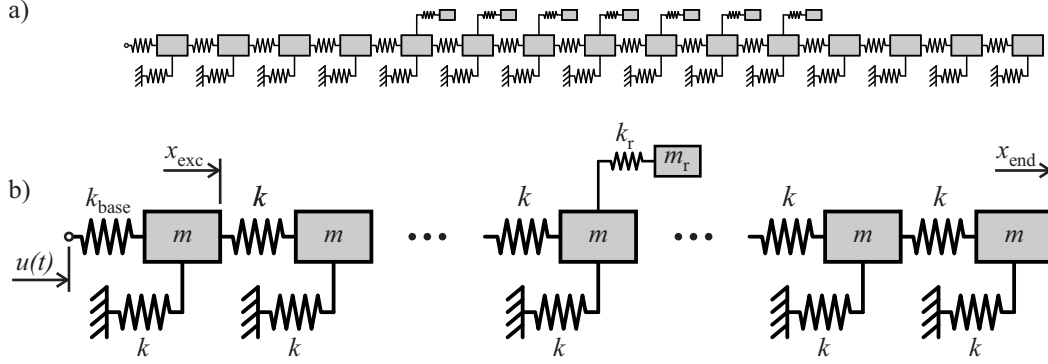


Fig. 1 Lumped-mass model of the metamaterial structure with base excitation: a) full system with 15 host masses and seven resonators; b) detailed view of parts with labeling.

Table 1 shows the parameters of the lumped-mass model of the metamaterial structure in Figure 1. The resonator masses of the reference model are 10 % of the host masses $m_r = 0.1 \cdot m$ for which a significant bandgap is already observed. For comparison, the resonator masses are doubled in another variant to 20 % of the host masses $m_r = 0.2 \cdot m$ to show the effect of bandgap enlargement by simply adding mass. Figure 1 does not show any dashpot dampers, but they are modeled proportional to the stiffnesses k and k_r using Rayleigh-damping in the form of $C = \beta K$.

Table 1 Parameters of lumped-mass model and equally tuned resonators, cf. Figure 1.

	k_{base} in $\frac{N}{m}$	m in kg	c in $\frac{Ns}{m}$	k in $\frac{N}{m}$	m_r in kg	c_r in $\frac{Ns}{m}$	ω_r in $\frac{rad}{s}$	k_r in $\frac{N}{m}$
equal (reference)	1000	0.1	$1.5 \cdot 10^{-7}$	8000	0.01	$15 \cdot 10^{-7}$	440	1936
equal 2-mass	1000	0.1	$1.5 \cdot 10^{-7}$	8000	0.02	$15 \cdot 10^{-7}$	440	3872

The focus is on investigating the influence of different resonator detuning patterns with respect to the bandgap width at DOF x_{end} . Figure 2 shows the different detuning patterns as bar plots. The metamaterial structure with seven resonators of equal mass each tuned via stiffnesses k_r to the natural frequency of $\omega_{r,1-7} = 440 \frac{rad}{s}$ serves as a reference, see Figure 2 a). Two different detuning patterns with five detuning levels each are investigated for the seven resonators. On the one hand, an alternating ABABABA-pattern, in which the resonators are detuned by $\pm 5\%$ ($\pm 10\%$, $\pm 15\%$, $\pm 20\%$, $\pm 25\%$) in relation to the base frequency of $440 \frac{rad}{s}$. These are referred to as AB-5, AB-10, AB-15, AB-20 and AB-25, see Figure 2 b)-f).

On the other hand, a triangular Tri-tuning in the form of ABCDCBA is investigated. Here, resonators #1 and #7 are detuned by -5% (-10% , -15% , -20% , -25%) and resonator #4 by $+5\%$ ($+10\%$, $+15\%$, $+20\%$, $+25\%$). The resonators in between are detuned according to linear interpolation. The latter variants are referred to as Tri-5, Tri-10, Tri-15, Tri-20 and Tri-25, see Figure 2 h) - l). Note that the resonator masses for all configurations of patterns AB-tuning and Tri-tuning are equal to the resonator mass of the reference structure. This way the resonator masses stay unchanged and the bandgap width is only affected by detuning and not by mass effect. The eigenfrequency is thus only set via the resonator stiffnesses k_r .

Analysis of Bandgap Properties

The analysis of the band gap properties with regard to the band gap width is carried out in two ways, which are explained in the following section. First, the bandgap width will be determined by modeling the metamaterial structure as a unit cell

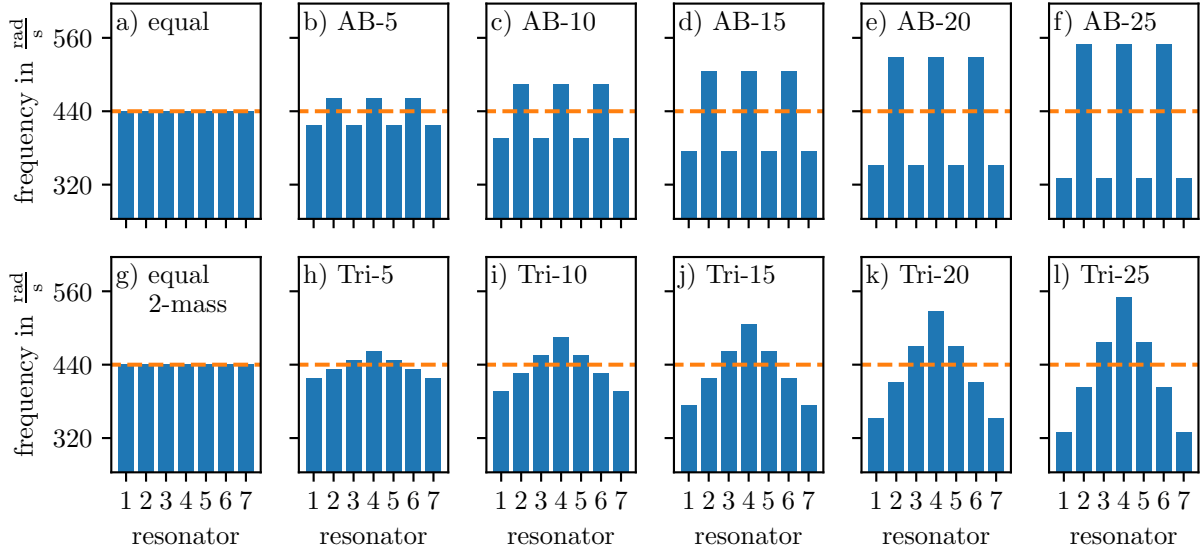


Fig. 2 Configurations of metamaterial structure with seven resonators with base angular frequency of $440 \frac{\text{rad}}{\text{s}}$: a) equally tuned; b) - f) with alternating AB-pattern at 5 %, 10 %, 15 %, 20 % and 25 % detuning level; g) equally tuned with doubled resonator mass; h) - l) with triangular pattern at 5 %, 10 %, 15 %, 20 % and 25 % detuning level.

model using Bloch's theorem. In a second approach, a method can be used in which the displacement transmissibility is calculated in order to obtain the bandgap width.

Bandgap width approximation via Bloch's theorem

The bandgap properties of periodic metamaterial structures are often determined using Bloch's theorem [9]. For this purpose, complex boundary conditions with continuous phase shifts μ between 0 and π (rad) are applied to the so-called unit cell. This is equivalent to an infinite repetition of this unit cell. For each phase shift μ , an eigenvalue problem of the Bloch matrices $\mathbf{M}_{\text{Bloch}}(\mu)$ and $\mathbf{K}_{\text{Bloch}}(\mu)$ must be solved [10]. This method is comparatively fast and computationally efficient, as it reduces the problem dimension to its unit cell representation. This type of bandgap calculation has however the following disadvantages: edge reflections of finite structures are not taken into account, detuning of the resonators cannot be mapped and resonators that are not evenly distributed over the host structure cannot be represented either. Nevertheless, the Bloch's theorem based calculation of the bandgap provides useful results with regard to the general design of a metamaterial structure. Figure 3 shows in a) the unit cell of the reference metamaterial structure with the parameters of equally tuned resonators (equal) and in b) the corresponding dispersion diagram. The blue line belongs to the square root of the first eigenvalues $\omega_1(\mu)$, while the orange line belongs to the square root of the second eigenvalues $\omega_2(\mu)$. The angular eigenfrequency of the resonator ω_r is shown as a dashed black line and the gray band refers to the bandgap, where no eigenvalues exist, which causes significant reductions in vibrations amplitudes transferred to the finite metamaterial structure with resonators. The calculated width of the bandgap via Bloch's theorem is $51.0 \frac{\text{rad}}{\text{s}}$ and serves as a first reference for a corresponding finite metamaterial oscillator.

Bandgap width calculation via transmissibility

Another way to quantify the bandgap width is to use the transmissibility method presented in this paper. Figure 4 a) shows the displacement amplitudes of the DOFs x_{exc} and x_{end} of the base excited host structure without resonators in comparison to the displacements of the reference metamaterial structure in b). It is clearly seen that there is an anti-resonance at $440 \frac{\text{rad}}{\text{s}}$, which is the natural angular frequency of the resonators. Besides, the displacement amplitudes of x_{end} are significantly lower than those of x_{exc} in a broad frequency range around the resonator's natural frequency. The shape is significantly wider compared to a typical anti resonance. Figure 4 c) shows the transmissibility in dB

$$T(\Omega) = 10 \cdot \log \left(\frac{|\hat{x}_{\text{end}}(\Omega)|}{|\hat{x}_{\text{exc}}(\Omega)|} \right) , \quad (1)$$

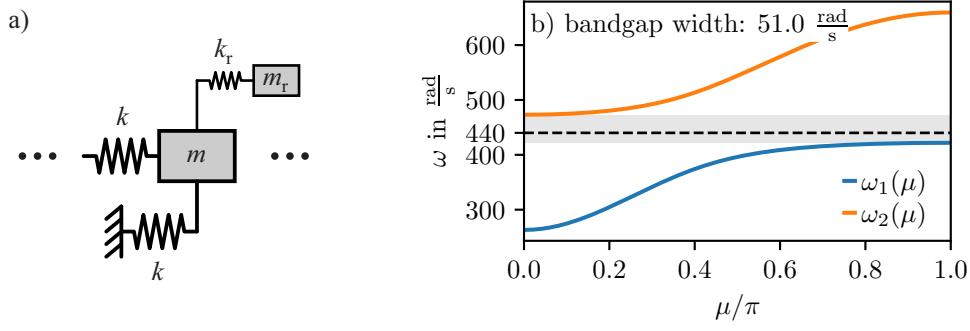


Fig. 3 Analyzing of bandgap width by unit cell approach using Bloch's theorem: a) lumped-mass model of unit cell; b) resulting dispersion diagram with dispersion curves $\omega_1(\mu)$ and $\omega_2(\mu)$ and bandgap (gray box).

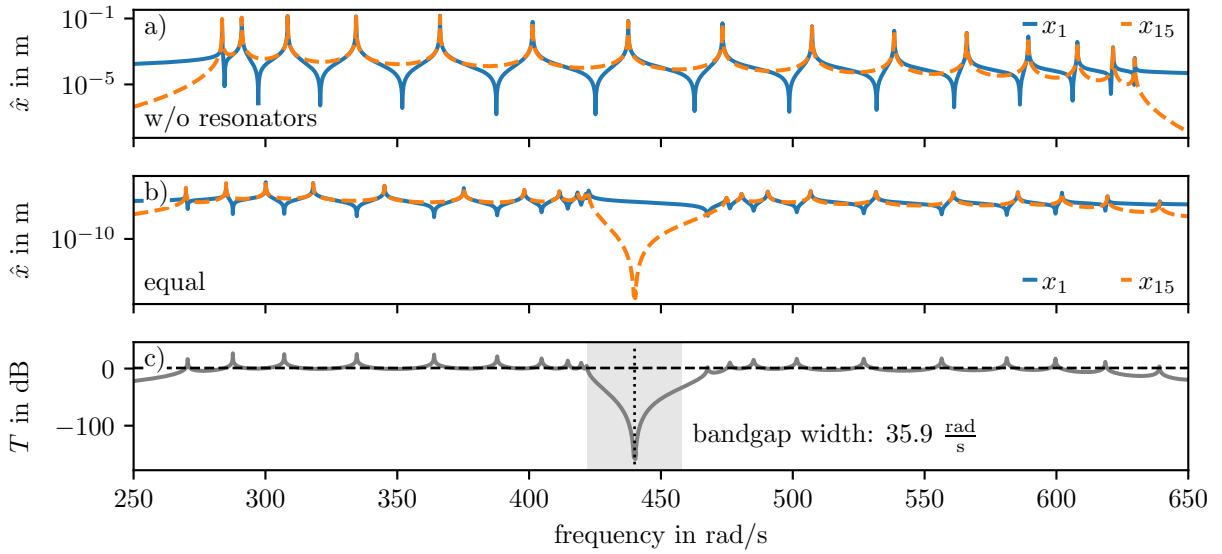


Fig. 4 Calculation of bandgap width via transmissibility: a) displacement amplitudes x_{exc} and x_{end} of host structure without resonators; b) displacement amplitudes x_{exc} and x_{end} of metamaterial structure (equal); c) corresponding transmissibility T for the configuration equal: gray box marks identified bandgap, vertical dotted line marks center of bandgap at resonator's base frequency of $440 \frac{\text{rad}}{\text{s}}$.

which provides a robust basis for calculating the bandgap width of a finite metamaterial structure using e.g. displacement amplitudes. The bandgap is defined by values of $T(\Omega) < 0$ dB near the resonator's base natural frequency. Another criterion is that the identified bandgap width is centered at the resonator's natural frequency to avoid biased result in bandgap evaluation. Consider a desired bandgap at frequency f_0 for which the goal is reduced to just increase bandgap width. One may end up with a large bandgap onesided to either high or low frequency values. For meaningful results, we therefore introduce this symmetry constraint. The bandgap width calculated via transmissibility is $35.9 \frac{\text{rad}}{\text{s}}$. In comparison, by applying the symmetry constraint to the identified bandgap information using Bloch's theorem in Figure 3 b), the adjusted bandgap width is $36.0 \frac{\text{rad}}{\text{s}}$ instead of $51.0 \frac{\text{rad}}{\text{s}}$ without symmetry constraint.

Results of Bandgap Design

Twelve metamaterial variants are analyzed with regard to their bandgap widths. The bandgap width is determined using the transmissibility method described above, as Bloch's theorem is not applicable due to the more complex detuning patterns of the resonators. The identified bandgap widths of the investigated configurations are shown in Figure 5. The relative bandgap

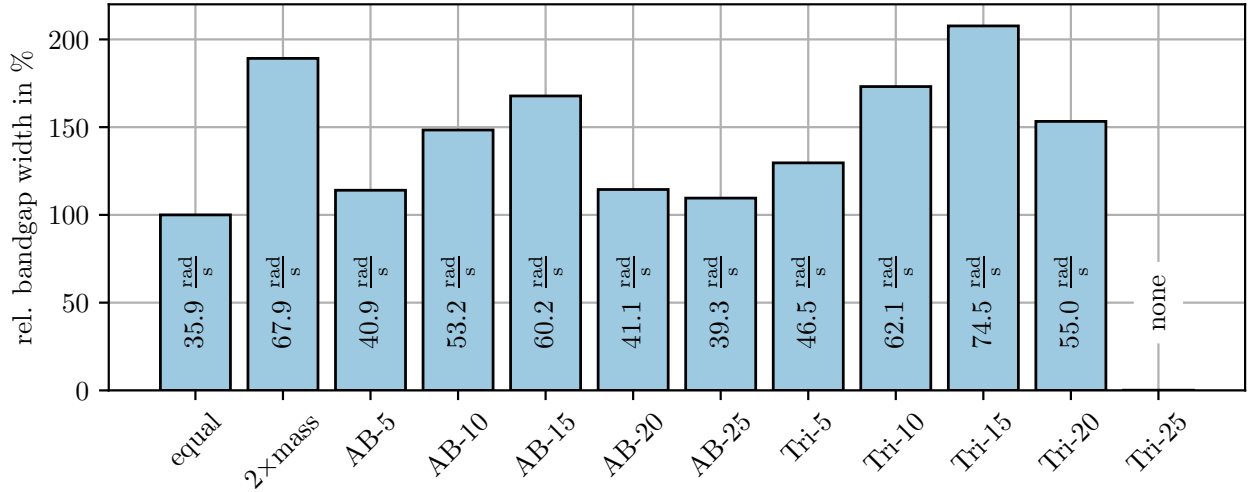


Fig. 5 Bandgap widths at different metamaterial configurations calculated via displacement relation method.

width is also shown, where 100 % corresponds to the reference configuration (equal) with $35.9 \frac{\text{rad}}{\text{s}}$. By using resonators that are twice as heavy but equally tuned, the bandgap can be widened, achieving a relative width of 189.2 %.

As increasing the mass of the resonators may not always be a feasible option, the bandgap width can also be influenced by systematically detuning the resonators holding the mass constant. In this study, two detuning patterns - alternating AB-tuning and triangular-tuning - each with five levels of detuning are analyzed for their potential to widen the bandgap, see Figure 2.

For the AB configurations AB-5, AB-10 and AB-15, an increase in the bandgap width compared to the reference is observed as the detuning level increases, see Figure 5. However, if the detuning level is increased further to 20 % or 25 %, the bandgap width is reduced again due to anti resonance phenomena as shown in the following.

Figure 6 shows in a) the displacement amplitudes at x_{exc} and x_{end} and in b) the corresponding transmissibility T . These plots visualize the drop in bandgap width of the configuration with ± 20 % detuned resonators. Looking at x_{end} in a), vibration response levels are reduced in a wide frequency range. The transmissibility however reveals due to an anti resonance of x_{exc} a local peak at $\approx 460 \frac{\text{rad}}{\text{s}}$ at a certain degree of detuning. As we are interested in a bandgap around $440 \frac{\text{rad}}{\text{s}}$, this phenomenon creates a new limit and corresponding sudden drop in effective bandgap width. The design of

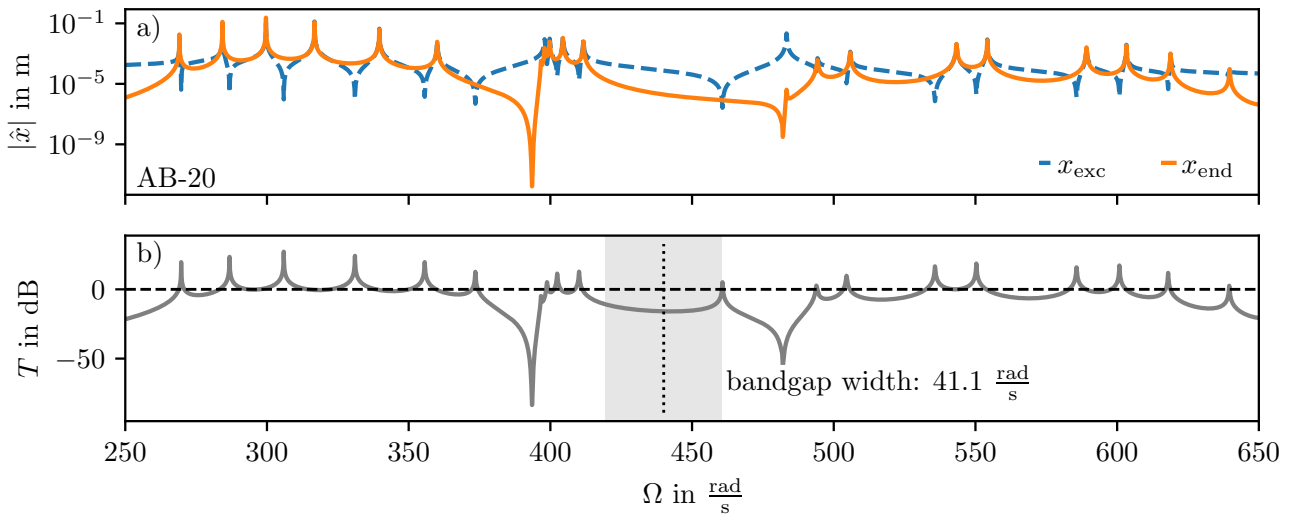


Fig. 6 Configuration AB-20: a) displacement amplitudes x_{exc} and x_{end} ; b) corresponding transmissibility T .

metamaterial structures with respect to multiple bandgaps using different types of resonators is described in [6] and beyond the scope of the current work.

The variants with triangular detuning exhibit an increase in bandgap width up to 15 % detuning level. Notably, the Tri-15 configuration results in the widest relative bandgap of 207.7 %, surpassing even the configuration with doubled resonator mass, see Figure 7. This is particularly remarkable considering the aforementioned limitations of increasing resonator mass. It should be emphasized that no bandgap could be identified in the Tri-25 configuration, as the transmissibility is greater than 0 dB at the resonator's base frequency of $440 \frac{\text{rad}}{\text{s}}$ and the bandgap criterion is therefore not fulfilled.

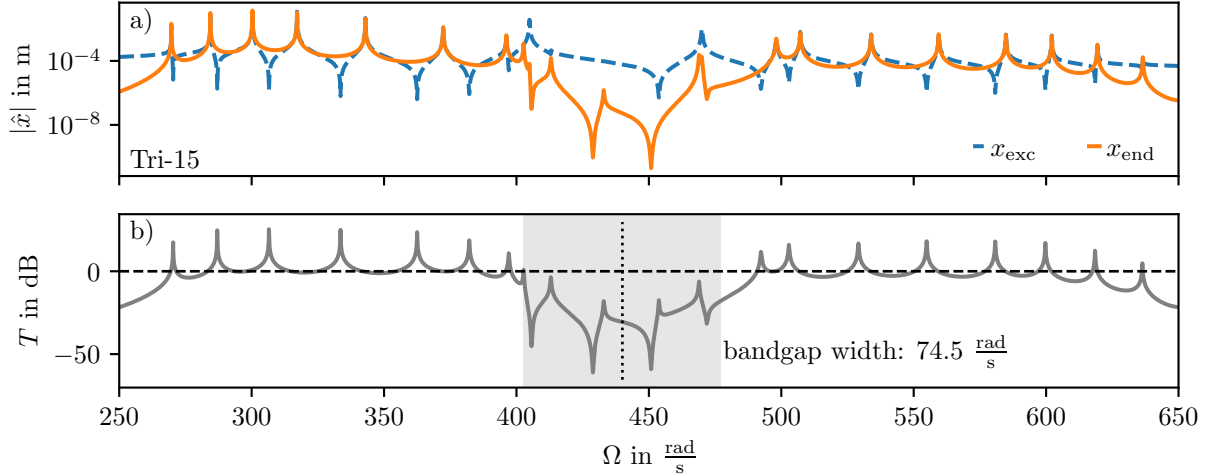


Fig. 7 Configuration Tri-15: a) displacement amplitudes x_{exc} and x_{end} ; b) corresponding transmissibility T .

Conclusion

In conclusion, the study investigates the effects of different detuning patterns on the bandgap width of a metamaterial structure with seven resonators, compared to the reference case of uniformly tuned resonators. It is well known that increasing the mass of the resonators results in an increased bandgap width. However, the focus of this work is to explore how detuning patterns, specifically an alternating detuning AB-pattern and a triangular detuning Tri-pattern, impact the bandgap width. The results demonstrate that a particular detuning pattern can achieve a wider bandgap than what could be attained merely by using heavier resonators. This insight is significant for the design and optimization of metamaterial structures, potentially leading to improved performance in various applications by strategically adjusting the detuning of resonators rather than solely relying on changes in mass.

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Chapter 10

Experimental Investigation of the Damping Effect of Inherent Particle Damping Under Centrifugal Load



Mirco Jonkeren and Sebastian Tatzko

Abstract The passive damping mechanism of particle damping is capable of influencing the structural dynamic behavior of mechanical structures in terms of vibration mitigation. Especially additive manufacturing technologies offer a huge potential to integrate powder-filled cavities into structures and thus enable this inherent damping effect. The potential was already shown for additively manufactured beams but also for final products like gears. However, for rotating components such as gears, the centrifugal load may affect the damping mechanism adversely as relative motion of particles could be reduced. In this experimental study, we investigate particle damping in structural dynamics under rotation. For this purpose, we compare the damping behavior of additively manufactured gears with and without inherent particle damping under centrifugal load. For varying rotational speeds the vibration response is measured using a laser vibrometer that rotates synchronously with the gear. An automatic impulse hammer ensures identical excitation in each measurement while the applied force is also monitored. Gears with particle damping show a significant increase in damping. An essential dependency on centrifugal load is shown here, which increases the damping even further. Our findings contribute to the basic understanding of particle dynamics under centrifugal load and opens new paths to exploit particle damping in rotating structures.

Keywords Particle damping · LPBF · Centrifugal load · Gear · Rotational laser vibrometer

Introduction

Particle damping is a damping mechanism where cavities in a vibrating structure are filled with particles, either added or left as unmelted powder through additive manufacturing, such as laser powder bed fusion (LPBF). The energy dissipation of this mechanism is based on friction mechanisms and inelastic impacts, which serve to effectively reduce the vibrations [1]. While structures with conventionally added particles have been the subject of detailed investigation in the past, research into powder-based particle dampers has developed in connection with the field of additive manufacturing. The additive manufacturing process enables the inclusion of unmelted powder within the cavities of a structure, thereby expanding the potential for innovative construction solution [2, 3, 4, 5].

The effect of particle damping on mechanical structures, such as beams, has been the focus of previous studies, particularly in relation to non-rotating motion. Nevertheless, with regard to rotating systems, the available research is less extensive. A general study on the energy dissipation characteristics of granular materials sloshing in a rotating container is presented in [6]. In [7], investigations are carried out on a rotating beam on the effectiveness of particle dampers under centrifugal loads, whereby the particle damper oscillate in a direction perpendicular to the centrifugal load. A dependence between the ratio of the centrifugal acceleration to the maximum acceleration amplitude and the performance of the particle damper is shown. The application of particle dampers in gear systems is presented in [8, 9, 10], where holes are drilled in the gears and filled with particles. It is observed that the total energy dissipation of the damper also increases with an increase in rotational speed. The positive correlation between the damping effect and rotational speed is only given within a limited speed range. The combination of particle damping utilizing powder bed based additive manufacturing processes and its application in gears has been demonstrated in [11]. It is shown that the integration of powder-filled cavities resulted in a significant reduction in the vibration amplitude. Nevertheless, further research is required to gain a more comprehensive understanding of the behaviour of particle damping in additively manufactured gears under centrifugal load.

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Experimental Methodology and Analysis

A series of tests were carried out on a set of gears manufactured additively from X3NiCoMoTi 18-9-5, as presented in [11]. Laser powder bed fusion is the additive manufacturing process employed to achieve this. The characteristics of a fully fused gear are compared with those of two gears containing unmelted powder in integrated cavities in the teeth. The two gears differ in terms of their cavity sizes (G-02: small cavities and G-03: large cavities). The experimental setup with one of these gears is shown in Figure 1. The gears are mounted on a sleeve that is connected to a shaft. The sleeve has been designed with the objective of minimizing the contact area between the gear and the rotating shaft of the test rig, thus reducing the friction damping between them (see detail view in Figure 1, in which the sleeve is shown in black and the contact surfaces are marked in red). The excitation is applied as an impulse to the gear with an automatic impulse hammer (see Figure 1 right). The out-of-plane vibration response is measured using a scanning laser doppler vibrometer (SLDV) that rotates synchronously with the gear (see Figure 1 left). An optical derotator, manufactured by *Polytec*, is employed for this purpose. This enables the non-contact measurement of vibration on rotating objects, based on a rotating Dove prism that enables speed-controlled tracking of the laser beam. The speed information required for control is provided by an encoder signal from the drive motor. Subsequently, a Fourier transformation was performed and the calculated spectra of a total of ten individual measurements per rotational speed were averaged. The rotational speed was varied from 0 rpm to 400 rpm in steps of 20 rpm. Figure 2 shows an example of an averaged spectrum of the fully fused gear. The identified natural frequencies were then employed in the application of a filter for specific eigenmodes. The present study examines the vibration mode, which is also shown in Figure 2. For the fully fused gear without integrated particle damping, the corresponding natural frequency is 12385 Hz as illustrated by the red circle. Signal filtering is conducted using a digital bandpass FIR filter (finite impulse response filter) with a cutoff frequency of 50 Hz below and above the peak of the frequency of the examined vibration mode. These upper and lower cutoff frequencies are shown by the red transparent area in Figure 2. In accordance with [12, 13], it is crucial that the input data is processed in both the forward and reverse directions during the filtering process. The filtered time data is exemplarily shown in Figure 3 for the fully fused gear (left) and the gear G-03 (large cavities) with particle damping (right). It is noticeable that the vibration amplitudes of the examined mode decay at a slower rate for the fully fused gear than for the gear with powder-filled cavities. This filtered time data is used to identify the damping ratio of that single mode. For this, an exponentially decaying function is utilized to fit an envelope to the filtered time data of each measurement:

$$\dot{x}(t) = \hat{x}_0 e^{-\delta t} \quad (1)$$

Here $\dot{x}$ represents the surface velocity, δ the exponential decay constant. $\hat{x}_0$ is the initial value of the surface velocity at the specified time, which is defined at the time of the maximum value of the measured time data after excitation (see in Figure 3 near 0.05 s). For such an individual mode, the damping ratio D is defined as:

$$D = \frac{\delta}{\omega_0} \quad (2)$$

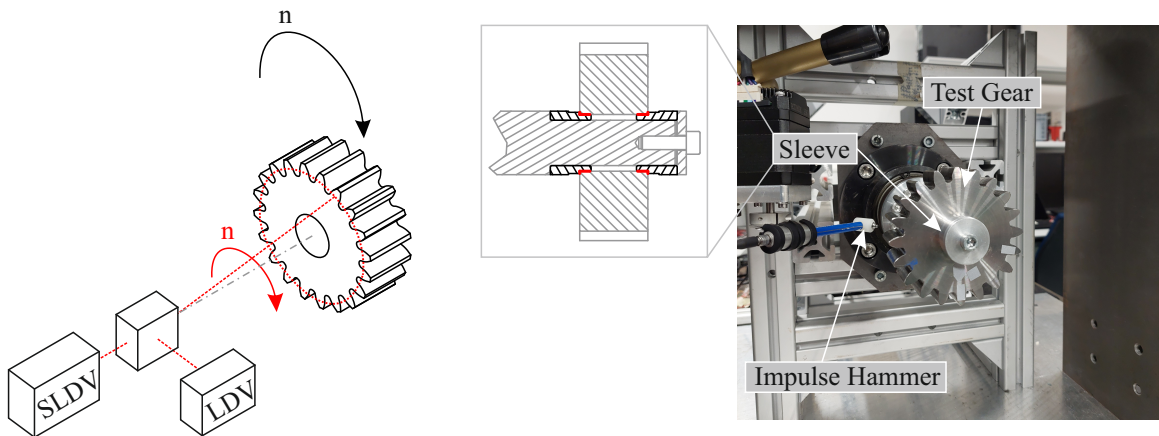


Fig. 1 Schematic illustration of the experimental test setup for measuring the system response of rotating gears out-of-plane (left). Experimental setup with mounted gear and the automatic impulse hammer used for excitation (right).

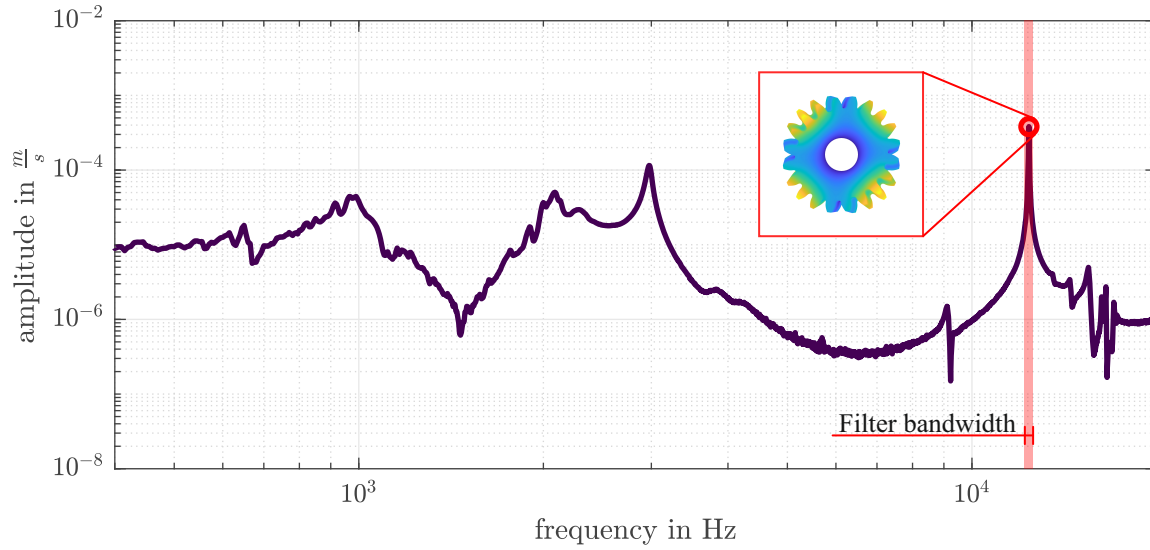


Fig. 2 Averaged frequency spectrum of the system response of the fully fused gear. The frequency range employed for signal filtering is indicated by a red transparent area above and below the analyzed resonance frequency (red circle). Corresponding vibration mode (detail view).

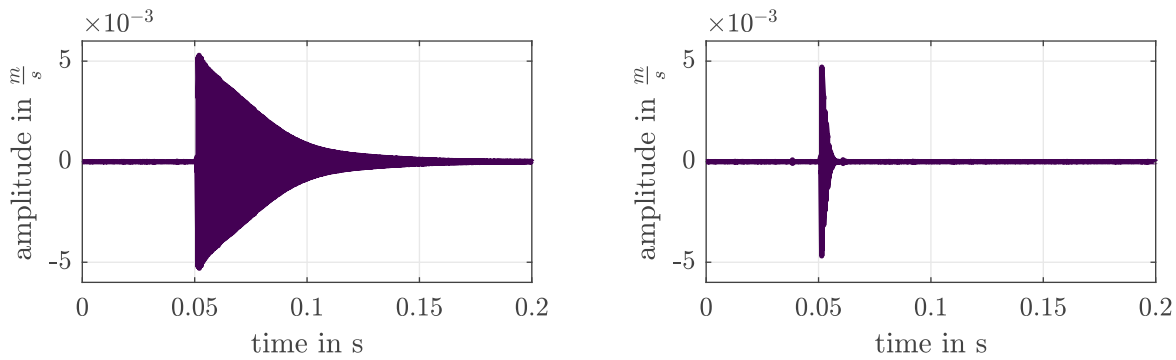


Fig. 3 Filtered time data at a rotational speed of $n = 300$ rpm. Free decay of the fully fused gear without integrated particle damping (left) and for the gear G-03 (large cavities) with integrated particle damping (right).

The natural frequency of the undamped system ω_0 is related to the damped natural frequency ω_d by:

$$\omega_0 = \frac{\omega_d}{\sqrt{1 - D^2}} \quad (3)$$

This leads to the following equation for calculating the damping ratio:

$$D = \frac{\delta}{\omega_d} \frac{1}{\sqrt{1 + \frac{\delta^2}{\omega_d^2}}} \quad (4)$$

Figure 4 shows the damping performance of the fully fused gear in comparison to the two gears with powder-filled cavities in terms of the calculated damping ratio according to equation (4) for different rotational speeds. The mean value of the ten individual measurements for each rotational speed along with the standard deviation are presented. It can be observed that the damping ratio of the gears with integrated particle damping is higher in all speed ranges. The examination of the data reveals a greater range in damping values for the gears with integrated particle damping in comparison to the fully fused gear without. In particular, the data set exhibits a larger standard deviation at 220 rpm, 280 rpm and above from 360 rpm, which may be indicative of outliers.

A comparison of the gears reveals that the damping ratio remains nearly constant over the examined range of rotational speed for the fully fused gear. For the gear with the large integrated and powder-filled cavities it can be observed that the

damping ratio increases with an increase of rotational speed. This trend is observable across the entire speed range. These results demonstrate a qualitative trend that is in accordance with the findings of [8]. This tendency is not as pronounced for the gear G-02 with the small cavities, but a slight increase can also be observed.

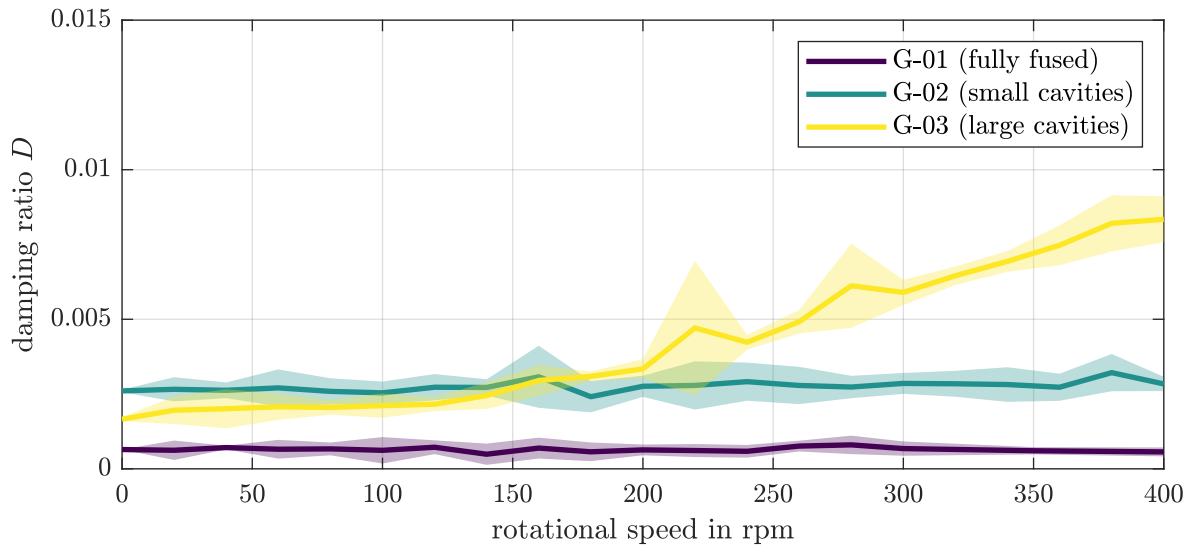


Fig. 4 Calculated damping ratios of the examined vibration mode for the fully fused gear and the two gears with integrated powder-filled cavities. The graphs show linear interpolation between calculated damping ratios at discrete rotational speeds. Corresponding standard deviations are indicated by transparent areas.

Conclusion

In conclusion, this experimental study investigates the dynamic response of three additively manufactured gears excited with a deterministic impulse excitation signal. The system response is measured using a laser vibrometer that rotates synchronously with the gear to capture the decaying time history at various rotational speeds. By conducting ten measurements at each rotational speed and analyzing the frequency spectrum of the system response, the decaying time data were filtered to isolate and examine one specific mode. The damping ratio for this vibration mode was calculated for each rotational speed. The results demonstrate that the damping increases with rotational speed for powder-filled gears, especially those with large cavities. Damping of the fully fused gear did not show any dependence on rotational speed. These findings contribute significantly to the understanding of the benefits of particle damping in gear systems.

In future, further investigations should aim to provide a detailed analysis of particle motions and interactions within the cavities under centrifugal load. It would be beneficial to analyze the trend in further studies at higher speeds in order to determine whether the damping ratio eventually reaches a plateau or whether it begins to decline at some point, as reported for example in [8]. A detailed analysis of the observed damping fluctuations at high speeds, in particular the outliers and speeds above 360 rpm, would be useful. It is important to determine whether these fluctuations are due to the system or are influenced by the test setup. This consideration is relevant as tangential forces can be induced during excitation by surface roughness and hammer tip, which may have a greater effect at higher speeds.

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Chapter 11

Greedy Source Placement Optimization for MIMO Control of Ground-Based Vibration Tests



Chandler Smith, Timothy Walsh, Wilkins Aquino, Drew Kouri, and Ryan Schultz

Abstract In this work, we investigate the effectiveness of greedy algorithms in optimizing the placement of electrodynamic shakers to achieve optimal control in Multiple-Input Multiple-Output (MIMO) vibration tests. In the context of ground-based vibration tests, control algorithms are used to estimate excitation signals that will produce a specific vibration response in the test object. The challenges in control lie in estimating the drive signal, which is inherently an inverse problem, and in satisfying the limited input energy available from the shaker's power. To this end, we investigate an inverse problem formulation that allows the user to weight their preference for matching target response data versus minimizing the demand on input power. The success of the proposed algorithm depends critically on the shaker locations, where poor placement results in diminished controllability resulting in unachievable drive powers and(or) lack of proper control of the system.

Optimizing the placement of sources is a Mixed Integer Programming (MIP) optimization problem, known for its computational expense and complexity. Greedy algorithms present a potential solution by trading optimality for enhanced computational performance. In our research, we conduct a comparative analysis between a greedy algorithm and Gurobi, a powerful optimization tool for solving MIP problems, to evaluate the greedy algorithm's capability in identifying effective shaker locations.

Our results demonstrate that the greedy algorithm often matches solutions to the MIP program. In addition, our study indicates that the ideal placement of the shakers is influenced by the user's preference for minimizing input energy versus achieving target response accuracy. This highlights the importance of customizable parameters in achieving optimal system control through strategic source placement.

Introduction

Ground-based vibration testing using Multiple Input Multiple Output (MIMO) control is an advanced technique for simulating flight environments in order to evaluate the dynamic behavior of the vehicle and its components. MIMO control involves the simultaneous application of multiple excitation forces and the measurement of responses at multiple points. This approach provides a more comprehensive understanding of the structural dynamics, enabling the accurate reconstruction of the system's dynamic response during operation in a flight environment. However, MIMO control also presents many challenges, which can significantly impact the effectiveness and accuracy of the vibration testing. This work focuses on two of those challenges: the spatial configuration of the electrodynamic shakers (sources) and the control strategy.

There are a few key considerations when placing sources and choosing a control strategy used to determine the excitation signals of the driving shakers. First, the control strategy and placement of the shakers must be carefully chosen to effectively excite the expected dynamic response produced in a flight environment. For example, authors in [1] demonstrated how the quality of a shock response MIMO test changes with actuator locations. Second, the shakers must be placed in physically feasible locations given the constraints of the testing space and test article. Third, the shakers maximum force capacity and

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output energy are finite; hence shaker configurations and control strategies should ensure that the available energy is evenly distributed across all the shakers [2]. A general framework for approaching the optimal source placement problem is as follows: i) select a control strategy, ii) determine a quantitative measure of the quality of the source configuration, and iii) optimize the source locations by maximizing the selected metric.

In the context of structural dynamics, optimal control and source placement applications has largely focused in the area of active structural damping for mitigating vibrations [3]. Conceptually, MIMO control and active damping are similar in that the goal is to achieve a desired structural state with minimal energy. Much of this theory is based on state-space formulations with a critical focus on placing source that maximize controllability [4]. Research on source placement for MIMO vibration testing is less prolific in the literature. In [5], a greedy-based approach was taken to select actuators that minimized the output energy of the actuator in order to achieve a desired random vibration specification. In [6], another greedy search method was proposed, which aimed to minimize the random vibration frequency response match to some target response. This method performed well compared to a brute force exhaustive search.

Regardless of the method, source placement optimization involves solving an NP-hard mixed integer optimization problem. While branch and bound methods can be employed, they are significantly constrained by problem size due to their computational complexity. As a more efficient alternative, greedy optimization approaches can be used, offering faster solutions at the expense of optimality. Greedy search algorithms involve iteratively adding sensors one at a time, each time choosing the sensor that minimizes some user-specified objective function.

In this work, we develop a MIMO control and source placement optimization framework. The core of the approach involves solving a regularized inverse problem, which ensures stability and robustness in the presence of measurement errors, while providing a user-specified regularization parameter that controls the trade-off between the fidelity to the target dynamic response and the smoothness and magnitude of the control inputs.

We propose two source placement optimization strategies that use a greedy search algorithm and compare these designs by applying them to the regularized inverse problem control strategy. The first method seeks to directly minimize the objective function corresponding to the l_2 -norm regularized inverse problem, with the solution depending on the user-selected regularization parameter. Recognizing that the greedy search is not guaranteed to provide a near-optimal or optimal solution, we compare this method to an equivalent mixed-integer quadratic programming (MIQP) problem and to randomized brute force search.

Inspired by the optimal sensor placement literature, the second proposed source optimal experiment design (OED) framework aims to minimize the E-criterion. The E-criterion benefits from guarantees on the quality of the greedy-generated solution [7], and has the added advantage of being independent of the choice of regularization parameter and of the target response data. In this work, we show that minimizing the E-criterion also indirectly minimizes the input energy of the least squares control solution.

We compare the two source placement OED frameworks in terms of their performance in solving the l_2 -norm regularized inverse problem. The control and source placement framework is applied to a MIMO problem that seeks to reproduce the dynamic response of a sharp cone flight vehicle subjected to a turbulence-induced, fluctuating pressure field using a limited number of shakers.

This work is broken down into the following sections. The Background and Methodology section describes the MIMO control formulation followed by the proposed source placement optimization algorithms. In the MIQP Greedy Search Comparison section, we compare the performance of the l_2 -norm regularized greedy search method to the MIQP problem in terms of solution accuracy and computational runtime. In the Greedy OED Performance on Full-Scale Model section, we investigate the performance of the greedy method on a full-scale wind tunnel model. In this section, we also compare the E-optimal design and l_2 -norm regularized designs, and investigate their robustness to changes in the regularization parameter used to solve the control inverse problem.

Background and Methodology

Assume we are given a target response $\hat{u}_o(\omega)$ where ω is angular frequency and a maximum number of actuators N . The goal is to find sources $s(\omega) \in \mathbb{C}^d$, with $d \leq N$, and their locations such that the output response $u(\omega)$ of the MIMO vibration test is close to $\hat{u}_o(\omega)$.

Assuming the structural response is defined by a Linear Time Invariant system, the map from the sources to the acceleration is taken as

$$u(\omega) = H(\omega)s(\omega), \quad (1)$$

where $H(\omega) : \mathbb{C}^d \rightarrow \mathbb{C}^n$ is the transfer operator (or system frequency response function matrix), $u(\omega) \in \mathbb{C}^n$ is the acceleration response, and $s(\omega) \in \mathbb{C}^d$ is the source vector.

To solve this inverse problem, we introduce an l_2 -norm regularization term to form the Tikhonov regularization problem. The regularized control inverse problem is taken as,

$$\min_{s(\omega) \in \mathbb{C}^d} \frac{1}{2} \|H(\omega)s(\omega) - \hat{u}_o(\omega)\|^2 + \frac{\lambda}{2} \|s(\omega)\|^2, \quad (2)$$

where $\lambda > 0$ is the regularization parameter that controls the trade-off between the fidelity to the target response and the smoothness of the solution. The l_2 -norm regularization can be interpreted as a way of minimizing the total energy in the solution. In this context, the energy of the control inputs s is quantified by

$$\frac{1}{N_\omega} \sum_{i=1}^{N_\omega} \|s(\omega_i)\|^2 = \frac{1}{N_\omega} \sum_{i=1}^{N_\omega} \sum_{k=1}^d s_k(\omega_i)^2 \quad (3)$$

where N_ω represents the total number of frequency lines. By including the regularization term, $\lambda \|s\|^2$, we penalize large values of s , which in turn, encourages solutions with smaller magnitudes. This may be useful in practical applications where excessive energy may be undesirable.

The optimization problem in equation 2 can be solved analytically for each frequency line with the following closed-form solution:

$$s^*(\omega) = (H(\omega)^T H(\omega) + \lambda \mathbb{I})^{-1} H(\omega)^T \hat{u}_o(\omega). \quad (4)$$

In this work, the regularization parameter is assumed constant across all frequencies.

Now that a control strategy is defined, we propose a source placement optimization algorithm. Consider a candidate index set $\mathcal{S} := \{1, \dots, d\}$ that corresponds to all d possible source (actuator) locations. The goal of source placement optimization to find, at most, N sources that minimizes equation 2.

To this end, we propose the following Mixed Integer Quadratic Programming (MIQP) problem:

$$\begin{aligned} \theta^*, z^*(\omega) = \min_{x, \theta, z} \quad & \frac{1}{2N_\omega} \sum_{j=1}^{N_\omega} \left(\|H(\omega_j)z(\omega_j) - \hat{u}_o(\omega_j)\|^2 + \lambda \|z(\omega_j)\|^2 \right) \\ \text{s.t. } \quad & \theta_i s_i(\omega) = z_i(\omega) \quad \text{for } i \in \mathcal{S} \\ & \theta_i \in \{0, 1\} \\ & \sum_{i=1}^d \theta_i \leq N, \end{aligned} \quad (5)$$

where θ_i is a boolean, s_i and z_i indicate the i^{th} entry of $s(\omega)$ and $z(\omega)$ respectively, and $z(\omega) \in \mathbb{C}^d$ is defined by the equality constraint.

MIQP problems are difficult to solve and computationally expensive. Hence, we propose a greedy search algorithm, which significantly reduces computational cost but at the expense of solution optimality. In the greedy search, let $\theta \in \mathcal{P} = \{i_1, \dots, i_N\} \subset \mathcal{S}$ represent the selected source indices, and let $s^*(\omega, \theta)$ and $H(\omega, \theta)$ represent the solution to the control problem and map, respectively, corresponding to the source locations indexed in θ . The greedy algorithm seeks to minimize the function $f(\theta)$, which is given by

$$f(\theta) = \frac{1}{2} \sum_{\omega} \left(\|H(\omega, \theta)s^*(\omega, \theta) - \hat{u}_o(\omega)\|^2 + \lambda \|s^*(\omega, \theta)\|^2 \right), \quad (6)$$

where $s^*(\omega, \theta)$ is the optimal control input given by equation 4 evaluated at the source locations θ . As described in Algorithm 1, we first initialize an empty set θ , then search over all candidate source locations, selecting the best source location that minimizes $f(\theta)$, and adding it to the set θ . The process is repeated until the source budget is satisfied.

In addition to the Tikhonov regularization objective function, we consider the greedy algorithm using the E-criterion. In this context, the E-criterion is taken as

$$f(\theta) := \frac{1}{N_\omega} \sum_{j=1}^{N_\omega} \sigma_{\max}(H(\omega_j, \theta)^T H(\omega_j, \theta))^{-1}. \quad (7)$$

where $\sigma_{\max}$ is the largest singular value of $(H(\omega, \theta)^T H(\omega, \theta))^{-1}$. Typically, E-optimality is defined by the maximum eigenvalue [7], however, we will use the singular values since $H^T H$ is positive semi-definite, and hence the eigenvalues and singular values are equivalent.

Algorithm 1 Greedy Algorithm

```

1:  $s^*(\omega, \theta) = (H(\omega, \theta)^T H(\omega, \theta) + \lambda \mathbb{I})^{-1} H(\omega, \theta)^T \hat{u}_o(\omega)$ 
2:  $f(\theta) = \frac{1}{2N_\omega} \sum_{j=1}^{N_\omega} \left( \|H(\omega_j, \theta) s^*(\omega_j, \theta) - \hat{u}_o(\omega_j)\|^2 + \lambda \|s^*(\omega_j, \theta)\|^2 \right)$ 
3: Initialization:  $\theta = \emptyset$  and  $M = 0$ 
4: while  $M < N$  do
5:    $\theta_j = \theta \cup \{j\}$ 
6:    $j^* = \arg \min_{j \in \{1, \dots, n\} \setminus S} f(\theta_j)$ 
7:    $\theta = \theta \cup \{j^*\}, \quad M = M + 1$ 
8: end while

```

The E-criterion is motivated by several considerations. The E-optimal design maximizes the minimum eigenvalue of $H^T H$. This can be interpreted as maximizing information by ensuring a more uniform distribution of information across all measurement directions. Additionally, maximizing the minimum eigenvalue tends to minimize the matrix condition number, which leads to more stable and reliable solutions to the inverse problem. The use of the E-criterion may also be more practical in some settings since it does not require the target response.

We also expect that E-optimal designs minimize the input energy of the least squares solution, since $\sigma_{\max}(H^T H)^{-1}$ contributes to an upper bound on the input energy. This follows directly from the definition of the least squares solution (i.e. $\lambda = 0$) and the norm inequality,

$$\begin{aligned}
 \hat{x}_{lsq} &= (H^T H)^{-1} H^T \hat{u}_o, \\
 \|\hat{x}_{lsq}\|_2 &= \|(H^T H)^{-1} H^T \hat{u}_o\|, \\
 &\leq \|(H^T H)^{-1} H^T\| \|\hat{u}_o\|, \\
 &= \sigma_{\max}((H^T H)^{-1} H^T) \|\hat{u}_o\|
 \end{aligned} \tag{8}$$

where the matrix norm is taken as the spectral norm, i.e. the maximum singular value of the matrix. Finally, we observe that minimizing $\sigma_{\max}((H^T H)^{-1} H^T)$ is equivalent to minimizing the E-criterion by the definition of the singular value decomposition:

$$\begin{aligned}
 \sigma_{\max}((H^T H)^{-1} H^T) &= 1/\sigma_{\min}(H), \\
 &= \sqrt{\sigma_{\max}(H^T H)^{-1}}.
 \end{aligned} \tag{9}$$

The final motivation for E-optimality is that the criterion is approximately supermodular [7]. Supermodularity guarantees that the greedy algorithm results in a near-optimal solution. The proof of approximate supermodularity is based on sensor placement optimization [7]. However, we can show that these results are applicable to the source placement problem by leveraging the fact that non-zero singular values of H and H^T are equivalent. We leave this derivation for future work.

Example Problem

In this work, we use a simulation of a wind tunnel test article subjected to turbulence to verify and demonstrate the proposed methods. The goal of this work is to optimally place a small set of virtual actuators to best reproduce the dynamic response generated from a wind tunnel environment.

The wind tunnel serves to reproduce turbulence-induced, fluctuating pressure loads experienced by a flight vehicle. To this end, the turbulent boundary layer and surface quantities were simulated using Sierra-SPARC's hypersonic computational fluid dynamics (CFD) code [8], and converted to a discretized pressure cross power spectral density (CPSD) on the surface of the sharp cone FEM using an extension to the Corco's model [9]. Time-domain pressures were realized from the pressure CPSD and then Fourier transformed into the frequency domain to obtain the problem form given in equation 1. The frequency domain acceleration response due to the fluctuating surface pressure was simulated using the structural FEM simulation code Sierra-SD [10]. The target acceleration responses are measured in the y and z degrees of freedom at 84 nodes as shown in Figure 1. The discretized pressure patches on the model's surface also serve as the candidate shaker (source) locations,

totaling 782 locations. Given the CFD turbulence model and the frequency range of interest, most of the pressure loading was towards the aft end of the vehicle.

The frequency range of interest was from one to three kHz. The vehicle deformation was dominated by four modes: two first order bending modes, and two second order bending modes. The first and second order bending modes are shown in Figure 2.

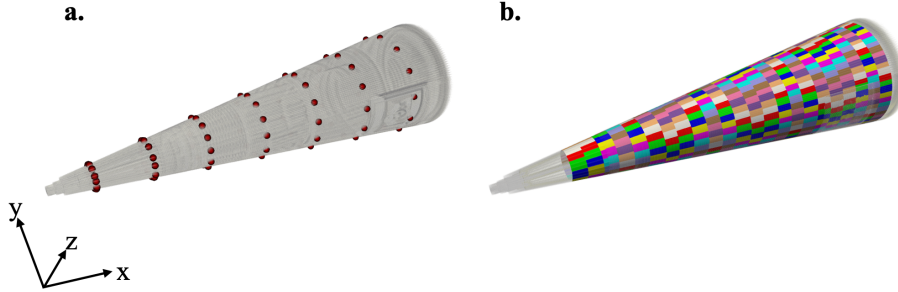


Fig. 1 a) The target response locations and b) the candidate source locations.

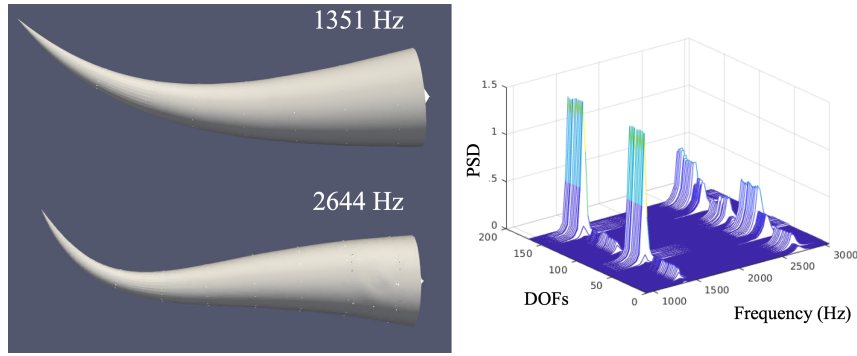


Fig. 2 Power spectral density and major resonances of structural FEM.

MIQP Greedy Search Comparison

In this section, we evaluate the quality of the greedy source placement algorithm's solution by comparing it to the optimal solution to the MIQP problem, solved using Gurobi's MIQP solver [11]. This method serves as a form of verification for the Greedy algorithm using the regularized inverse problem objective function. Gurobi uses a branch and bound technique to solve the MIQP problem [11]. For this initial study, we simplified the problem to ensure the MIQP problem could be solved in a reasonable amount of time. Specifically, the OED algorithms included only one frequency line corresponding to the first bending mode of the structural model. The map H is complex-valued with 168 rows and 100 columns, representing a small subset of all possible source locations. We set $\lambda = 10^{-3}$ in order to provide numerical stability to Gurobi's optimization algorithm. When solving the regularized inverse problem, $\lambda = 10^{-3}$ resulted in a solution close to the least squares solution.

We first verified the solution to the MIQP by comparing it to 50,000 randomly selected sources. We ran the optimization algorithm for five source budgets (one through five). We then computed the quantile of the MIQP solution with respect to randomly selected sources by counting the number of times that the MIQP solution $f(\theta_{mip}^*)$ was less than the random sensor solution $f(\theta_{random}^i)$ for $i = 1, \dots, 50,000$. The quantile is defined as:

$$quantile = \frac{1}{50,000} \sum_{i=1}^{50,000} \mathcal{B}(f(\theta_{mip}^*) \leq f(\theta_{random}^i)),$$

where $\mathcal{B}$ returns one if true and zero if false. Table 1 summarizes the results for the five different budgets. For each source budget, the MIQP solution outperforms 100% of the randomly selected source combinations.

We then performed the same study using the greedy algorithm and computed its quantiles, which are also included in table 1. While the Greedy method performs well, it clearly results in suboptimal solutions, and in some case is outperformed by a brute force random search. For example, the greedy solution outperformed 90% of the randomly selected sources for a source budget of four. However, the differences between the objective $f(\theta)$ using greedy and the MIQP method were relatively small.

Table 1 Quantiles and optimal objective values for MIQP and greedy solutions

Budget	1	2	3	4	5
MIP Objective Value	0.1297	2.090e-3	3.225e-4	2.321e-4	1.517e-4
Greedy Objective Value	0.1297	4.020e-3	6.606e-4	2.825e-4	2.301e-4
MIP Quantile	1.000	1.000	1.000	1.000	1.000
Greedy Quantile	1.000	0.928	0.992	0.898	0.997

We now investigate the accuracy of the greedy search relative to the MIQP problem as a function of the regularization parameter. For this investigation, we solved individual source placement optimization problems for regularization parameters ranging from $\lambda = 10^{-3}$ to $\lambda = 10^{10}$. For each regularization parameter and the corresponding optimal source design, we solved the a control inverse problem using the same regularization parameter. The results of this study are presented as Pareto curves in Figure 3.

The Pareto curve presents the residual, $\|H(\theta^*)s^*(\theta^*) - \hat{u}_0\|^2$, as a function of the input energy, $\|s^*(\theta^*)\|^2$, where θ^* indicates optimal source design, for all regularization parameters of interest. The Pareto curve helps visualize the trade-off between the residual and the input energy. We emphasize for clarity that a source placement optimization problem was solved for each regularization parameter. Figure 3 indicates that the Greedy and MIQP algorithms exhibit similar performance in terms of the residual and input energy. For some regularization parameters, the Greedy algorithm outperformed the MIQP by a small, negligible amount. We expect this was because the MIQP did not completely converge.

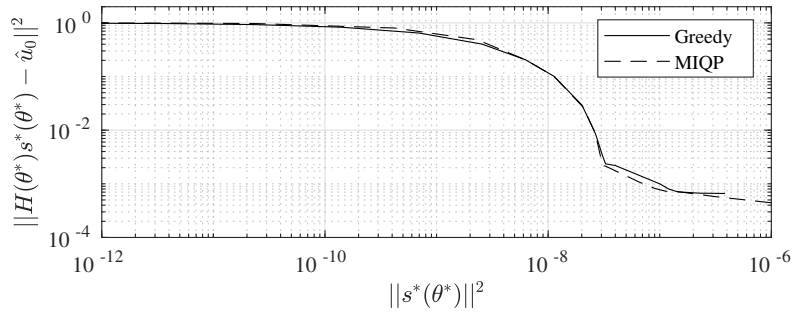


Fig. 3 Pareto curve of solutions using the greedy search algorithm and MIQP for one hundred candidate input locations with a total source budget of three.

One of the drawbacks of attempting to solve the full MIQP OED problem is the extensive runtime. While the greedy method results in suboptimal solutions, it completes in seconds. In contrast, the MIQP can take hours to solve even relatively small problems, and its computational cost increases with the source budget and the total number of candidate source locations.

In this study, we investigated the MIQP runtime as a function of the source budget and the total number of candidate source locations. We ran Gurobi on 16 CPUs with a base clock speed of 2.7 GHz. We generated a single-frequency, strictly real transfer matrix with 168 target response degrees of freedom (DoFs). We varied the source budget from one to five and the total candidate source locations from 100 to 600. Figure 4 presents the runtime in minutes for each budget on a logarithmic scale. The maximum observed runtime was approximately 300 minutes. These performance results by no means represents the optimal performance of Gurobi, only our observation using finite computing resources and the optimizer's default options.

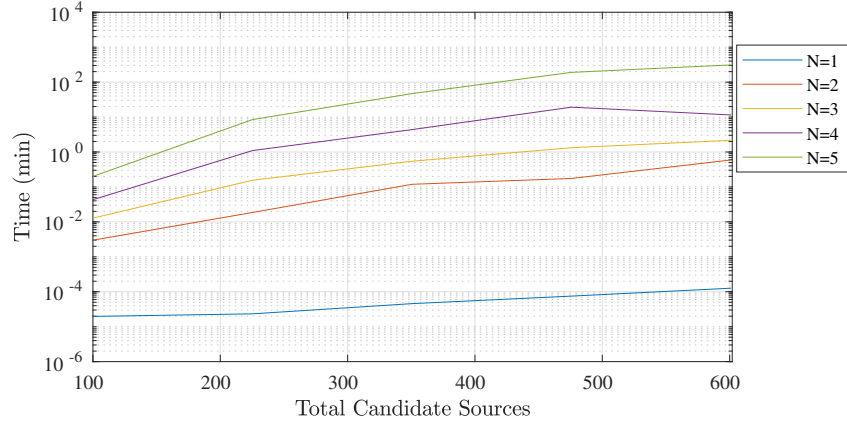


Fig. 4 MIQP solver runtime using Gurobi as a function of total candidate source locations and source budget (N).

Greedy OED Performance on Full-Scale Model

In this section, we applied the greedy source placement algorithms to the full wind tunnel model over a wide band of frequencies. The optimization algorithms include 71 frequency lines spanning one to three kHz. In this example, we set the source budget equal to 5, and include all 168 target response DoFs and all 782 candidate source locations ($H(\omega_i) \in \mathbb{C}^{168 \times 782}$).

We focus on three source configurations using the greedy algorithm corresponding to the following objective functions: i) E-optimality, ii) the regularized objective with $\lambda = 1.7 \times 10^6$, denoted *Tikhonov* $\lambda = 1.7 \times 10^6$, and iii) the regularized objective with $\lambda = 10^{-3}$, denoted *Least Squares*. Each criterion was chosen because of their unique spatial configurations as shown in Figure 5. The *Least Squares* design (Figure 5a.) favored sources near the aft end of the vehicle as this is where most of the pressure generating the target response occurs. In other words, encouraging minimum residual over minimum input energy resulted in sources physically close to the true source locations. The *Tikhonov* $\lambda = 1.7 \times 10^6$ design encouraged lower source energy, which resulted in a clustering of sources near the nose of the vehicle (Figure 5b.). The result is intuitive since we expect the largest dynamic response magnification at the nose due to the vehicle's active bending modes. Finally, the E-optimal criterion results in a design that is visually between the two previous scenarios, spreading the sources evenly across the model (Figure 5c.).

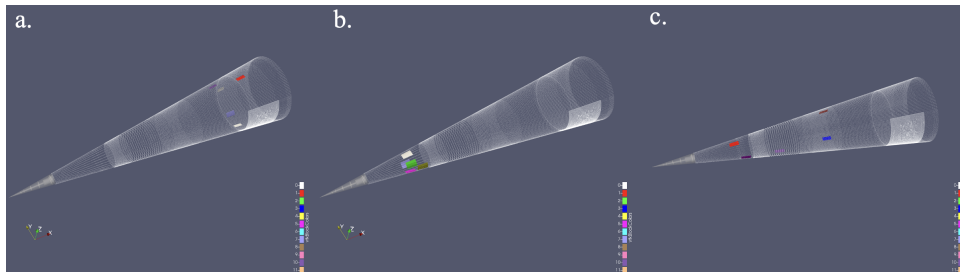


Fig. 5 a) The *Least Squares* OED solution, b) the OED solution using Tikhonov regularization with $\lambda = 1.7 \times 10^6$, and c) the OED solution using E-optimality.

We first confirmed that the Greedy algorithm resulted in good solutions relative to random source placement. For each of the three optimality criteria, we drew 5,000 unique random combinations of 5 source locations and evaluated the corresponding OED objective function for each sample. Figure 6 presents the histograms of the three objective functions evaluated at the random samples alongside the solution to the OED. In each of the three cases, the greedy search resulted in better solutions than the randomized search.

In the next studies, we investigate the design performance of each of the three solutions by plotting their Pareto curves. To generate a Pareto curve, we used the optimal designs from the three cases: i) *Least Squares*, *Tikhonov* $\lambda = 1.7 \times 10^6$, and *E-Optimal*, and solved the regularized inverse problem for all values of λ . In these studies, λ ranged from 10^{-3} to 10^{10} . The

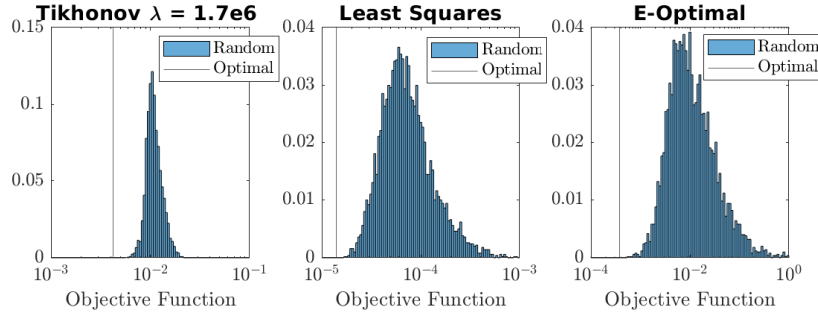


Fig. 6 a) OED solution using least squares formulation. b) OED solution using Tikhonov regularization with $\lambda = 1.7 \times 10^6$. c) OED solution using E-criterion..

Pareto curve then plots the residual, taken as,

$$Residual = \frac{1}{N_\omega} \sum_{i=1}^{N_\omega} \|H(\omega_i)s^*(\omega_i) - \hat{u}_o(\omega_i)\|^2,$$

as a function of input energy, taken as,

$$Input\ Energy = \frac{1}{N_\omega} \sum_{i=1}^{N_\omega} \|s^*(\omega_i)\|^2.$$

The Pareto curves are shown in Figure 7 along a reference Pareto curve corresponding to the optimal design as a function of λ denoted *Tikhonov All λ* . The reference curve was obtained by solving the l_2 -norm regularized source placement problem and the regularized inverse problem with the same value of λ .

One of the take-aways of this work is that the optimal source locations vary with the choice of λ . For instance, the source locations optimized with $\lambda = 10^{-2}$ do not necessarily yield optimal results when the control inverse problem is solved with $\lambda = 10^3$. We observe this behavior by comparing the reference Pareto curve (*Tikhonov All λ*) to the other fixed design cases, and note the improvement in the solution to the inverse problem in terms of fidelity to the target response and input energy when λ is consistent between the inverse problem and the source placement algorithm.

We also use Figure 7 to interpret how robust the optimized source design is to changes in the regularization parameter. For context, there are likely cases in practice that the source configuration is first chosen, but then the regularization parameter used to solve the inverse problem is adjusted to improve the control performance. Inspecting Figure 7, we observe that the *Least Squares* design results in a smaller residual than the E-optimal design and *Tikhonov $\lambda = 1.7 \times 10^6$* . If we are interested in control solutions that both minimize input energy and residual then we would likely prefer the *Tikhonov $\lambda = 1.7 \times 10^6$* design. However, this design's fidelity to the target response is limited as indicated by the flat curve plateau. Based on the Pareto curve and physical positions on the vehicle, the *E-Optimal* design tends to strike a balance between the two extreme designs.

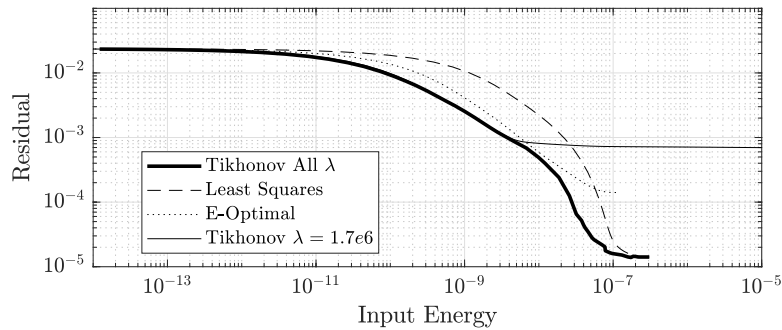


Fig. 7 Pareto curve for three fixed greedy-based designs: *Least Squares* (dashed), *Tikhonov $\lambda = 1.7 \times 10^6$* (solid), and *E-Optimality* (dotted). *Tikhonov All λ* (thick solid) plots the residual and input energy corresponding to multiple source designs obtained from the regularized objective where each design is optimized for each λ .

Conclusion

We conclude this work with the following observations. The greedy algorithm performs well compared to the MIQP method, but is not hindered by computational runtime. This allowed us to solve larger problems in terms of number of frequency lines, candidate input locations, and total source budget with minimal computational effort. We also demonstrated the advantages in choosing the regularization parameter in the source placement optimization algorithm in a manner consistent with the inverse control problem. Finally, we conclude that each of the designs studied in this chapter had a notable trade-off between the fidelity to the target response and the magnitude of the inputs. From the practitioner point of view, these trade-offs should be investigated prior to assembling the test and weighed according to the total source budget and physical energy constraints of the equipment.

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Chapter 12

Design and Testing of Vibroacoustic Metamaterials for Active Damping of Traffic Noises

W. Kaal, M. M. Becker, L. S. Kollmannsperger, and S. C. L. Fischer

Abstract Metamaterials take advantage of internal architecture to achieve functional integration and targeted property changes. Vibroacoustic metamaterials (VAMM), a sub-class of metamaterials, enable interaction with acoustic waves in different applications, inducing damping of specific frequency regimes through their geometrical architecture. To increase the functionality and flexibility of those materials, methods to actively trigger and shift regions of high damping are being developed.

In the present paper, recent advances in the design and characterization of active vibroacoustic metamaterial unit cells for traffic noise reduction will be presented, for example for noise protection in urban areas in cities or next to highways. According to literature, the noise spectrum of traffic noise spans the range of the hearable spectrum and varies according to the type of vehicle, their speed, their tires, the current road conditions and the weather. The two main influencing factors are the tire-road friction and the noise of the drive train. While there are efficient damping solutions for high frequencies, especially frequencies below 2 kHz are challenging to tackle with conventional materials. Additionally, the needed damping region may change according to the above factors and therefor active adaptation will improve the efficiency of damping.

For this reason, this study focuses on the design of two programmable VAMM (PVAMM), one whose range of action is tunable a-priori (passive VAMM) and another requiring an external trigger to be applied, in this case a change in the internal pressure (active VAMM). For the active VAMM, wood, a renewable raw material, is to be used as the base material. In order to fully understand the potential of the VAMM, experimental characterization was performed regarding vibroacoustic properties. For this purpose, the samples in this study are excited with an electrodynamic shaker and the structural response is recorded. The research aims to provide new concepts for metamaterials for noise reduction applications which can be scaled from laboratory experiments to industrial application.

Keywords Acoustic · Metamaterial · Damping · Noise · Testing

Introduction

In times when safety, comfort and environmental requirements are of equally great importance for the design of novel material solutions, engineers are faced with the challenge of developing components that meet these conflicting requirements. In the area of noise reduction, there are many scientific studies focusing on adverse effects of noise on human health [1, 2], showcasing the need for innovative material solutions for its mitigation.

Traffic noise is encountered in many different situations with varying loudness, exposure time and frequency spectrum: As a pedestrian on the side of the road, in a vehicle or for example in offices, schools or residential buildings that are in proximity to major roads. Every situation requires different approaches to mitigate or reduce the noise. Therefore, there is a

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great need for application-specific, customized solutions for different noise protecting applications such as noise in building and shielding residential areas from traffic noise (see Figure 1).

Traffic noise is a superposition of many individual sound emitters. The sound spectrum perceived is influenced, among many other factors, by the type of vehicles present, the number of vehicles and their speed, the distance to the road, the road type and its humidity [3, 4]. Generally, on dry asphalt roads, the peak intensity of traffic noise is in the frequency range between 500 and 1000 Hz, with buses and trucks contributing more to the lower frequencies and passenger vehicles to the higher frequencies [3]. However, presence of water on the road due to rainfall and other factors can shift the frequency spectrum quite drastically, e.g. with maximum intensities shifted from 1 kHz to around 3 kHz in the presence of water on a road as shown by Cai et al. [5]

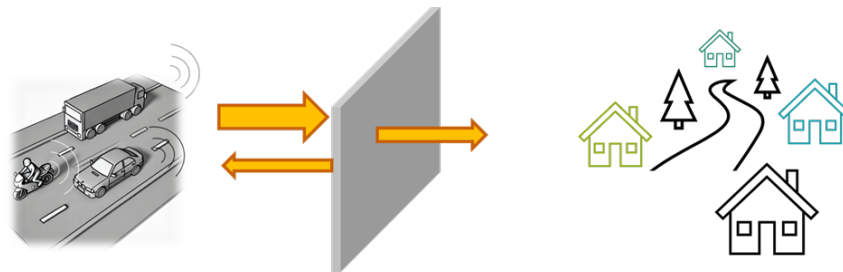


Fig. 1 Schematic representation of the requirements of noise attenuation materials

The simplest method of reducing noise and structural vibrations is to select suitable materials and components as large and heavy as possible. However, this is neither economically sensible nor ecologically sustainable. Vibroacoustic metamaterials (VAMM), a subclass of metamaterials [6], influence structure-borne sound propagation and sound radiation in a specified frequency range through annihilating interference and negative stiffnesses or masses [7]. They provide one approach to optimize acoustic properties through engineering of the geometry while minimizing the use of resources. These consist either of an array of distributed mass elements (Bragg effect) or of a basic structure with periodic local resonators [8]. Typical resonator designs are membrane-like structures [9], bending beams or columns [10]. The interaction of the structural vibration with the resonance vibration of the local resonators leads to specific stop bands with a high suppression of wave propagation [8]. The position and width of the stop bands in the frequency range can be influenced by the size of the unit cell incorporating the resonators, the effective resonator mass, the materials used and the structural properties, which allows great flexibility for application-specific customization.

Based on literature, it can be seen that a large number of VAMMs have already been developed [7,11–14]. These are generally designed as passive elements for a specific application so that a stop band occurs in a desired frequency range. However, it is known from the literature that the acoustic emission and the structural vibration of traffic noise are strongly dependent on external factors such as the type of vehicles, speed and atmospheric conditions. Active VAMM with tunable band gaps could provide an interesting new approach to integrate more functionality within a single material system. In previous works, the authors have demonstrated different passive and active VAMM concepts (see Figure 2). Kaal et al. [13] were able to demonstrate the variable damping behavior of a mechanical metamaterial under dynamic load. The cell showed lower damping at lower strains, whereas the damping increased significantly at higher strains. In addition, Kollmannsperger et al. [15] showed a concept for an inflatable VAMM unit cell with tunable band gaps based on applied air pressure.

This study focuses on the comparison of two designs of a programmable VAMM (PVAMM), one with a passive and one with an active noise damping integrated. First, the design of the two unit cells is presented followed by the characterization of the acoustic damping effects based on mechanical excitation in the frequency range relevant for traffic noise mitigation. For more detailed information and further characterization results on the individual unit cells, we refer the reader to our original publications [14, 15]. Finally, both designs will be discussed in terms of advantages and disadvantages to consider for application in traffic noise mitigation.

Materials and Methods

Vibroacoustic metamaterials can be designed in various different ways. To showcase the potential for application for traffic noise reduction, two concepts are presented and characterized in this paper (see Figure 3):

- An active, inflatable VAMM manufactured from wood and silicone sheets and including a steel mass
- A passive VAMM manufactured by laser-powder-bed fusion with integrated powder

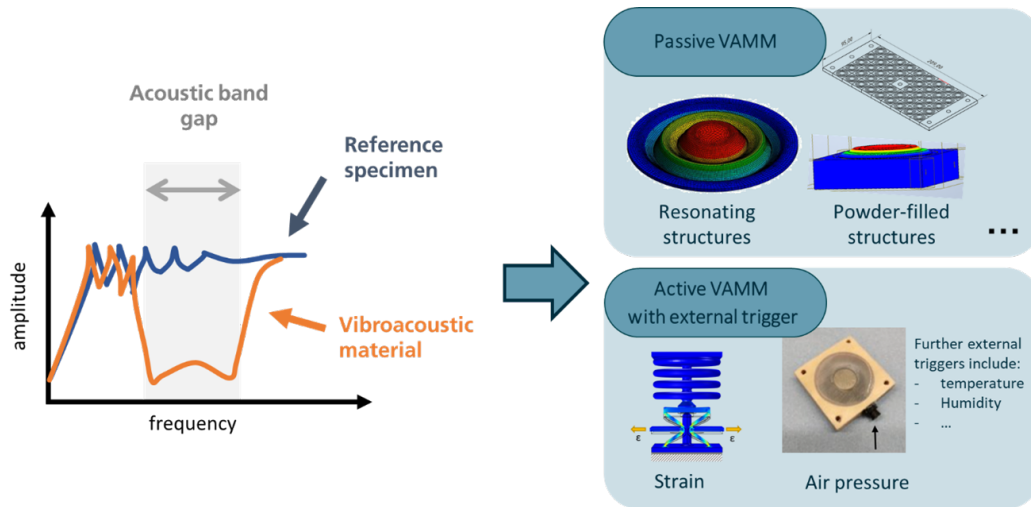


Fig. 2 Vibroacoustic metamaterials: Properties and classification of PVAMM

Unit cell design 1: Active VAMM

In order to demonstrate the usability of wood as a renewable raw material in the field of vibroacoustic metamaterials and the adjustability of damping ranges using an external trigger, a demonstrator unit cell is being constructed. The basic structure of this cell consists of a 9 mm thick wooden panel with a recess in the middle. A conical steel mass weighing 27.9 g is inserted concentrically into this recess. The individual components are bonded together with a 1 mm thick silicone mat. The additional attachment of a valve, which is directly connected to the recess, allows the pressure within the unit cell to be regulated. A more detailed overview of the unit cell and further measurements can be found in a proceedings publication by Kollmannsperger et al. [15].

Unit cell design 2: Passive VAMM

In order to demonstrate a passive monolithic VAMM concept, an array of resonant cavities was designed. The cavities can additionally be filled with powder to enhance the damping through powder motion and frictional effects. The passive vibroacoustic plates were fabricated by selective laser sintering (EOSINT P395 system by EOS GmbH, Krailing, Germany) with polyamide powder (PA 2200). Three plates of identical outer dimensions of 210 mm × 95 mm × 5 mm were fabricated: A fully sintered plate, a plate with empty resonators and a plate with powder-filled resonators. The two later resonators comprise 44 local resonators with a spacing 19 mm. A more detailed overview of the unit cell and further measurements can be found in a publication by Kollmannsperger et al. [14].

Experimental characterization

The acoustic properties with regard to sound transmission and the vibromechanical properties of the two VAMM were investigated experimentally using an electrodynamic shaker (vib TV52122-M, TIRA GmbH, Schalkau, Germany). The VAMM specimen were excited with broadband white noise in two frequency ranges, notably between 0 - 300 Hz for the active VAMM and 0 - 5000 Hz for the passive VAMM. For the active VAMM, the internal pressure was varied between 0 and 0.75 bar using a hand driven airpump.

Analysis

Experimental validation of the active VAMM

By evaluating the force and acceleration signals in the frequency range, the characteristic transfer function of the cell was determined for each internal pressure (see Figure 4). A shift of the maximum and minimum of the measurement curves to higher frequencies with increasing internal pressure of the cell can be recognized both from the observation of the amplitude and from the observation of the phase. This shift is directly accompanied by a shift of the stop band to higher frequencies. The sharpness of the transition from the maximum to the minimum of the amplitude and the depth and width of the negative peak

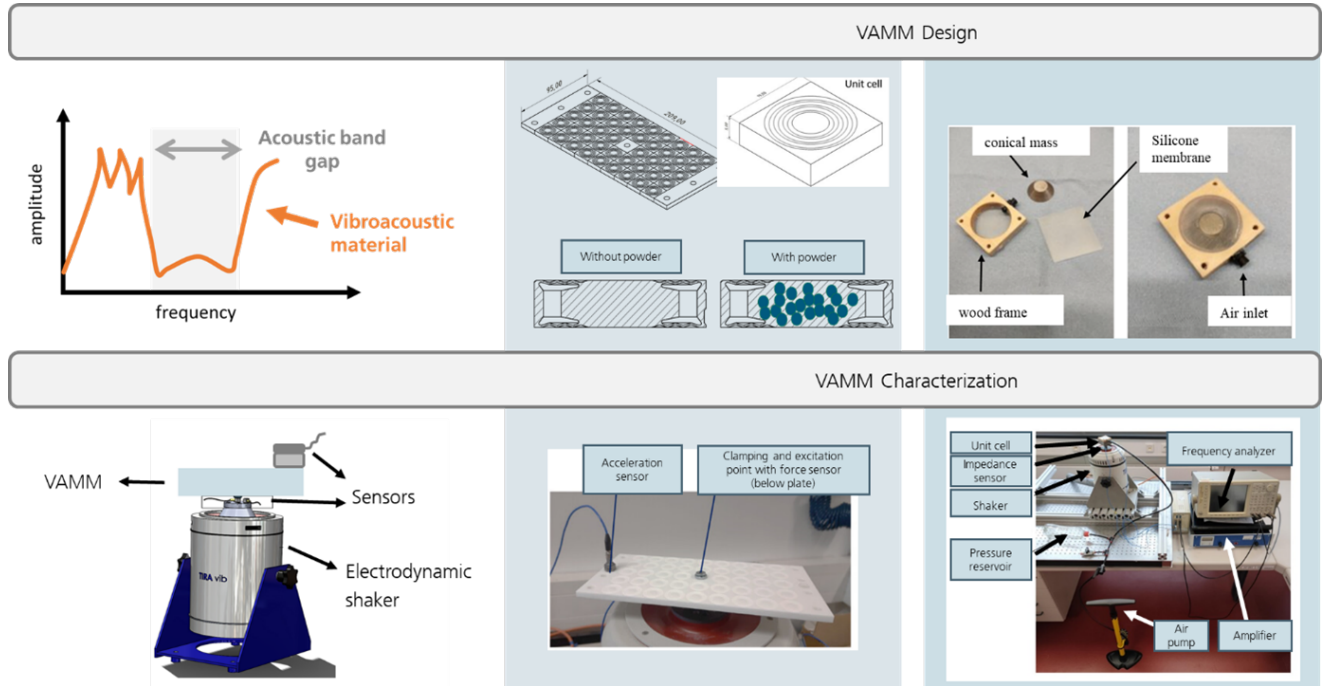


Fig. 3 Design and characterization of two VAMM unit cells: Passive unit cell manufactured by laser powder bed fusion with embedded powder cavities and active inflatable wood-silicone composite with tuneable properties by air pressure

in the observation of the phase can already be used to qualitatively recognize the attenuation properties. The more blurred the transition from the maximum to the minimum of the amplitude of the transfer function and the flatter the negative peak of the phase, the greater the effective attenuation of the PVAMM.

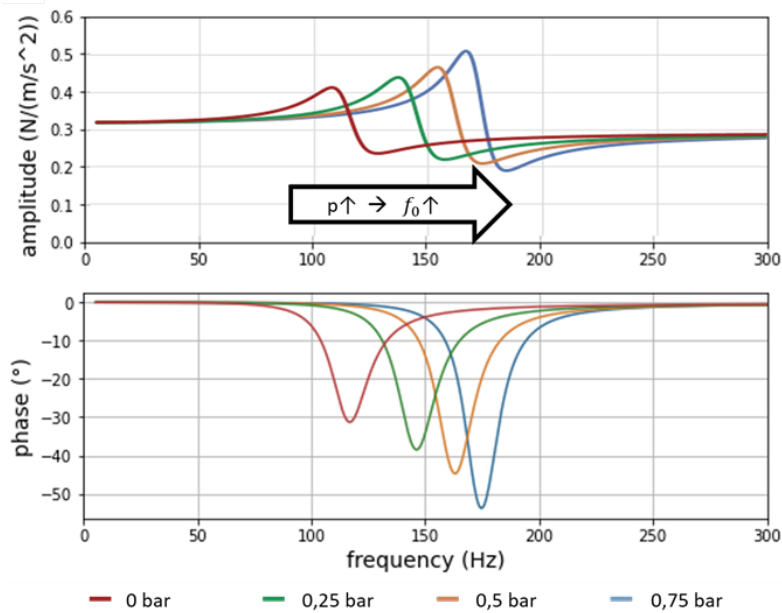


Fig. 4 Experimental characterization of the active VAMM unit cell as a function of air pressure. A) Amplitude as a function of frequency b) Phase as a function of frequency. The figure is adapted from [15].

The experimentally determined transfer function of the unit cell was used for comparison with an analytical four-parameter model (see Figure 5). This model consists of a spring with stiffness k , a damper with damping constant d and two masses (m_1 and m_2).

In this way, the effective stiffness k and the effective dimensionless damping ratio D of the unit cell could be quantitatively determined as physical parameters for each measurement. Finally, a value for the stiffness and damping can be assigned to each internal pressure using suitable interpolation. Figure 5 shows that the effective damping decreases with increasing internal pressure from 8 % at 0 bar to 4 % at 0.75 bar, whereas the effective stiffness increases from 13 Nmm^{-1} to 32 Nmm^{-1} . By comparing the measured values with the analytical model, the frequency shift can also be quantified, so that an increase in the internal pressure by 0.75 bar results in a frequency shift of 57 Hz.

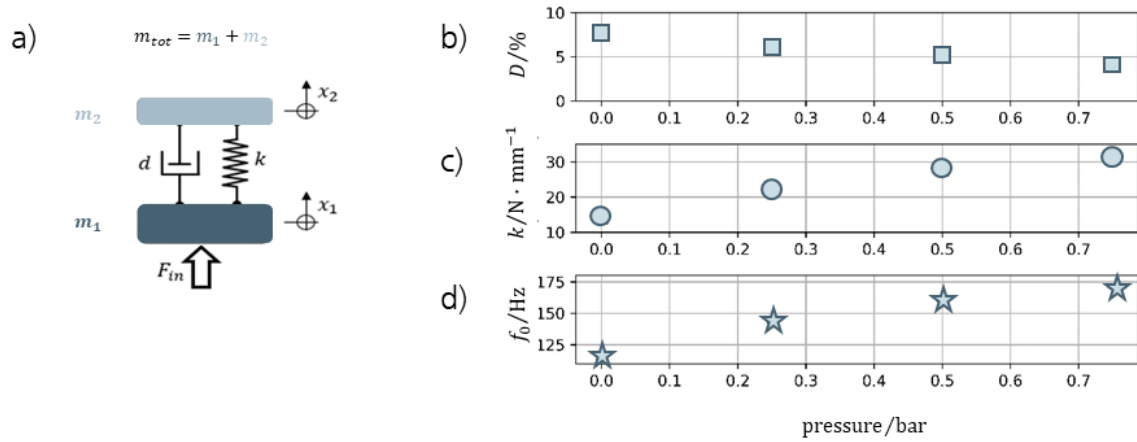


Fig. 5 Analytical description and modelling of the active VAMM unit cell. a) Mathematical model based on two masses, a spring and a damper. Determination of b) the effective damping, c) effective stiffness and d) resonant frequency based on the analytical model. The figure is adapted from [15].

Experimental validation of the passive VAMM

In the following section, the vibrational properties of the passive VAMM are described (Figure 6). The plates were excited with broadband noise and their vibrational response measured. Below 700 Hz and above 3 kHz, the three samples exhibit similar response to excitation. Between 700 Hz and 3000 Hz, the VAMM plates with and without powder exhibit a band gap with an average damping of the amplitude by 20 dB. While the powder-filled VAMM plate exhibits maximum acoustic damping in this frequency range, it can be observed that the air-filled plate actually has a wider band gap and more pronounced attenuation at higher frequencies starting from around 2200 Hz.

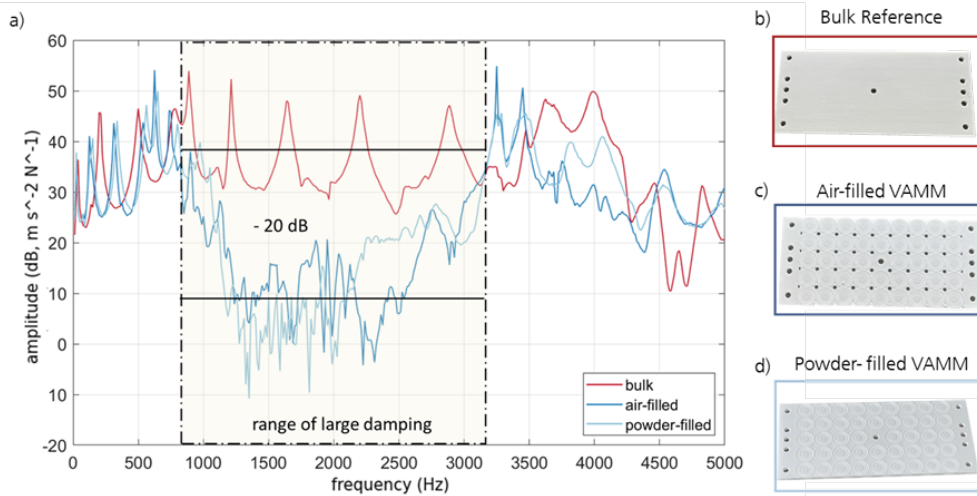


Fig. 6 Experimental characterization of the passive VAMM. The figure was adapted from Kollmannsperger et al. [14] with permission from Wiley.

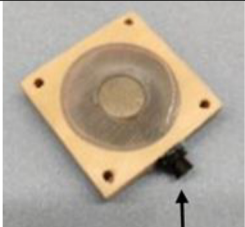
Discussion of application of passive and active VAMM in traffic noise control

Vibroacoustic metamaterials offer promising advantages for traffic noise mitigation. One key advantage is their ability to achieve superior sound attenuation compared to traditional materials, particularly in specific frequency ranges like low-frequency noise, which is a significant component of traffic noise. They can be designed to target and reduce particular noise frequencies while maintaining a lightweight structure, making them more efficient and adaptable. Additionally, these materials can be tailored for specific environmental and structural requirements and tunability, providing flexibility in design and application.

However, there are also disadvantages to consider. Vibroacoustic metamaterials can be more complex and costly to produce compared to conventional noise barriers, which may limit their widespread adoption. The performance of metamaterials may also be sensitive to environmental factors, such as temperature or wear over time, potentially reducing their effectiveness in long-term outdoor applications. Moreover, the current stage of research and development may mean that extensive testing is still needed before these materials can be broadly implemented for traffic noise mitigation.

As a summary, the two studied metamaterial types are compared side to side in Table 1 concerning the studied acoustic band gap, advantages and disadvantages in the application for traffic noise mitigation.

Table 1 Comparison of the active and passive VAMM concept for application in the area of traffic noise reduction

	Active VAMM concept	Passive VAMM concept
Design		
Band gap	- Band gap tunable - Range from 125 to 180 Hz	- Band gap preprogrammed - Range from 750 Hz to 3 kHz
	- Tunable band gap by air pressure - Changing mass and membrane enables tuning of VAMM unit cell without geometrical change - Production can be scaled - Enables layerwise production techniques - Use of wood as base material successful	- No infrastructure needed - Good recyclability due to use of only one single feedstock; possibly the VAMM could be reused to manufacture powder again - Lowered chance of local production defects due to one-material-build
	- Air pressure providing infrastructure needed - Energy needed to operate the active switch of properties - VAMM complex to recycle - High complexity of the multi-dimensional design-space complicates the computer aided development	- Preprogrammed acoustic damping band gaps - Production of the VAMM requires LPBF machine and handling of powder - Production is relatively slow - Limited design-space due to small amount of possible materials and so limited space of specific density

Conclusion

The approach shows that it is possible to design VAMM concepts with stop bands in the frequency range relevant for traffic noise and the active VAMM concept can be adapted to changing environmental conditions using external triggers. In the future, the unit cells can further be optimized by experimental and numerical methods. Larger scale prototypes will enable

the effect of the material systems in application like - scenarios. Both passive and active VAMM concepts can be scaled to different applications, however, the specific requirements regarding present infrastructure and energy consumption will then determine which concepts are most viable.

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Chapter 13

pyFBS: A Python Package for Frequency Based Substructuring and Transfer Path Analysis

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Abstract pyFBS is a Python package for Frequency Based Substructuring, Transfer Path Analysis, multi-reference modal identification, and a variety of different expansion techniques. It enables the user to use state-of-the-art dynamic substructuring methodologies in an intuitive manner. With the package, basic and application examples are also provided, together with real datasets, so you can directly try out the capabilities of pyFBS. It is currently being used by several undergraduate students and postgraduate researchers, but also offers great potential in various industrial projects. The pyFBS team plans to share the most advanced tools and methods in the field of dynamic substructuring and transfer path analysis. The goal is to create a common working base for scientists interested in this topic. Within this paper, the use of transfer path analysis within the pyFBS package is introduced. The package facilitates several approaches to model the transfer paths through the interface between the active and passive side of the assembly. This enables the indirect characterization of the source, commonly by estimating a set of equivalent forces that replicate the responses at the passive side of the assembly and lead to the identification of the critical transfer paths in terms of noise and vibration transfer. An example on a numerical case study is provided, demonstrating the efficiency and applicability of pyFBS in executing these tasks.

Keywords Python · Open Source · Transfer Path Analysis · Source Characterization · Numerical Case Study

Introduction

Transfer Path Analysis (TPA) is a useful engineering tool used to identify noise and vibration transfer paths traveling from the source substructure of an assembly to the connected passive substructures [1, 2]. TPA quantifies actively vibrating components, where vibrating mechanism are usually difficult to model or measure directly, and vibration transfer to connected passive substructures. By identifying the key contributors to noise and vibration, engineers can propose targeted solutions to reduce these unwanted effects.

There are conceptually different families of TPA methods with variations in their implementations [2]. Component-based TPA characterizes the source excitations by a set of equivalent (also known as blocked) forces, which are an inherent property of the active component. Equivalent forces are thus transferable to assemblies with modified passive sides, enabling component optimization of the assembly. Directly measured equivalent forces assume an infinitely rigid ground to which the source is mounted using force transducers. Load, measured during operation of the source, directly represents the blocked force set. Alternatively, an indirect approaches can be applied, where equivalent forces are estimated from responses measured near/at the interface and a separate admittance measurement of the structure. From the inverse problem, equivalent

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force set is calculated that causes the same responses on the passive side when the source is turned off. Special attention is given to the *in-situ* approach [3] even eliminating the need to dismount any part of the assembly to obtain equivalent force set.

An open-source implementation of the TPA techniques is pyFBS [4], a python package for frequency based substructuring and transfer path analysis. Additionally, in recent years, pyFBS has developed into a compact and versatile package that expanded beyond FBS and TPA techniques only. Most recent additions to the package include various expansion techniques [5] and multi-reference modal identification tool [6]. This paper however, the use of transfer path analysis within the pyFBS package is introduced. First, short theoretical background on *in-situ* TPA is given, followed by a commonly adopted interface model using virtual point transformation [7]. Usage of the package is then demonstrated using numerical case study.

Theoretical background and notation

Consider a linear and time-invariant assembly of substructures A and B, coupled at the interface (Fig. 1a). Substructure A is an active component with the operational excitation $\mathbf{f}_1$. Meanwhile, no excitation force is acting on the passive substructure B. The responses on B are observed in three different sets of DoFs: at the interface DoFs ($\mathbf{u}_2$), in the proximity of the interface at the indicator DoFs ($\mathbf{u}_4$), and away from the interface at the target DoFs ($\mathbf{u}_3$).

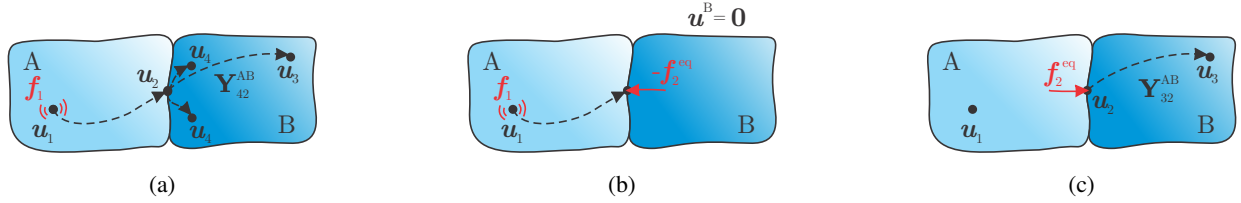


Fig. 1 *In-situ* TPA: **a)** assembly of substructures A and B, **b)** $\mathbf{f}_2^{\text{eq}}$ blocking the motion at the interface, **c)** replicating operational responses with $\mathbf{f}_2^{\text{eq}}$.

Source excitations $\mathbf{f}_1$ are often not measurable in practice; therefore, *in-situ* TPA adopts a different approach for describing the operational excitations. A set of equivalent forces $\mathbf{f}_2^{\text{eq}}$ is introduced, applied at the interface DoFs. If the source is deactivated, $\mathbf{f}_2^{\text{eq}}$ yields the same responses on the passive side as $\mathbf{f}_1$ ¹:

$$\mathbf{0} = \underbrace{\mathbf{Y}_{41}^{\text{AB}} \mathbf{f}_1}_{\mathbf{u}_4} + \mathbf{Y}_{42}^{\text{AB}} (-\mathbf{f}_2^{\text{eq}}). \quad (1)$$

Expressing the equivalent forces $\mathbf{f}_2^{\text{eq}}$ from the indicator responses $\mathbf{u}_4$ yields:

$$\mathbf{f}_2^{\text{eq}} = (\mathbf{Y}_{42}^{\text{AB}})^+ \mathbf{u}_4. \quad (2)$$

Receiver operational responses for the arbitrary assembly (with the same source) can be replicated by the set of equivalent forces (Fig. 1c) as follows:

$$\mathbf{u}_3 = \mathbf{Y}_{32}^{\text{AB}} \mathbf{f}_2^{\text{eq}}. \quad (3)$$

Interface modeling is critical to prevent redundancy/bad conditioning or neglecting important transfer paths through the interface. This means the experimentalist must account for all significant DoFs at the interface. Recently, virtual point transformation (VPT) has been proposed to model the interface where interface DoFs (rigid or flexible) are selected manually by the experimentalist to avoid redundancy yet retain the full controllability of the interface.

The loads $\mathbf{m}$ at the virtual point (VP) are obtained for a given vector of forces $\mathbf{f}$ in the proximity of the VP (Fig. 2).

The contribution from all the input forces can be combined and expressed as follows:

$$\mathbf{m} = \mathbf{R}_f^T \mathbf{f}, \quad (4)$$

where the IDM matrix $\mathbf{R}_f^T \in \mathbb{R}^{m \times n_f}$ contains the positions and orientations for all the excitation locations with respect to the VP (Fig. 2). The inverse relationship of Eq. (4) is derived with a constrained minimization for forces:

$$\mathbf{f} = \mathbf{R}_f (\mathbf{R}_f^T \mathbf{R}_f)^{-1} \mathbf{m} = \mathbf{T}_f^T \mathbf{m} \Rightarrow \mathbf{T}_f^T = \mathbf{R}_f (\mathbf{R}_f^T \mathbf{R}_f)^{-1}. \quad (5)$$

¹An explicit dependency on the frequency is omitted to improve the readability of the notation, as will be the case for the remainder of the paper.

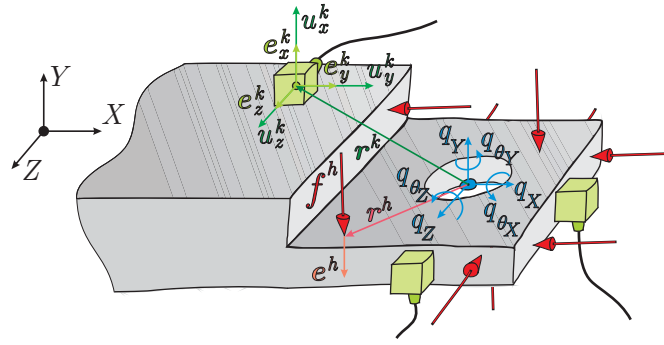


Fig. 2 Projection of responses on the k -th triaxial accelerometer and the h -th excitation onto the virtual point².

Transfer path admittance in Eqs. (2) and (3) can be expressed as follows with regards to the VPT:

$$\mathbf{Y}_{42}^{AB} = \mathbf{Y}_{4f}^{AB} \mathbf{T}_f^T, \quad (6)$$

$$\mathbf{Y}_{32}^{AB} = \mathbf{Y}_{3f}^{AB} \mathbf{T}_f^T. \quad (7)$$

Numerical case study

This numerical case study demonstrates how one can perform *in-situ* TPA using pyFBS. Within this example, the use of VPT is adopted to model the interface between the source and the receiver.

First, a 3D viewer within pyFBS is initialized, which is intended for the user to plan his experiment in virtual environment. *stl* model of the examined structure is displayed, alongside interactively positioned impacts and sensors for the purpose of obtaining admittance of the transfer paths:

```
# 3D view
view3D = pyFBS.view3D()

A = view3D.add_stl(stl_dir_A, color = "#83afd2", name = "A");
B = view3D.add_stl(stl_dir_B, name = "B");

# show sensors, channels and impacts from the dataframe
view3D.show_acc(df_acc_AB)
view3D.show_imp(df_imp_AB)
view3D.show_chn(df_chn_AB)
```

Listing 1 Initialize 3D viewer.

Full 3D display obtained using the code above is presented in Fig. 3.

The pyFBS package enables user-friendly modal analysis and FRF synthetization based on the mass and stiffness matrices imported from the FEM software. Currently, only data import from Ansys is supported. A numerical model of the previously showed AB assembly is imported, on which *in-situ* TPA will be showcased:

```
# MK model
MK_AB = pyFBS.MK_model('file.rst', 'file.full', no_modes=100, allow_pickle=True,
    recalculate=False, read_rst=True, scale=1)
```

Listing 2 Initialize MK model.

Next, locations of channels and impacts should be snapped to the nearest FE node, at which FRF will be evaluated later:

The position vector from the VP to the center of the sensor is denoted by $\mathbf{r}^k$. The unit vector for each accelerometer axis is $\mathbf{e}_i^k$ and the response in each axis is denoted by u_i^k ($i \in (x, y, z)$). The position vector from VP to the force impact is $\mathbf{r}^h$, the impact direction is $\mathbf{e}^h$ and the impact magnitude is f^h .

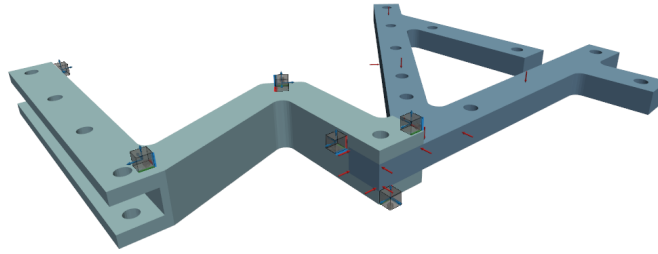


Fig. 3 3D viewer.

```
# Update locations of channels and impacts to snap to the nearest FE node.
df_chn_AB_up = MK_AB.update_locations_df(df_chn_AB)
df_imp_AB_up = MK_AB.update_locations_df(df_imp_AB)
```

Listing 3 Updating impact and channel locations.

And finally, for the numerical model, FRFs from the inputs at the impact locations/directions and outputs at the channel locations/directions are calculated using mode superposition method:

```
# Perform the FRF sythetization for each component based on the updated locations.
MK_AB.FRF_synth(df_chn_AB_up, df_imp_AB_up, f_start=0, f_end=1000, f_resolution=1,
    modal_damping=0.003, frf_type = "accelerance")

freq = MK_AB.freq
FRF = MK_AB.FRF
```

Listing 4 FRF synthetization.

For the VPT, positional data is required for channels, impacts and for virtual points. Note that only transformation of forces is required for successful identification of equivalent forces using in-situ TPA, but in order to define the VPT object in pyFBS, VP for displacements must also be defined (but neglected latter in the transformation):

```
# Prepare the reduction matrices for the VPT.
df_vp = pd.read_excel(pos_xlsx, sheet_name='VP_Channels')
df_vpref = pd.read_excel(pos_xlsx, sheet_name='VP_RefChannels')

vpt_AB = pyFBS.VPT(df_chn_AB_up, df_imp_AB_up, df_vp, df_vpref)

# Apply VPT
Y_um = FRF @ vpt_AB.Tf
```

Listing 5 Virtual point transformation.

With the admittance of the transfer paths, only operational responses must be imported. Source is then characterized in the following manner:

```
# Calculate blocked forces at the interface
f_eq = np.linalg.pinv(Y_um[:, :9, :6]) @ u4_op

# On-board validation
_Y_temp = Y_um[:, 9:, :6]
u3_tpa = _Y_temp @ f_eq
```

Listing 6 Obtaining equivalent force set and on-board validation.

The latter line already gives us on-board validation, responses at the target DoFs predicted directly from the obtained equivalent force set, when the source is turned off. Agreement between actual measured and predicted response for the given example is shown in Fig. 4.

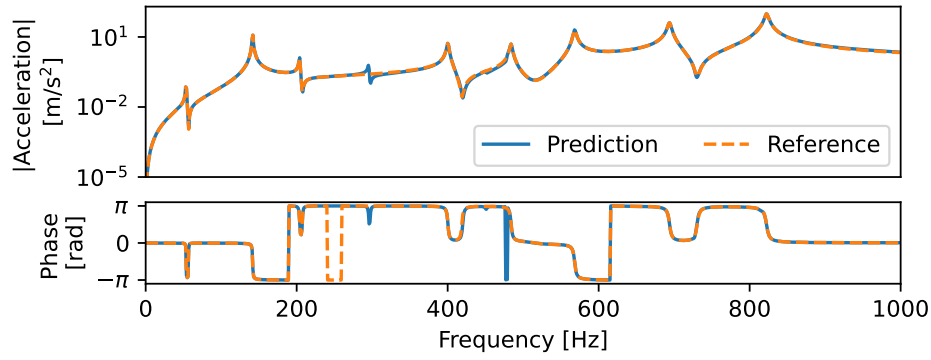


Fig. 4 On-board validation.

We can also calculate and evaluate partial transfer paths (Fig. 5):

```
# Partial transfer path can be calculated and evaluated.
sel_i = 1
u_partial = []

for j in range(6):
    gg = _Y_temp[:, sel_i:sel_i+1, j:j+1]@f_eq[:, j:j+1, 0:1]
    u_partial.append(gg[:, 0, 0])

u_partial = np.asarray(u_partial).T
```

Listing 7 Partial transfer paths.

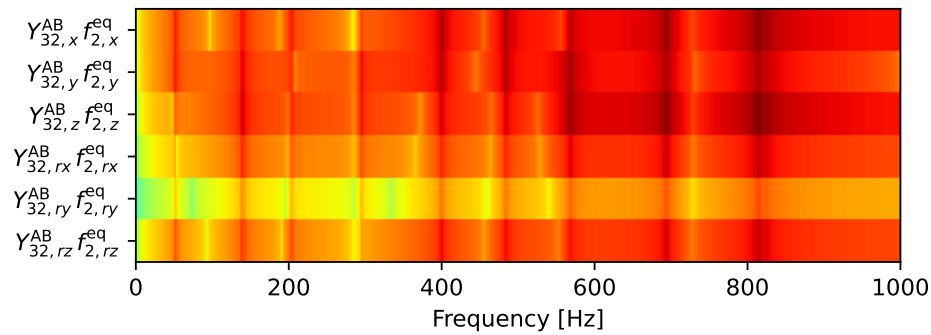


Fig. 5 Partial transfer path contribution to the full response of the target DoF.

The predicted response or each path contribution can be evaluated by listening to synthesized time response. Within pyFBS auralization can be performed with ease and the response can be played within Jupyter notebook with single line of code:

```
# Auralization of the operational response
import IPython.display as ipd

xt, s_op = pyFBS.auralization(freq, u3_tpa[:, sel_out, 0])
ipd.Audio(s_op, rate=1/(xt[1]-xt[0]))
```

Listing 8 Auralization.

Conclusions

The pyFBS is intended as an open-source Python package for Frequency Based Substructuring and Transfer Path Analysis. The development of pyFBS is ongoing and is at present used as a research tool. Current status of the package enables user-friendly implementation of the Transfer Path Analysis methods, with the *in-situ* TPA showcased within this paper. For templates on other TPA approaches, describing the source either using equivalent or interface force set, an interested reader is referred to www.pyfbs.com, where they are elaborated in detail.

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Chapter 14

Non-contact Video-based Heart Rate Monitoring using Hilbert Motion Magnification



Camilly S. Costa, Gustavo N. Castro, Yuri S. A. Alencar, Victor H. R. Cardoso, João C. W. A. Costa and Moisés F. Silva

Abstract Heart rate measurement is vital for health monitoring and medical diagnosis. Although traditional methods like oximeters and electrocardiograms are precise, they have limitations such as the need for physical contact, high costs, and operational complexity. Therefore, this study explores a non-invasive, cost-effective alternative: measuring subtle head movements caused by the Newtonian blood flow reaction from heartbeats captured via video measurements. The process involves applying the Hilbert transform to obtain the analytical signal, from which we extract its real and imaginary components. Next, Principal Component Analysis is employed to compress the data by identifying the principal components from each set of real and imaginary data. Subsequently, the identified principal components are combined into a single dataset for further analysis. At this point, we apply blind source separation using complexity pursuit to isolate individual motion patterns that closely match the heartbeat responses measured by the oximeter. Tests with individuals in various situations have shown the achieved results were nearly identical to those obtained by a reference oximeter measuring real-time heart rate. We also establish comparisons to a widely known phase-based optical flow algorithm on benchmark videos. It is demonstrated that the proposed approach can derive comparable results in a simple fashion, reducing the number of steps for data processing and the efforts in terms of setup preparation. The study indicates the proposed video-based heart rate estimation approach is a viable and effective alternative, offering comparable results to conventional and other video-based alternatives. This non-invasive approach can be widely applied in medical fields and health monitoring contexts, significantly advancing the democratization of diagnostic and monitoring tools. It has the potential to be optimized for system embedding, such as into smartphones and security cameras, to ease its integration in practical applications.

Keywords Hilbert transform · Non-contact monitoring · Heart rate · Blind source separation · Principal components analysis

Introduction

Heart rate (HR) measurement is an essential practice for assessing cardiovascular health and monitoring in various clinical situations. Its importance extends to areas such as sports medicine, rehabilitation, and stress assessment. HR is a sensitive indicator of the body's physiological responses to different stimuli, such as physical exercise, mental stress, and pathological conditions [1]. Heart rate variability is a relevant indicator, as it reflects the ability of the autonomic nervous system to regulate HR in response to various conditions. High variability is generally associated with good cardiovascular health and an efficient adaptive response to stress, while low variability may indicate autonomic dysfunction and is linked to pathological states, such as specific stress and cardiovascular diseases. To further elaborate, studies show that regular monitoring of heart rate can help in the early detection of abnormalities in cardiovascular function, thereby preventing severe conditions such as myocardial infarction and stroke. This underscores the importance of accessible and accurate heart rate measurement tools in both clinical and remote health monitoring contexts. [2]

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The demand for telemedicine services and contactless medical monitoring methods has increased significantly, especially in response to the COVID-19 pandemic. Telemedicine, which includes virtual consultations and remote monitoring, has emerged as a viable solution to ensure continuity of medical care during periods of social restrictions and increased demand for healthcare. The use of information and communication technologies has been crucial to the expansion of these services, enabling safe and efficient interactions between doctors and patients [3]. Studies show that telemedicine not only facilitates access to care but also improves patient satisfaction. The implementation of telehealth services, such as teleconsultations and telemonitoring, has shown positive results in terms of clinical control and treatment adherence, especially in vulnerable populations, such as the elderly and those with chronic diseases [4].

Video-based heart rate (HR) monitoring has gained attention for its non-invasive nature and convenience, with optical flow-based algorithms standing out due to their ability to detect blood pulses by analyzing variations in light reflected by the skin. These algorithms, alongside techniques like Eulerian motion magnification, allow video cameras to capture subtle changes caused by blood flow, enabling remote HR estimation without the discomfort of electrodes [5, 6]. Despite their benefits, such methods face limitations, particularly when influenced by environmental factors like lighting variations and patient movement. The accuracy of these techniques heavily depends on image quality and the algorithm's ability to process data in real-time, with ambient light fluctuations sometimes leading to inaccurate measurements.

Proposed [7], the Hilbert transform-based video motion magnification method offers significant advantages over other techniques, such as Eulerian motion magnification. While Eulerian motion magnification is effective in revealing subtle temporal variations in videos, it faces challenges related to noise sensitivity and the need for a controlled environment to obtain reliable results. In contrast, the Hilbert transform-based approach allows for more robust handling of small motions by utilizing the imaginary component of the transform to access the amplitude of phase variations in the video over time. This technique is less susceptible to external interference and can withstand higher amplification factors, making it more suitable for applications in uncontrolled environments.

Given this, the aim of this paper is to explore a novel, non-invasive and cost-effective approach for HR monitoring using Hilbert-based motion magnification. The proposed methodology involves capturing subtle head movements resulting from the reaction of blood flow to heartbeats through video measurements. Initially, the Hilbert transform is applied to obtain the analytical signal, from which its real and imaginary components are extracted. Tests conducted with individuals in various situations demonstrated that the results obtained with the proposed approach were almost identical to those measured by a real-time reference oximeter.

The main contributions of this work include simplifying the process of non-contact HR monitoring and proving its effectiveness in practical situations. This non-invasive approach can be widely applied in medical and health monitoring contexts, significantly advancing the democratization of diagnostic and monitoring tools. Furthermore, the method has the potential to be optimized for incorporation into embedded systems, such as smartphones and security cameras, facilitating its integration into practical and everyday applications.

Theory

To measure heart rate (HR) by extracting motion patterns, we start from the premise that the algorithm detects a person's pulse by analyzing the periodic movement of the head caused by the pulsation of blood in the carotid arteries. This phenomenon, often referred to as Newtonian motion, results from the mechanical forces of blood flow during each heartbeat, based on Newton's third law of motion. As blood is pumped from the heart to the head, primarily through the carotid arteries, it generates subtle but measurable movements of the head due to the forces involved in this pulsate flow.[8, 9]

This displacements can be captured through video recordings that show frames with displaced or time-shifted image intensity. As described in [10], the algorithm is based on a combination of spatial and temporal processing to emphasize subtle changes in a video. After processing, the video goes through three stages to obtain the necessary parameters and calculate the heart rate: Hilbert Transform(HT), Principal Component Analysis (PCA), and Blind Source Separation (BSS). Once processed, the video is reconstructed with the appropriate amplification factor.

The algorithm combines the Hilbert Transform, PCA, and BSS to analyze the time series of pixels obtained from video measurements. The process begins by transforming the pixel intensity measurements into a displacement representation over time using the Hilbert Transform. This approach extracts the real and imaginary parts of the analytic signal estimated from the video, providing motion information of the scene.

The video is then reshaped into a matrix or time series, allowing for further processing. The first step is applying the Hilbert Transform. The HT is defined as the convolution of a function in the frequency domain. This phase shift implies that the imaginary part of the transformed signal is related to the real part through a specific phase relationship [11]. By calculating the instantaneous frequency and amplitude of signal components, it is possible to visualize the movement of objects in the scene. The resulting spectral analysis provides detailed information on how objects move over time.

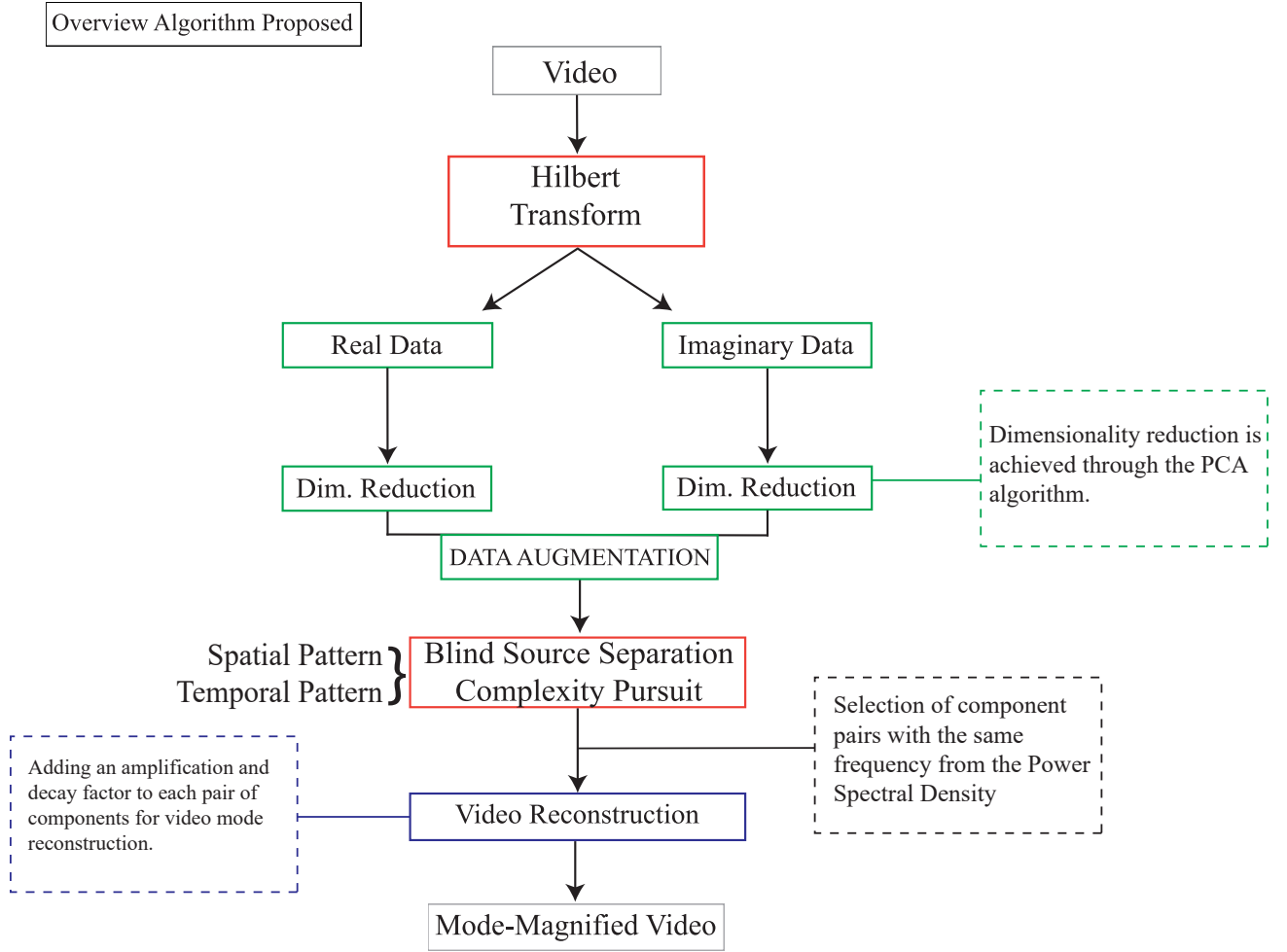


Fig. 1 Overview of the video Hilbert magnification algorithm.

Principal components analysis

After extracting the real and imaginary parts of the pixel time series, Principal Component Analysis is applied to reduce the dimensionality of these matrices. PCA is a widely recognized and used statistical technique for dimensionality reduction, especially in high-dimensional datasets like videos. Its main goal is to transform the original dataset into a new set of orthogonal components that preserve most of the variance present in the original data, but with fewer dimensions. Mathematically, PCA starts by calculating the covariance matrix C of the data. Where X is the centered data matrix, and n is the number of observations. Once the covariance matrix is obtained, the eigenvalues λ and eigenvectors v are calculated as follows:

$$Cv = \lambda v \quad (1)$$

The principal components are then determined by ordering the eigenvalues in descending order and selecting the first k eigenvectors, where k is the desired number of dimensions. This process reduces the dimensionality of the dataset, preserving the most significant characteristics, which is crucial for subsequent analysis [12]. This allows for the elimination of redundancies and noise, improving the accuracy of subsequent steps, such as BSS.

Blind source separation

After applying PCA, the reduced data is processed using Blind Source Separation (BSS). BSS is widely used for the modal identification of individual motion patterns. Here, the Complexity Pursuit (CP) is used to identify motion parameters of structures subjected to ambient excitation. These algorithms belong to the BSS class, which aims to recover mixed sources

from their observed outputs without prior knowledge of the mixing process or original sources. The relationship between the mixed signals $x(t)$ and the sources $s(t)$ can be mathematically represented as:

$$x(t) = As(t) \quad (2)$$

Where A is the unknown mixing matrix, and $s(t)$ are the independent source signals. The goal of Blind Source Separation is to find the matrix A and the sources $s(t)$, using only the observations $x(t)$. Once the mixing matrix A is estimated, the sources can be recovered by inverting the matrix

$$s(t) = A^{-1}x(t) \quad (3)$$

Complexity pursuit

In the Complexity Pursuit algorithm [13], the temporal complexity of the extracted signals is minimized. Complexity is defined as the unpredictability of a signal over time, and CP seeks to find a linear combination of the mixed signals that is the least complex, or most predictable. The complexity function $F(w_i, x)$ to be minimized is:

$$F(w_i, x) = \frac{V(w_i, x)}{U(w_i, x)}$$

Where $V(w_i, x)$ represents the long-term variance of the extracted signal, and $U(w_i, x)$ represents the short-term variance. The separation matrix W is found by maximizing the temporal predictability of the separated signals. The resulting signal sources correspond to the temporal patterns of each mode of movement in the scene. Based on this, the columns of the mixing matrix provided by BSS are used to expand the identified motion patterns to all frames of the video at high spatial resolution, using the projection matrix obtained during the PCA stage. This process allows the detected movement modes to be applied to the entire visual content, resulting in a detailed and accurate representation of the subtle movements present in the scene. After BSS, temporal and spatial patterns can be extracted, from which we can identify the temporal frequency of the subtle displacements caused by the heart rate pattern.

One of the main advantages of the proposed algorithm over Eulerian Motion Magnification is the absence of the need for manual selection of a region of interest (ROI) in the video. While the Eulerian method requires the selection of a specific area to amplify movements, the algorithm automatically amplifies movements across the entire scene. This eliminates selection bias, enabling a more complete and automated analysis of motion.

Moreover, the use of BSS in the algorithm allows for the isolation of relevant movements without amplifying noise or other unwanted movements. In the Eulerian method, all movements are amplified globally, which can lead to noise amplification and distortions, especially in noisy videos. Additionally, unlike phase-based optical flow, this method does not require the manual creation of spatial filters tailored to the movement direction. For example, if the displacements are horizontal, a horizontal filter is needed for optical flow, which requires prior knowledge of the motion direction in the video. This necessitates direct and critical human intervention. By separating and analyzing motion sources individually, the method results in more precise and focused amplification without the need for such manual adjustments.

Experimental results

The approach was implemented and executed in Python(3.12.16 version). Was performed on a device featuring an 11th Gen Intel(R) Core(TM) i5-1135G7 processor running at 2.40 GHz, with 24 GB of installed RAM. The system is a 64-bit operating system with an x64-based processor. The videos were recorded using mobile phone cameras in non-isolated environments with varying lighting conditions. All videos have a frame rate of 60 frames per second in HD resolution. Oximeters were attached to the subjects for comparison of the results. The results are presented in pairs because, at the beginning of the process, the Hilbert transform outputs the data in two components: one representing the real part and the other the imaginary part.

After identifying the principal frequency, we apply a simple formula to convert the frequency from Hertz (Hz) to beats per minute (BPM), where f is the frequency in Hertz:

$$\text{BPM} = 60 \times f \quad (4)$$

In Figures 3, 4, 5, and 6, the tests were conducted with individuals facing the camera. As shown in Table 1, the error between the results obtained from the oximeter and the frequency extracted by the algorithm is minimal, further demonstrating the effectiveness of the technique.

Visual representation of data from each test scenario

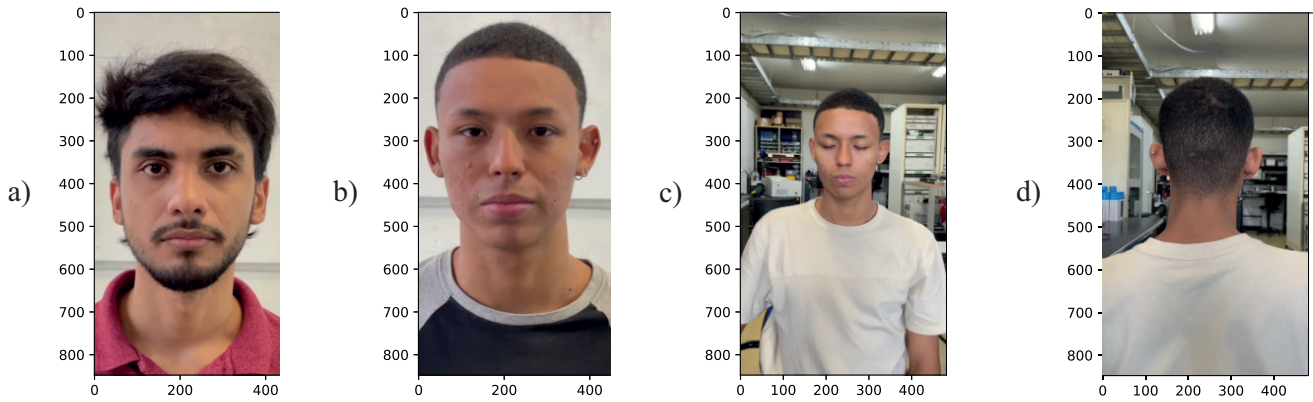


Fig. 2 (a) Subject 1, (b) subject 2, (c) subject's eyes closed, and (d) subject facing away.

Up to this point, promising results have been obtained. However, to push the analysis further, additional tests were conducted from different perspectives. A test with closed eyes (Figures 5 and 6) was performed to determine whether blinking could influence the outcome. Additionally, another test was carried out with the head positioned at a different angle, facing away from the camera (Figures 7 and 8), to assess where head movements related to arterial pulsation could be more effectively captured by the algorithm.

In all experiments, the error was minimal, and variations in head angle did not significantly impact the results. Each setup successfully captured the subtle oscillations of the head caused by the pulsation of the aorta artery, and the heart rate was consistently detected with accuracy.

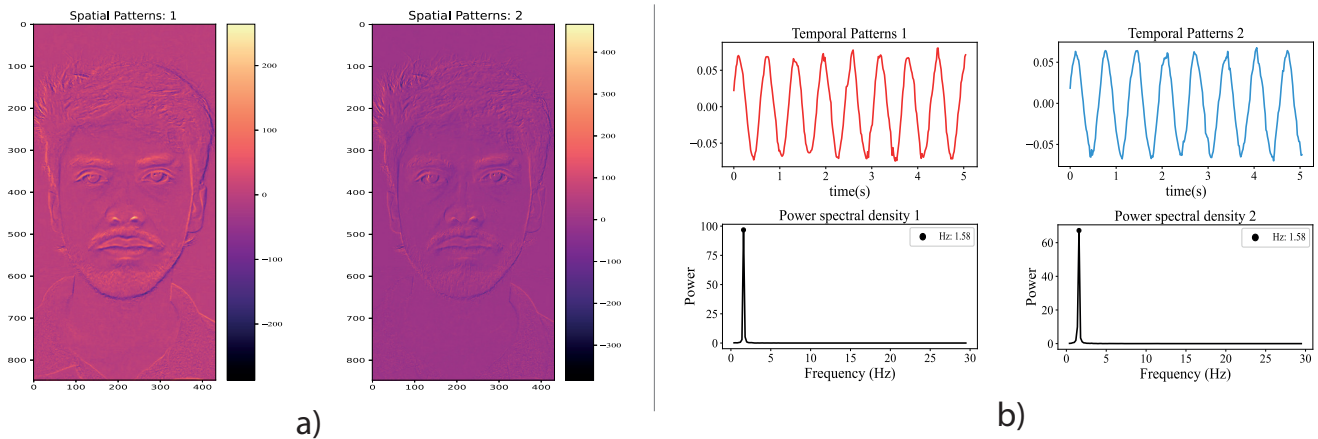


Fig. 3 (a) Spatial patterns, (b) temporal patterns, and power spectral density from the test with subject 1.

Table 1 Comparison of Test Results with Oximeter Readings in BPM

	Subject 1	Subject 2	Eyes closed	Facing away
Oximeter	100	73	75	75
Hilbert Magnification	94.8	72	72.6	74.4
Error(%)	5.2	1.37	3.2	0.8

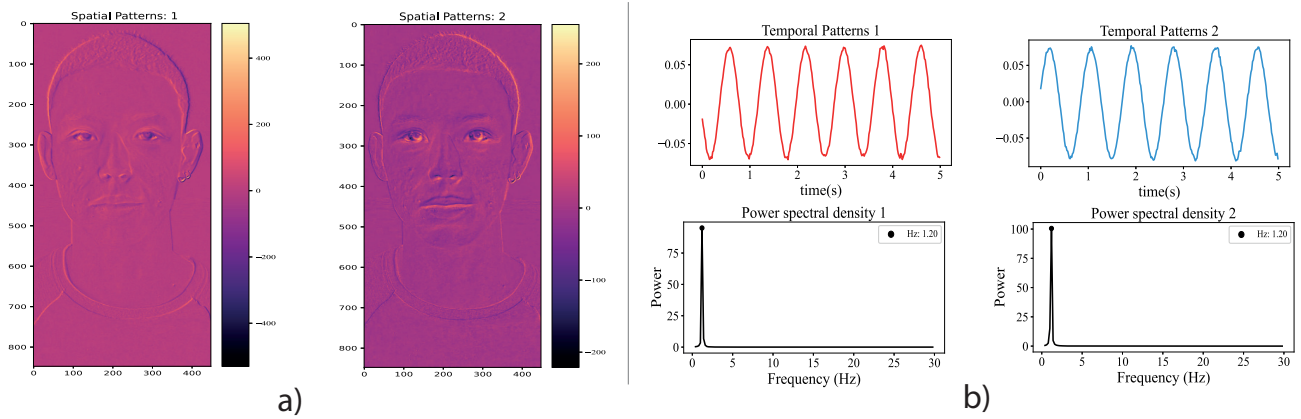


Fig. 4 (a) Spatial patterns, (b) temporal patterns, and power spectral density from the test with subject 2.

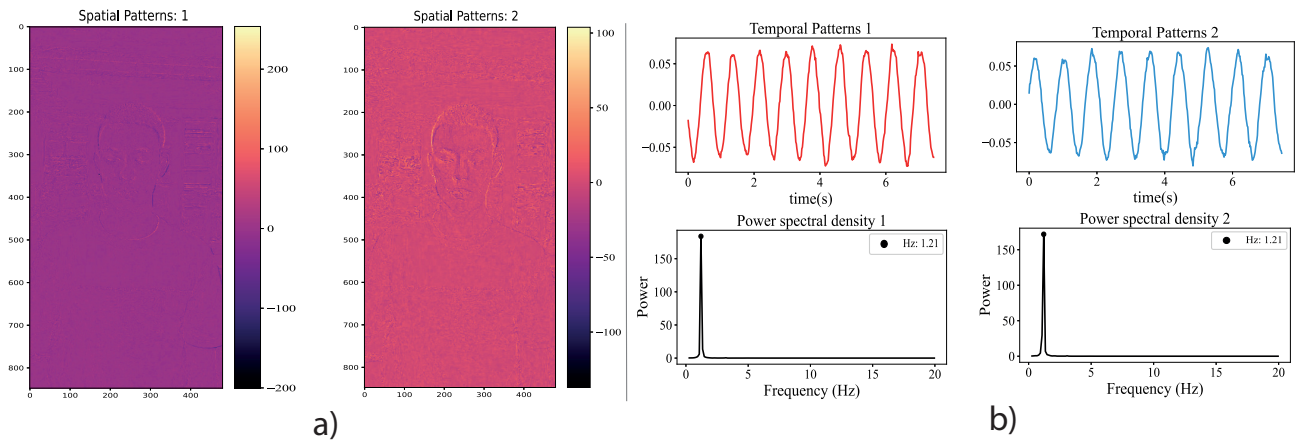


Fig. 5 (a) Spatial patterns, (b) temporal patterns, and power spectral density from the test with subject with the subject's eyes closed.

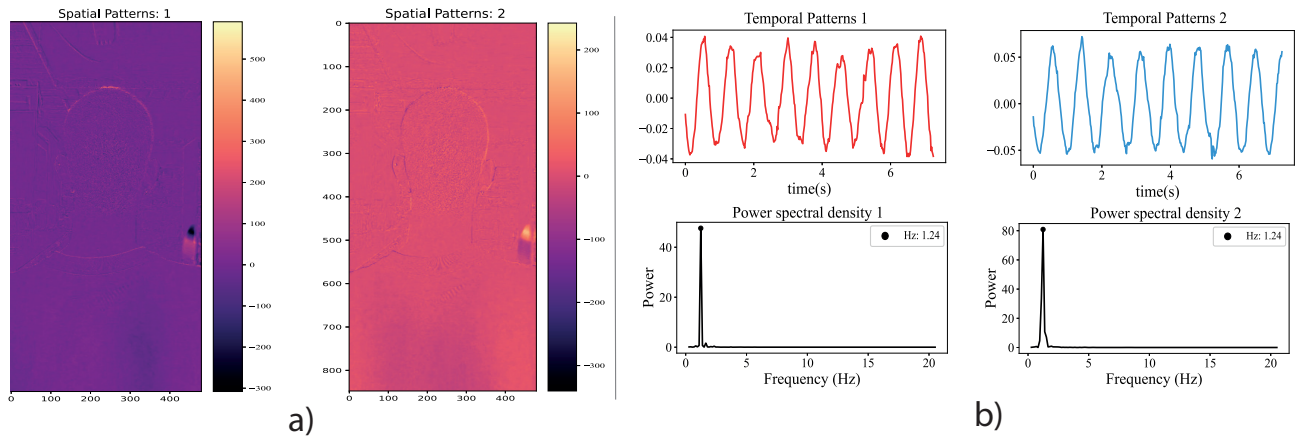


Fig. 6 (a) Spatial patterns, (b) temporal patterns, and power spectral density from the test with subject with the subject facing away.

Conclusion

This study explored an innovative, non-invasive video-based heart rate monitoring approach using Hilbert motion magnification. The proposed methodology proved to be effective in capturing subtle head movements caused by blood pulsation,

providing results comparable to traditional methods such as oximeters. Furthermore, comparisons with other algorithms demonstrated that the developed technique can achieve similar accuracy while simplifying the data processing steps and configuration preparation.

The results of the tests, carried out under various lighting conditions and head angles, indicate that the algorithm can be widely applied in health monitoring contexts without the need for physical contact, which is advantageous for applications such as telemedicine and remote monitoring. With potential for optimization and integration into embedded systems, such as smartphones and security cameras, the presented technique offers an accessible and low-cost solution for monitoring cardiovascular health. This approach can significantly contribute to the democratization of diagnostic and monitoring tools, paving the way for the development of technologies that promote more inclusive and efficient healthcare.

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Chapter 15

Variable Atwood Number Rayleigh-Taylor Instabilities Induced by Volumetric Energy Deposition

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Abstract Instabilities similar to the Rayleigh-Taylor instability (RTI) are inherent to internal confinement fusion (ICF) during the compression of target fuel capsules and ultimately lead to a reduction of fuel efficiency. While computational models for RTI in the context of ICF exist, lab-scale experimental work is needed to validate the assumptions and results of these models. Much experimental work investigating RTI has been performed, but few studies involve the condition of volumetric heating or the resultant temporal variation in fluid density that is present in ICF. This work investigated RTI induced by microwave volumetric energy deposition (VED), which caused buoyancy-driven mixing due to density stratification inversion. Experimentation involved a vessel with an initially stable system of two stratified immiscible fluids with different densities. As the two-fluid system was heated and heat transferred from the top fluid to the bottom fluid, the initially denser bottom fluid experienced preferential thermal expansion at the fluid interface, inducing buoyancy-driven mixing. The temperature at various points in the fluids was recorded using fiber optic temperature probes, and the temperature data were then used to calculate density. It was found that density inversion did not occur at the temperature probe locations, but the data support the hypothesis that density inversion occurred close to the fluid interface. Since the refractive indexes of the fluids are different, the phenomenon was visualized with shadowgraph videography. Once the mixing process was recorded, images were processed with edge detection and motion tracking techniques. Image processing was used to unveil the presence of morphological features of the flow such as spikes and bubbles and to quantify the evolution of the wave properties. The results were compared to traditional RT equations, and it was found that initial wave amplitude growth agreed with traditional RT literature, while the later stage was inconsistent with traditional RT literature.

Keywords Rayleigh-Taylor instability · Unstable stratification · Volumetric energy deposition · Microwaves

Introduction

Hydrodynamic instabilities have long been studied because of their relevance in several physical processes, including challenges such as geological storage of CO₂ [1], multi-phase chemical reactions [2], droplet breakup [3, 4], and inertial confinement fusion (ICF). This work primarily addresses the challenges found in ICF, where development has been hindered by a lack of understanding of the hydrodynamic instabilities inherent to the process [5]. In ICF, a spherical fuel capsule containing a deuterium-tritium (DT) core is bombarded with x-rays, which initiates a shock wave travelling radially inward. The shock wave heats and compresses the DT core to initiate a fusion reaction [6]. During this process, the fuel capsule shell is vaporized into plasma, and the surface irregularity of the shell combined with the shock wave causes mixing of the plasma and the DT fuel, causing some of the fuel to escape the highly pressurized and heated center before it can fuse. The

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hydrodynamic instability at the interface between the plasma and capsule shell after the shock demonstrates similar fluid dynamic behavior to Rayleigh-Taylor instability (RTI).

RTI has been studied extensively to further understand the ICF process. Traditionally, RTI experiments and simulations involve an initially stratified system of two fluids of different densities in a gravitational or acceleration field. A measure of the buoyancy of the bottom fluid in the top fluid is given by the dimensionless Atwood number, At :

$$At = \frac{\rho_1 - \rho_2}{\rho_1 + \rho_2}, \quad (1)$$

where ρ_1 and ρ_2 are the densities of the top and bottom fluids in the system, respectively. In the presence of an acceleration field such as gravity, the system is considered unstable when At is positive (i.e., when the top fluid is denser than the bottom fluid, and $\rho_1 > \rho_2$). Surface tension maintains the initial state until there is a perturbation (disturbance) at the interface. Then, gravity acts to propagate the disturbance towards the stable state where the denser fluid rests at the bottom of the system. For a perturbation with amplitude η and wavelength λ in a simple system of two immiscible and incompressible fluids, the initial growth rate of small amplitude perturbation, i.e. ($\eta_o \ll \lambda$), is given by Eq. (2), where g is a constant acceleration (often gravity) [7].

$$\eta(t) = \eta_o \cdot \exp \left[\left(\frac{2 \cdot \pi \cdot At \cdot g}{\lambda} \right)^{1/2} \cdot t \right] \quad (2)$$

As mixing progresses, η and λ become similar, η is thought to grow at a constant rate [7, 8]. Further in time, the perturbations become broadband and chaotic in nature, a mixing region of height h develops. At this stage, $h \gg \lambda_m$, where λ_m is the maximum wavelength of the initial perturbations [7, 8]. In idealized conditions (where the viscous and diffusion length scales are much less than h), the growth of h is given by

$$h(t) \approx \alpha \cdot At \cdot g \cdot t^2. \quad (3)$$

In this traditional understanding of RTI, the density of each fluid is constant (and thus At is constant). However, in ICF, constant volumetric energy deposition (VED) imparted by fusion products (neutrons and alpha particles) and the incoming x-rays cause thermal expansion and subsequent density variation of pellet materials. This creates a situation in which At varies with time (Eq. (4)).

$$At(t) = \frac{\rho_1(t) - \rho_2(t)}{\rho_1(t) + \rho_2(t)} \quad (4)$$

Some work has been done to demonstrate variable At , VED-driven RTI at the lab scale. Joshi et al. [9] investigated VED-driven RTI using dielectric fluids in a nonuniform electric field, where one fluid is preferentially heated due to its permanent or induced dipole. Wachtor et al. [10] demonstrated this result experimentally with microwave-driven RT mixing of two miscible fluids. In another experiment, it was demonstrated that the mixing layer growth rate in VED-driven RTI is increased over the classical RT t^2 scaling of Eq. (3) [11]. Recently, Root et al. [12] demonstrated RT mixing of two immiscible fluids through microwave heating, leveraging the different refractive indexes to obtain shadowgraph videography of the growth of perturbations. There has yet to be an experiment that collects and analyzes thermal and video data resulting from RTI between two immiscible fluids through microwave heating to perform a quantitative comparison to classical RTI literature.

The focus of this work is to further inform modeling RTI in ICF with an improved experiment involving microwave heating of two immiscible fluids. The experimental setup was designed to reduce the 3D effects of the mixing and isolate the early stages of RTI described by Eq. (2) and Eq. (3). Temperature data were collected at a vertical cross-section of the fluids and utilized to calculate the fluid density gradient during mixing. Processing and analysis of video capturing the mixing process allowed for the quantification of perturbation amplitude growth, which was compared to the traditional, constant At RT models.

Methods

Fluid Selection

Fluid selection was based on work performed by Root et al. [12], who demonstrated VED-driven RTI using 7 cSt polydimethylsiloxane (PDMS) and soybean oil. The pair was chosen based on their immiscibility, sufficient microwave penetration depths for the fluid vessel, and capability of density inversion through heating. The fluids' immiscibility improves the visibility of the RTI behavior and isolates macroscopic behavior with the tradeoff of introducing interfacial tension effects.

The 7 cSt PDMS is the denser of the pair at an ambient temperature of $\sim 25^\circ\text{C}$, but for any shared temperature above 43°C , soybean oil is denser (Figure 1). Both fluids are also non-hazardous. Soybean oil is primarily composed of triacylglycerols, a type of lipid. PDMS is a linear-chain polymer with chemical formula $(\text{SiO}(\text{CH}_3)_2)_x$, whose length dictates its viscosity [13]. The combination of 7 cSt PDMS and soybean oil was found to support RTI development as demonstrated by [12] and was used in the experiments reported below. Each fluid's density as a function of temperature in Figure 1 was used to estimate the fluid density over time during mixing from collected temperature measurements.

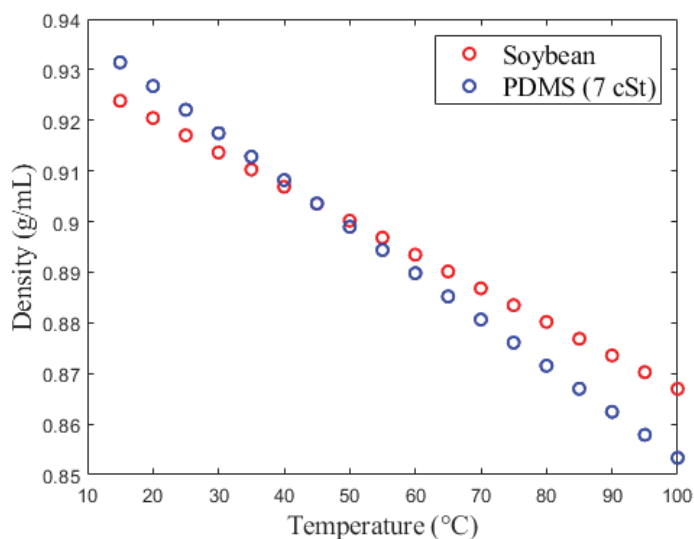


Fig. 1 Density as a function of temperature for soybean oil (Thermo Scientific Chemicals, A12728.0E) and 7cSt PDMS (Gelest, DMS-T07-KG) as measured using a densitometer (Anton Paar; Model: Modulizer Polymers).

Experimental Setup

A custom, rectangular quartz vessel was used to contain the fluids. Quartz is optically transparent, thermally resistant in comparison to other glasses, and has low dielectric loss, making it an ideal vessel material for the microwave heating experiments and video imaging. The vessel had external dimensions of 150.01 mm (length – L) x 21.35 mm (width – W) x 152.40 mm (height – H). All vessel walls were 3.175 mm thick, giving an internal vessel width of 15 mm. The internal width of 15 mm was chosen, because a video image from [12] shows the base of a spike to be about 15 mm. The goal was to produce a system as 2-dimensional as possible while minimizing wall effects. A Teflon stand was constructed to stabilize the thin fluid vessel. Like quartz, Teflon was also chosen for its low microwave absorption. Fiber optic temperature probes (Omega Engineering Inc., FOS-LU-2) connected to a data logger (Omega Engineering Inc., FOM-H201) were utilized to collect temperature data with an accuracy of $\pm 0.8^\circ\text{C}$.

A large format microwave (Microwave Research and Applications, Inc, Model BP-210) was used to supply VED to the fluids. The microwave has an operating frequency of 2.45 GHz and maximum power output of 2,100 Watts, and its cavity has dimensions of 53.5 cm (W) x 33.0 cm (L) x 25.1 cm (H). Three 3.8 cm diameter access ports were added to the microwave — one on each side and another on the door — for the purpose of imaging. The access port on the left side was used to provide LED backlighting to the vessel (Nila, Model NZL1D-D). A piece of parchment paper taped against left port and a 0.635-cm thick Teflon plate placed next to the vessel diffused the backlighting to provide more even backlighting to the two-fluid system. A CMOS camera (Point Grey, Model GS3-U3-23S6M-C) and a focusing lens (Fujinon, Model HF35SA-1) placed through the access port on the right side of the microwave door were used to obtain grayscale, 10 frames per second (fps) videos of the mixing process (Figure 2).

Single-Fluid Calibration Procedure

To develop an understanding of how each fluid would change temperature over time for a given microwave power level and vessel, single-fluid calibration experiments were conducted. Only measurements conducted at a microwave power level of 840 W (40% of the microwave's maximum power) are reported in this work. The fluids' microwave penetration depths

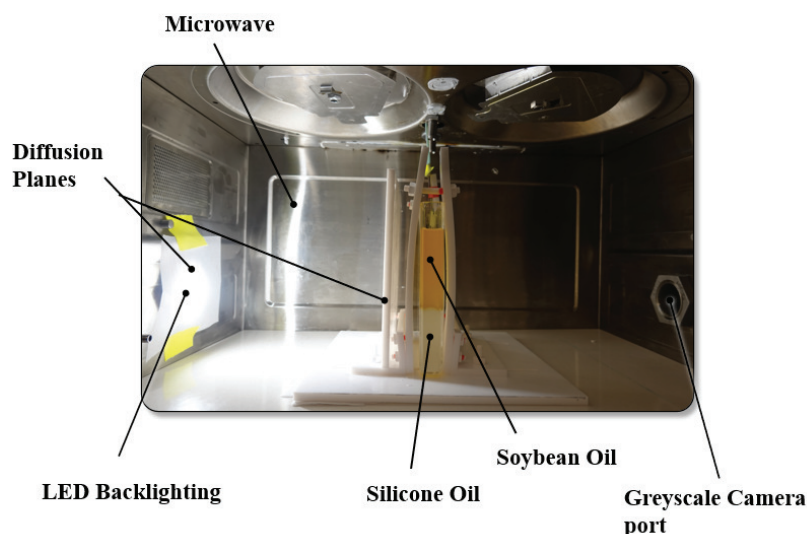


Fig. 2 Experimental setup of two-fluid system in the microwave cavity.

were much larger than the dimensions of the vessel and the total fluid volume for the single-fluid experiments was chosen to be equal to that of the mixing experiments. As such, the single-fluid temperature data were intended to inform the fluid temperatures during the mixing experiments. 270 mL of each oil was heated from room temperature (between 19 °C and 23 °C) in the quartz vessel for 3000 seconds. Six fiber optic temperature probes were fed through 4mm diameter holes spaced 16 mm apart in a 0.635-cm thick Teflon plate resting on top of the vessel (Figure 3). The tips of the probes reached approximately halfway down into the fluid. The temperature measurement from each probe was recorded at a frequency of 1 Hz. Three trials were performed for each fluid.

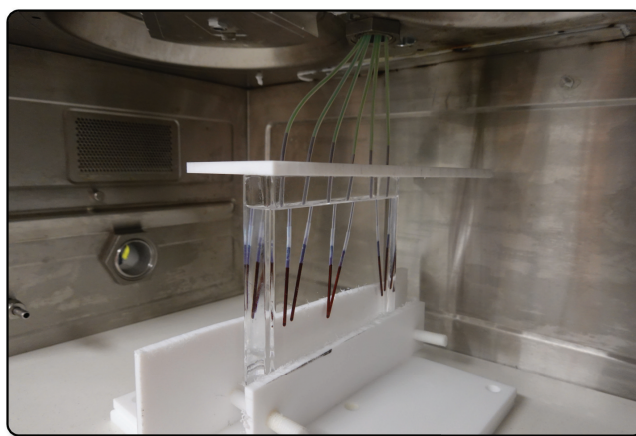


Fig. 3 Single-fluid calibration setup. Six fiber optic temperature probes were distributed along the length of the fluid vessel.

Rayleigh-Taylor Mixing Experiment Procedure

For mixing experiments, an equal volume (135 mL) of room temperature (~ 21 °C) soybean oil and PDMS were loaded into the vessel. Soybean oil was poured in the vessel first, then the initially denser PDMS was loaded into the bottom of the vessel using a syringe and tube. The vessel was secured in the Teflon stand, and the microwave was powered at ~ 840 W for 3000 seconds. Six temperature probes were placed outside the field of view of the camera at incremental heights in the fluids, with three in the bottom fluid and three in the top fluid (Figure 4). The temperature data was then used to calculate the fluid densities over time at each measurement height using relationships between density and temperature data found for each fluid.

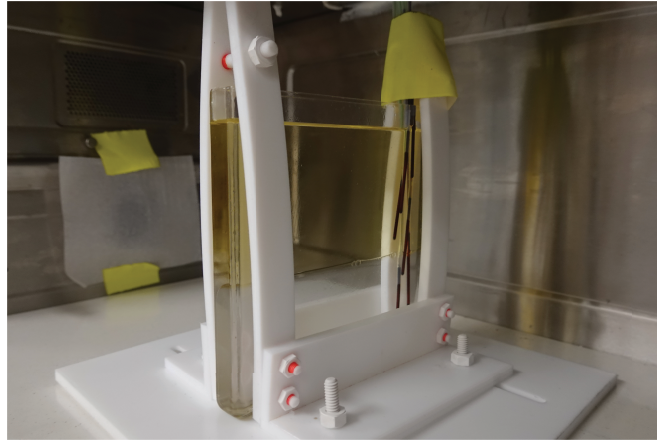


Fig. 4 VED RTI mixing experimental setup before the Teflon diffusing plane was inserted into the microwave cavity.

Image Processing

An image processing workflow was applied to the videos of the mixing process to characterize the evolution of the fluid interface (Figure 5). Preprocessing steps included cropping the video frames to a region of interest, background correction to account for uneven backlighting (Figure 5b-c), and smoothing to reduce noise and enhance the fluid interface (Figure 5f). Background correction was performed by using an iterative Gaussian smoothing filter on each raw image (Figure 5a) to

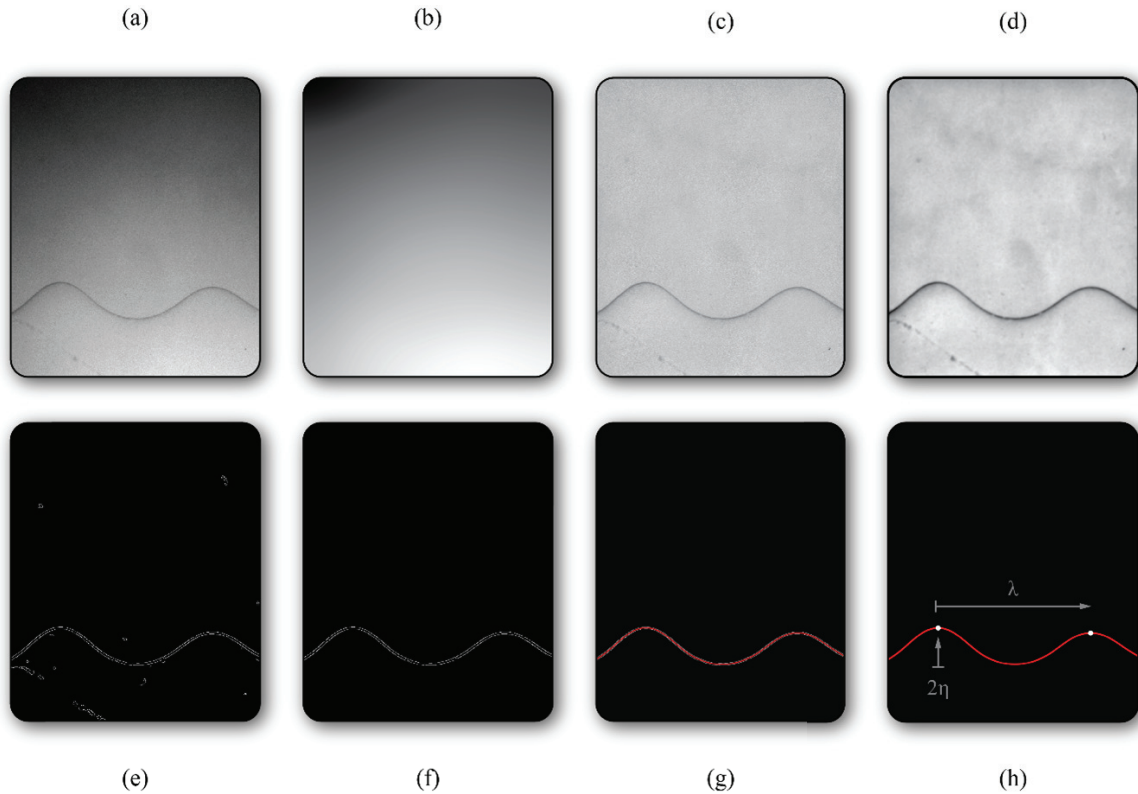


Fig. 5 The image processing workflow used to extract the amplitude (η) and wavelength (λ) of the wavelike fluid interface. (a) Raw video frame, (b) estimation of the uneven backlighting, (c) frame corrected for uneven backlighting, (d) smoothed image to enhance the fluid interface and remove noise, (e) result of Canny edge detection, (f) cleaned edge detection image, (g) interface estimate using a piecewise polynomial (spline), and (g) estimate η and λ from automated image analysis workflow.

capture the background produced by uneven backlighting (Figure 5b). The raw image was then corrected using the estimated background to produce an image where contrast is concentrated at the fluid interface (Figure 5c). Smoothing was performed via guided filtering, technique used to reduce noise while preserving the sharpness of edges. Canny edge detection was then used to locate the fluid interface (Figure 5e). Noise in the resulting binary image was reduced using a tuned cleanup step in which long, continuous edges were preserved, and short edges were removed (Figure 5f).

The pixels identified as the fluid interface were used to obtain η and λ . The locations of edge pixels were used to fit a piecewise polynomial (or spline, Figure 5g). Smoothing of the spline was adjusted to capture the curvature of the wave while avoiding overfitting. The distance between the minimum and maximum of the spline was considered 2η , and the distance between the maxima of the wavelike interface was considered λ (Figure 5h). The maxima were found by computing the first and second derivatives of the spline and selecting those points whose first derivative was approximately zero and second derivative was negative. In some cases, the λ was instead calculated using the minima, which was done using the same procedure but with a positive second derivative condition.

The image processing workflow was automated and applied to each frame in the videos capturing the mixing process. The evolution of η and λ in time was tracked. Curves for η versus time were fit for two periods during the mixing process: when $\eta \ll \lambda$ and when $\eta \approx \lambda$ (the threshold defined at 0.1λ).

Results

Single-Fluid Calibration

Six temperature probes captured the temperature of each isolated fluid while heated in the microwave. The temperature evolution followed the shape of a decaying exponential, with the soybean oil having a much higher end temperature and time constant than 7cSt PDMS (Figure 6). For each trial, it was found that the variation in temperature between various probes increased over time and reached a maximum of less than 10% of the increased temperature difference ($\sim 10^\circ\text{C}$ for soybean oil, $\sim 3^\circ\text{C}$ for the 7cSt PDMS). When the single-fluid temperature data was combined with fluid density-temperature to estimate the At variation, it was found that the fluid system would never an unstable configuration under the assumption that the mixing fluid temperatures would be equivalent to the single-fluid calibration temperatures (Figure 7).

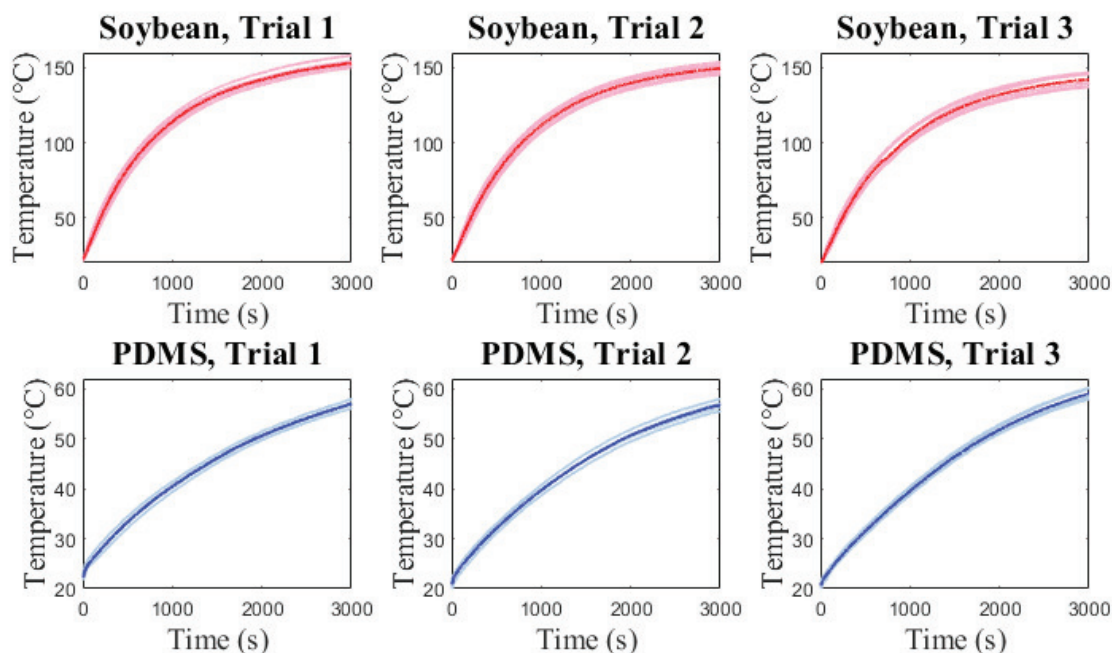


Fig. 6 Single-fluid calibration temperature curves for soybean oil (red) and PDMS (blue) heated at 840 W. Lighter lines indicate individual temperature probes and bold lines represent the average temperature over all the probes for that trial.

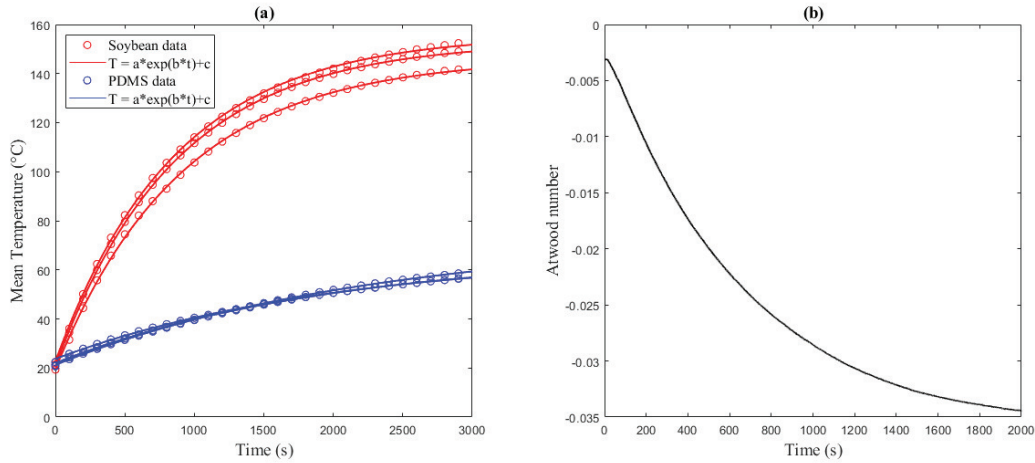


Fig. 7 (a) Average temperatures for all single-fluid trials with decaying exponential fits. (b) Estimated Atwood number determined using single-fluid temperature measurements and density-temperature relationship measured in Figure 1.

VED Mixing Experiment

Temperature probes were aligned vertically in the two-fluid system outside the field-of-view of the imaging system to capture the temperature of each fluid at the top, middle, and bottom of each respective fluid. As in the single-fluid experiments, the soybean oil exhibited higher overall temperatures than the PDMS. The middle region of each fluid reached a maximum temperature difference of about 50 °C after 900 seconds at the neighboring probe locations. During this time, the soybean oil exhibited a temperature difference of ~ 20 °C from the top of the fluid to near the fluid interface, while the PDMS exhibited an increase in temperature ~ 12 °C between the bottom and the near fluid interface (Figure 8a).

The shadowgraph video of the mixing process captured the fluid interface well. An initial single wavelength perturbation began at about 600 seconds, which grew in amplitude until the PDMS spikes experienced necking and bubbles separated from the vertical fluid jets. As fluid heating continued, the spikes cyclically grew, necked, and broke off into bubbles (Figure 9).

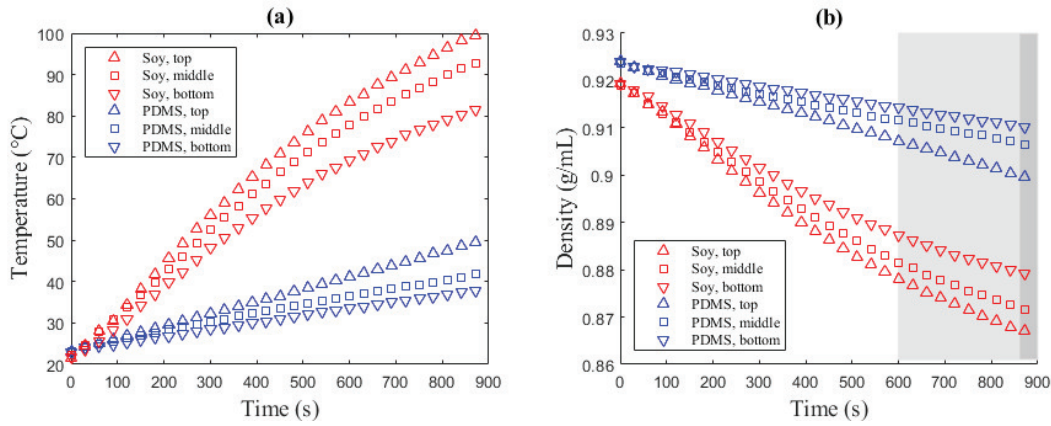


Fig. 8 Temperature data (a) and estimated density (b) versus time for RT mixing experiment. The light grey area in the density graph indicates the time region in which video footage showed growing perturbations, while the dark grey area indicates the time region of bubble formation.

Image processing and analysis of the mixing experiment reveal growth in η and a decrease in λ , as shown in Figure 10. Analysis was limited to a specific timeframe of interest, beginning around 600 s (when the initial perturbation at the interface was detected) and ending at 840 s (when bubble separation began to occur). Some noise is present near the end of the timeframe in the amplitude data (Figure 10). This occurs where there was not sufficient contrast of the interface for the image processing algorithm to detect, e.g. Figure 9C. An exponential model of the form of Equation 2 was fit to the data where $\eta \ll \lambda$ (Figure 6, $R^2 = 0.998$). In this preliminary analysis, no satisfactory fit was found for data where

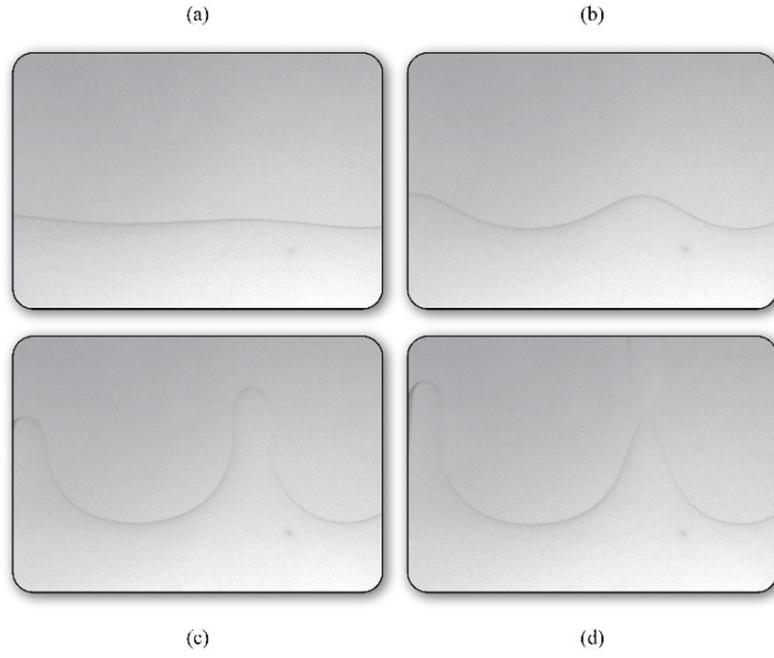


Fig. 9 Frames from video of the mixing experiment with six temperature probes. (a) 600 s – shows the beginning of a wave-like form, (b) 750 s – shows more amplitude growth, (c) 840 s – shows a spike right before necking begins to occur, (d) 853 s – shows necking of the right spike, with a bubble about to break off.

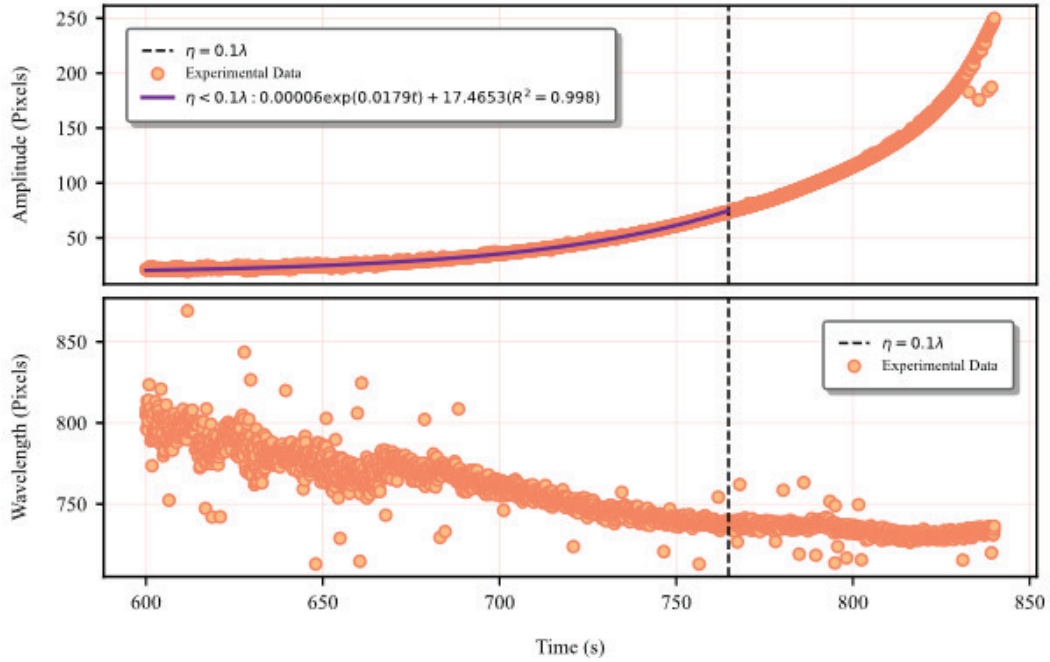


Fig. 10 The amplitude (top) and wavelength (bottom) determined from the mixing experiment video images. A dashed black line indicates the point in time where $\eta = 0.1\lambda$, which was the threshold used to distinguish the period when $\eta \ll \lambda$ and when $\eta \approx \lambda$. A model form from the literature (Equation 2) was fit the amplitude data where $\eta \ll \lambda$.

$\eta \approx \lambda$, but the growth is nonlinear, which disagrees with classical linear RTI growth reported in literature [7, 8]. The wavelength calculations showed a decrease during the initial period when $\eta \ll \lambda$, but approached steady-state as η continued to increase.

Discussion

Single-fluid calibration experiments were used to characterize of the microwave field and each fluid's temperature response to it. Given the large microwave penetration depths of both fluids compared to the fluid vessel size, Figure 5 suggests that it is possible that the microwave field was not uniform within the cavity and further investigation of this matter should be performed. The uneven microwave field could have been the generator of the initial perturbation at the fluid interface.

It was originally hypothesized that the temperature response of the fluids in the single-fluid experiment would be comparable to that of the mixing experiment; however, the Atwood number estimated from the single-fluid data never turned positive to indicate the system would become unstable. The soybean oil increased in temperature faster than PDMS, which is explained by soybean oil having a dielectric loss about 5 times greater than PDMS with a similar heat capacity and complex permittivity [14, 15, 16, 17, 18]. The dielectric loss quantifies the dissipation of electromagnetic energy through a fluid, heat capacity indicates the amount of energy required to change the temperature of the fluid, and the complex permittivity determines the reduction of the electromagnetic field intensity when it enters a material. In reference to Figure 1, this led to the soybean oil maintaining a lower density in comparison to the 7 cSt PDMS. In the future when selecting fluids for microwave VED-driven RTI, both the dielectric constant and the relationship between temperature and density need to be considered for identifying a fluid pair that will undergo density inversion with VED.

The temperature probe data from the mixing experiment indicated heat transfer from the soybean oil to the PDMS due to conduction at the fluid-fluid interface; this is supported by the soybean oil decreasing in temperature closer to the interface and the PDMS increasing in temperature closer to the interface. This supports the development of RTI captured by the video despite the simple estimation of the At remaining negative. Future work should identify fluids in which bulk density inversion, rather than localized density inversion, can be achieved.

Wave amplitude growth in the $\eta \ll \lambda$ region of the video appears consistent with the literature description of exponential growth (Equation (2)). In the $\eta \approx \lambda$ region, the At continued to change and deviation from the literature description of growth was observed; however, this stage of growth is not well understood [7]. Since there is conduction between fluids over the timescale of the reported experiment, further investigation is required to understand the late stage growth behavior and identify the assumptions used in traditional RTI models that are not valid and formulate an alternative growth model.

Conclusions

Advancements were made to improve an experiment involving microwave heating of two immiscible fluids intended to aid modeling efforts of VED-driven, variable At RTI for ICF applications. Temperature measurements were collected during heating to characterize the interactions between the microwave field and fluid system. It was found that the fluids heated at significantly different rates and conduction between the two fluids drove the instability; this was in opposition to the intended preferential heating and buoyancy-driven mixing caused by VED alone. Further work to select fluids that would exhibit the desired preferential heating and density subsequent density inversion is needed. An image processing workflow was developed to extract the wavelength and amplitude of the wave-like fluid interface over time. Preliminary analysis shows behavior consistent with the classical RTI literature when the amplitude was much less than the wavelength; however, classical linear RTI behavior was not observed when the amplitude and wavelength were similar orders of magnitude. Future work that remains before these experiments can be used to validate mixing models with VED source terms includes imposing initial conditions to the two-fluid interface, better characterizing the conduction and viscous effects present in experiment, and further extraction of relevant video and temperature data to better inform modeling. Experiments performed at different microwave power levels and investigation into the uniformity of the microwave field uniformity would also further elucidate the performance of the experimental setup for VED-driven RTI.

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Chapter 16

Identification of Linear Time-variant Rotor Components in Resonance Crossing Experimental Validation

Luigi Carassale, Roberto Guida, and Davide Iaffaldano

Abstract The mechanical properties of rotor blades used in turbomachinery are identified exploiting the transient amplification that appears when an integer multiple of the rotor speed matches a resonance frequency of the component. To this purpose, the measured dynamic response is suitably processed and fitted with an analytic model representing a linear time-invariant system. In some cases, however, the variation of the blade stiffness due to the changing rotor speed is large enough to make the mentioned approach inaccurate as the system under study is time variant. A mathematical method to deal with this problem formulated using a perturbation method was recently proposed and validated against numerical simulations.

This paper describes an experimental study carried out to validate the identification technique mentioned above. To this purpose, it has been realized a test rig consisting of a vibrating beam whose natural frequency can be continuously varied through a pneumatic cylinder. The excitation is provided through an electromagnet driven by a linear chirp simulating and engine-order excitation. The response is measured by a laser doppler vibrometer. The system is set in such a way to have two closely-spaced resonances on multiple harmonics of the excitation.

Keywords Resonance crossing · Time variant system · Identification

Introduction

The modal identification of rotor blades in turbomachinery is typically carried out by exploiting the amplification that occurs whenever an integer multiple of the rotor speed crosses a resonance of the component during the run-up or cast-down of the engine. This operation implies three main difficulties that are somehow addressed by the techniques that are presently available.

The first problem regards the fact that tests cannot be repeated several times like it is usually done in experimental modal analysis. It is because the engine startup usually requires a long preparation, the instrumentation installed on rotor components have usually a very short life, the time employed for testing have a very high economic impact for operating engines. Besides, even when it is possible to repeat tests, the results are not fully repeatable as the operating conditions of the component, in particular the temperature, can be hardly controlled. For this reason, identification techniques that perform a local averaging of the data are preferred, e.g. the short-time Fourier transform is used instead of the global Fourier transform.

The second issue that makes the modal identification difficult is related to the unavoidable transient response produced by the resonance crossing, combined to the impossibility of measuring the force acting on the component, which has an aerodynamic origin. The transient nature of the response prevents the use of Operational Modal Analysis (OMA) techniques, at least when transient effects are relevant. This issue has been addressed following two alternative strategies. The first idea consists of fitting the transient response of a mode isolated through band-pass filtering with an analytic (approximated) expression of the transient response of a s-dof system in resonance crossing [1]. The second approach is based on the separation of the synchronous response from the non-synchronous transient component through a time-frequency filter [2]. The synchronous response is then fitted by the Frequency response Function (FRF) of a linear system.

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The third difficulty of modal identification appears as the stiffness of rotor blades can be significantly modified by the rotor speed as the centrifugal force has a stiffening effect. For this reason, the natural frequency of the blades changes during the engine run-up (usually increases) and may vary significantly even during the amplification phase produced by the resonance crossing. This effect makes the system under observation as time variant. To address this issue, two alternative strategies have been proposed. The first one consists of modifying the synchronous response extracted by time-frequency filtering exploiting a time-variant phase-shift defined according to an expression dependent on the rate of variation of the natural frequency [3]. The second approach relies on the application of a specifically-designed time-frequency filter realized through a wavelet transform whose mother wavelet is rotated according to the rate of variation of the natural frequency [4].

This paper describes the laboratory validation of these techniques by means of a test rig in which the natural frequency of a beam can be modified in a continuous manner. The laboratory setup enables the comparison of the modal parameters identified in transient and time variant condition with the parameters identified by the traditional experimental modal analysis.

Test Rig

The test rig is composed of a thin still beam whose natural frequencies are modified by changing its axial load. One extremity of the beam is fixed to the frame, while the other end is fixed to an anchor block that only allows the axial component of the displacement. The axial load of the beam is controlled by a pneumatic cylinder (Fig. 1a). The excitation is produced through an electromagnet and measured by a piezoelectric load cell (Fig. 1b). The moving end of the beam is connected to the frame through a pair of elastic elements in such a way to be relatively flexible in axial direction of the beam and very stiff in all the other directions (Fig. 2).

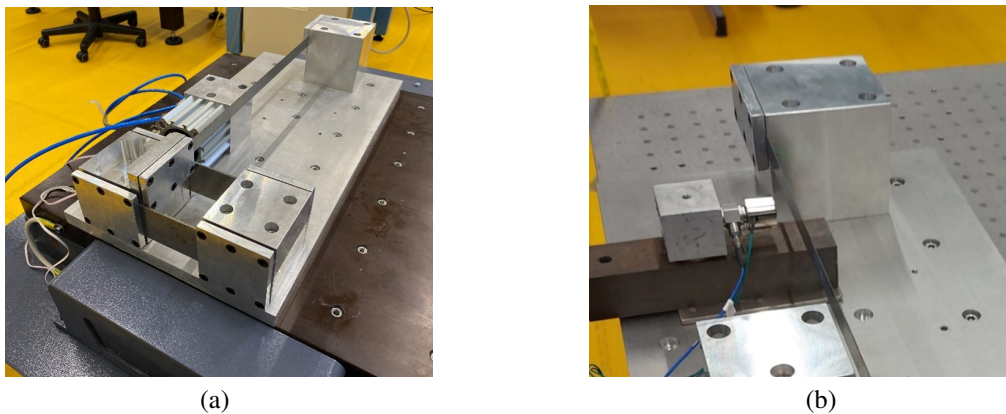


Fig. 1 Test rig (a), electromagnet used for excitation (b).

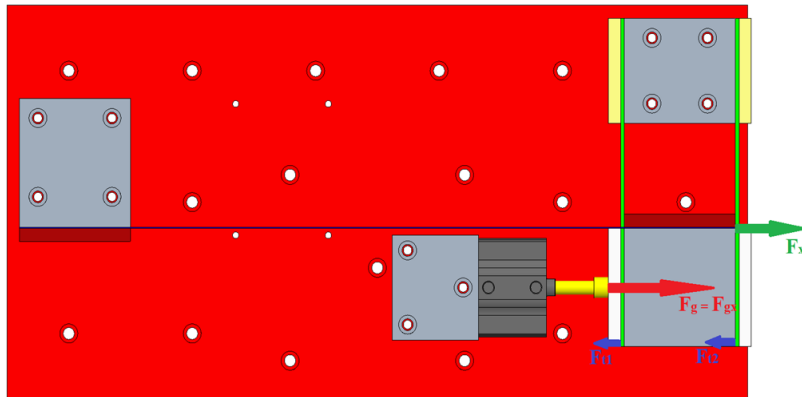


Fig. 2 Working schema of the test rig.

Modal Identification Of The Beam In Stationary Condition

The modal properties of the beam have been identified using the magnet force as input and the velocity measured in a series of points of the beam through a laser doppler vibrometer as output. The test has been repeated for several values of the axial force, produces by the pneumatic cylinder and measured by a strain gauge mounted on the beam. The results discussed herein refer to the second bending mode of the beam (Fig. 3).

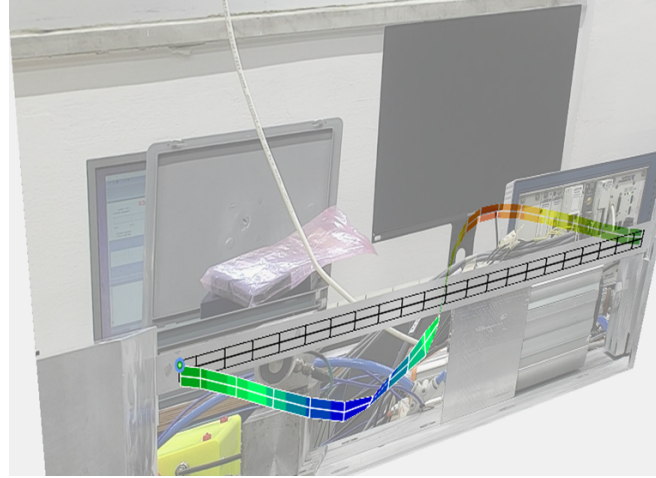


Fig. 3 Mode 2 of the beam used for comparison.

Figure 4 shows the natural frequency and damping ratio of mode 2 as a function of the axial force F (circles). The natural frequency increases linearly as the force increases. The estimated damping initially decreases, but for high force levels have an irregular behavior that still needs to be explained. One hypothesis is related to the difficulty of keeping the pressure in the cylinder constant for the whole experiment. The slight variation of the natural frequency turns into an increment resonance peak bandwidth and, as a consequence, of estimated damping. A refinement of the test rig to ensure a more stable axial force is currently under realization.

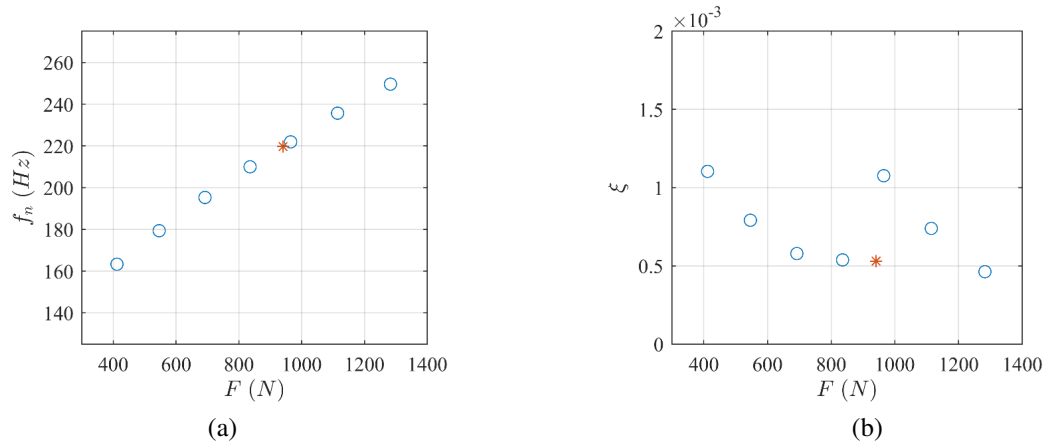


Fig. 4 FRF of the beam in the frequency range of mode 1 for different values of the pressure.

Modal Identification Of The Beam In Transient And Time Variant Condition

For this second test, the beam has been excited through the magnet drive by a linear chirp to simulate the synchronous force acting on rotor blades during the engine cast down. During the reduction of the excitation frequency, the air in the pneumatic

cylinder has been progressively released to produce a reduction of the natural frequency. Figure 5 shows a spectrogram of the vibration velocity of the beam measured during the test. It can be noted that the first and the second harmonics of the magnetic force cross mode 1 and mode 2 of the beam almost simultaneously.

The resonance crossing of mode 2 has been analyzed using the method described in [3] leading to the results shown in Fig. 4 (star). It can be observed that the results obtained in transient and time variant condition are very close to what it was obtained by the experimental modal analysis.

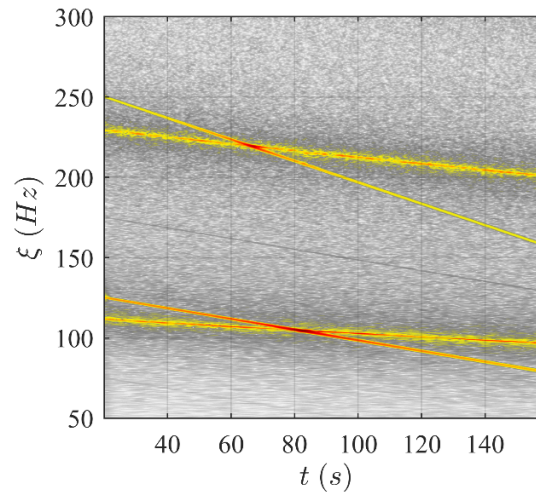


Fig. 5 FRF of the beam in the frequency range of mode 1 for different values of the pressure.

Conclusion

This paper describes a test rig designed to reproduce the resonance crossing of systems whose natural frequency changes progressively with time. The preliminary results pointed out some issues with the stability of the working pressure that affected some results (in particular damping for high pressure levels). However, the results confirmed that it is possible to identify correctly the modal property of a system that is observed during a resonance crossing with its natural frequency changing according to a linear law.

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Chapter 17

Efficient Simulation of Soft Tissue for Human, Robotics, and Exoskeleton Applications



Stefan Holzinger, Andreas Zwölfer, and Daniel Rixen

Abstract This paper explores the simulation of human lower limb dynamics using the floating frame of reference formulation to model bones and soft tissues as flexible bodies discretized with 3D volume finite elements. The study compares joint forces during landing on one leg using different modeling approaches: a purely rigid body model, a rigid body model with lumped rigid wobbling masses, and the aforementioned floating frame of reference formulation-based model. Results demonstrate that wobbling masses reduce joint forces compared to purely rigid body models. Although more computationally demanding than the rigid body models, the floating frame of reference formulation approach offers insight into soft tissue behavior, making it a promising tool for human body dynamics and related applications.

Keywords Multibody · FEM · Human · Robotics · Exoskeletons

Introduction

Human pain and injuries are often caused by loads imposed during movements of the human body especially when high accelerations are present [1]. Loads within the human body are difficult to measure directly, but can be calculated using multibody and finite element simulations [2]. Multibody dynamics refers to the computational methods employed for the dynamic analysis of systems that consist of interconnected rigid and deformable components. Multibody human models conventionally represent human segments as rigid bodies actuated by lumped muscle (e.g. Hill type) force elements [3]. However, bones are covered with muscles and other soft tissues. This so-called wobbling mass represents more than 80% of the total mass of a segment [4], and it has been shown that it cannot be neglected due to its essential contribution to human body dynamics [4]. Modellers interested in the behaviour of wobbling mass often employ the aforementioned rigid/lumped musculoskeletal multibody models to calculate skeletal motion and muscle activation, in order to use this information as an input for computationally expensive non-linear finite element analyses, see e.g., [5]. This one-way coupled simulation approach is tedious from a modelling perspective, computationally intricate, and physically inaccurate when the effects of wobbling masses (inertia and damping) has a significant impact on the overall human motion. The floating frame of reference formulation is a standard method in multibody dynamics and used to model bodies with complex shapes that can undergo large rigid body motions but exhibit only small deformations [6]. An advantage of the floating frame of reference formulation is that it accounts for the nonlinear coupling of small deformations superimposed to arbitrarily large rigid body motion [7] and modal reduction techniques can be applied, reducing the system size to just the included modes [8]. A promising approach, both from the computational and the modelers perspective is the so-called nodal-based floating floating frame of reference formulation, as it eliminates inertia shape integrals entirely, simplifying both the derivation and implementation without needing lumped mass approximations [6, 8]. Although the floating frame of reference formulation is conventionally employed for small deformation problems and linear elastic materials, it still can be used to analyze the dynamics of a human lower limb. This is because bone and muscle make up about 70% of the total weight of a human lower limb [9] and it has been shown that a linear material law is often sufficient to model their mechanical behavior [10]. While the floating frame of reference formulation has already been used to model the human skeletal system [11], to the best of the authors knowledge, it has not been applied for the dynamic analysis of a human lower limb with wobbling masses. In this paper, we demonstrate

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how the floating frame of reference formulation can be used to analyze the combined dynamics of skeleton and wobbling masses during human movement, and examine how wobbling masses affect body motion and joint forces.

Simplified Model Of A Human Lower Limb

As a representative example, we investigate the loads arising in the human knee and hip joint during landing on one leg and compare the results to ones obtained from conventional modeling approaches. The drop height is 0.2 m. For simplicity, we consider in this work a simplified model of a human lower limb, which is illustrated in 1. The considered model consists of six bodies: foot, upper and lower leg bones, hip, one body replacing the entire thigh muscles and one body replacing the entire lower leg muscles. Anthropomorphic data for each body segment is taken from [12, 13]. The thigh bone connects to

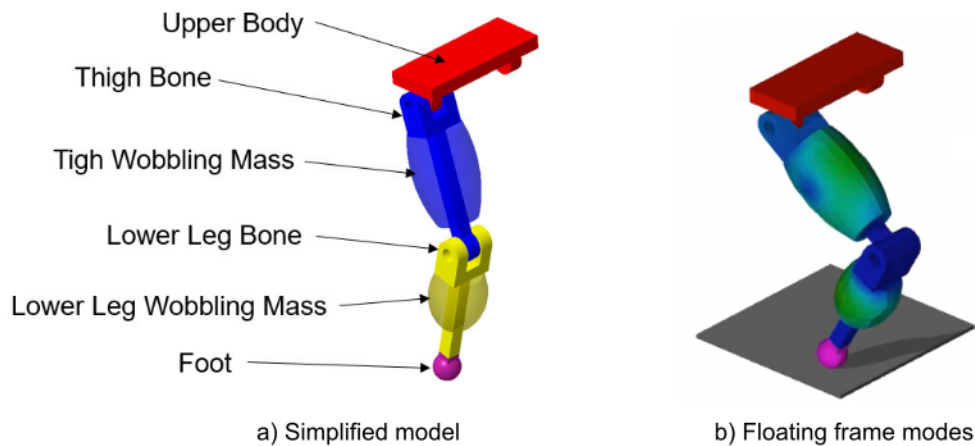


Fig. 1 Simplified model of a human lower limb

the hip and the lower leg bone at the knee, both using springs and dampers forming compliant revolute joints. A torsional spring-damper connects the lower leg and thigh representing the muscle forces acting on both body segments during landing. When using the nodal-based floating frame of reference formulation for modeling, both the lower leg and thigh bone as well as both muscle bodies are considered to be deformable. Component mode synthesis is used for modal reduction.

In the first conventional modeling approach, all system bodies are modeled as rigid bodies. The lower leg (bone and muscle) is modeled as a single rigid body, and similarly, the thigh (bone and muscle) is modeled as another rigid body. In the second conventional approach, all system bodies are considered to be rigid bodies, but in contrast to the first conventional approach, the bodies representing lower leg and thigh muscles are connected to the respective bones using three translational and three rotational springs allowing the muscle rigid bodies to move relative to the bones. All herein considered multibody simulations are performed using the open source C++-based multibody dynamics simulation software exudyn [14]. In all considered modeling approaches, the foot is modeled as a rigid body and is connected to the lower leg using a compliant joint that mimics a rigid joint. Explicit time integration is used to allow efficient contact computation in exudyn. As time integration scheme, we use the explicit Lie group Euler method, cf. [15].

Results

In 2, the joint forces in the knee and hip joint during landing are shown for all three considered modeling approaches. As can be seen in 2, the joint forces obtained from the first conventional modeling approach, denoted with RB, are approximately 28% higher than the joint forces obtained from the second conventional modeling approach (RWM) and the floating frame model (FWM).

In contrast, the peak joint forces obtained from the RWM-model are close to those of the floating frame model. As expected, it turns out that the first conventional modeling approach is computationally the most efficient one, requiring 12 seconds for computing 240000 time steps using a time step size of 1.25×10^{-6} s. The second conventional modeling approach

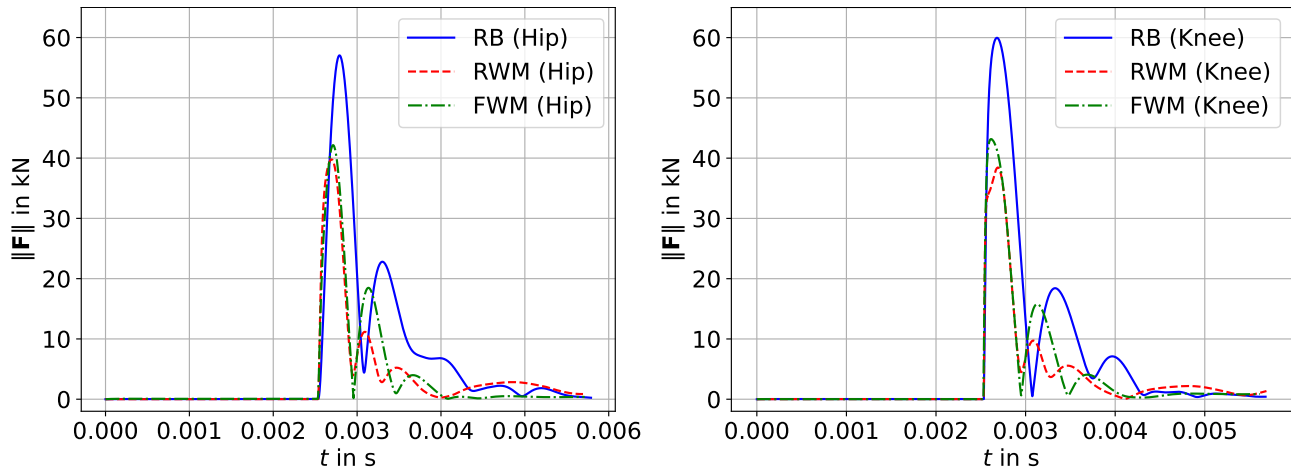


Fig. 2 Joint forces in the knee and hip joint during landing on one leg

required 14.3 seconds of CPU-time for simulation and the floating frame model with 10 modes required approximately 1307 seconds.

Conclusion

Wobbling masses significantly affect joint forces and rigid body motions, potentially reducing joint forces. While being computationally more demanding than the rigid body models, using the floating frame of reference formulation with modal reduction yields a substantial boost in computational efficiency as compared to nonlinear transient finite element simulations. For computing joint forces, a rigid wobbling mass approach may serve as an alternative to the floating frame of reference approach. However, using the floating frame of reference formulation for modeling wobbling masses in multibody dynamics offers substantial advantages, such as eliminating the need to compute inertia tensors from CAD models and enabling direct coupling with bones through finite element methods (FE) without the need for estimating spring stiffness and damping parameters for compliant joints. Additionally, the FFRF approach provides access to stresses and strains, which are not accessible in rigid body models.

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Chapter 18

Piezo Elements to Improve Dynamic Response of Load Cells

M. Vanali, C. Rossi, S. Pavoni, M. Berardengo, and S. Manzoni

Abstract This work addresses the passive reduction of structural vibration of a strain gauge load cell by means of shunted piezoelectric patches. The main goal of this work is the improvement of strain gauge load cells' dynamic behaviour increasing the damping coefficient. Tests have been firstly modelled and analyzed in ABAQUS commercial finite element (FE) code. Four piezoelectric patches were added in the regions of maximum deformation of the load cell to enhance their electromechanical coupling. Their performance is estimated in terms of the so-called electromechanical coupling factor (MEMCF), which is evaluated from forced-vibration analyses under short-circuit and open circuit electrodes of the patches. Piezo patches with similar geometrical and electrical characteristics and connections are so glued to the load cell. The addition of a negative capacitance in series to the resonant shunt shows an enhancement of its performance. The efficiency is evaluated in terms of MEMCF and attenuation of the resonant peaks. Experimental results have good agreement with the model, validating it.

Keywords Load cell · Piezoelectric shunt · Negative capacitance · Damping vibration control

Introduction

Strain gauge load cells are among the most widely employed force transducers in various applications, from industrial scales, for example in automatic weighing instruments, to automotive and aerospace engineering. They are appreciated for their accuracy, versatility and reliability. For example, they are employed in check weighers or weigh-price labellers. Since the performance of these machines is constantly improving, it is crucial to minimise the duration of the weighing process. The latter is linked to the load cell dynamic behaviour, both in terms of frequency response and time to settle after a perturbation is given. Therefore, the improvement of strain gauge load cells dynamic parameters, such as eigenfrequency and damping is a topic of great interest. The higher these parameters are the larger the bandwidth of the transducers will be, and the lesser the settling time will be. During the last years, researchers have focused on different aspects to enhance these performances. In some cases, the focus is on the post-processing of the signal, as in [1] where an adaptive filter is implemented to reduce the measurement time of a load cell, while others focus on the improvement of the bandwidth or the decrease of the settling time modifying the mechanical properties of the component. For example, M. Vanali et al. [2] employed AM techniques with damping materials to enhance the dynamic performance of a load cell by increasing its natural frequency.

In this work, the improvement of the transducer's mechanical properties, especially the damping ratio, is pursued through piezoelectric shunt damping associated with a specifically designed electrical network. This technique relies on the conversion of the mechanical energy to an electric energy that is dissipated through the resistive or a resonant component of a shunting electronic circuit [3]. Thomas et al. [4], [5] and Ducarne et al. [6] demonstrated that once the electric impedance is optimally tuned, the performance of the control depends only on the modal electromechanical coupling factor (MEMCF, k_i) of each mode of the electromechanical system (EMS) (approximating the system to a single degree of freedom system); the

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higher the MEMCF is, the higher the reachable attenuation and added damping is. This parameter can be easily evaluated since it is a function of the natural frequencies of the EMS in short circuit (SC) and open circuit (OC) of the mode considered through the formula:

$$k_i = \sqrt{\frac{\omega_{iOC}^2 - \omega_{iSC}^2}{\omega_{iSC}^2}} \text{ where} \quad (1)$$

ω_{iOC} is the i th natural frequency of the EMS in open circuit and ω_{iSC} is the i th natural frequency of the EMS in short circuit. Higher MEMCF can be obtained by choosing the best geometrical and electrical characteristics of the piezoelectric actuators, the best positioning of them, and adding a synthetic circuit, such as negative capacitances (NCs) to the shunt.

Starting from these bases the dynamic behaviour of the strain gauge cell was modelled and analyzed in ABAQUS commercial finite element (FE) code first.

Finite Element Model

As mentioned before, the strain gauge cell was first modelled using ABAQUS. The transducer understudy is a single-point load cell equivalent to the one produced by Laumas Elettronica S.r.l., FTP75 [7] model (75 kg full scale load). Our prototype is an additive manufacturing load cell produced with Atomic Diffusion Additive Manufacturing technique already studied by S. Pavoni et al. [2]. It is made with stainless steel 17-4 PH, a high resistance and hardness material with an ultimate tensile strength of 1250 MPa, yield strength of 1100 MPa and a tensile modulus of 170 GPa. The stressed areas of the load cell have been designed to maintain the component filled with material due to the thickness of the outer layers whereas the other parts follow a standard triangular 55% infill pattern with a 1 mm thickness. Its layout is reported in Figure 1.

It was found that the load cell's natural frequency increases by about 43% with the AM technique, while the critical damping ratio decreases. So, the dynamic parameters of the load cells are reported in Error. L'origine riferimento non è stata trovata. below. For further details, refer to [2].

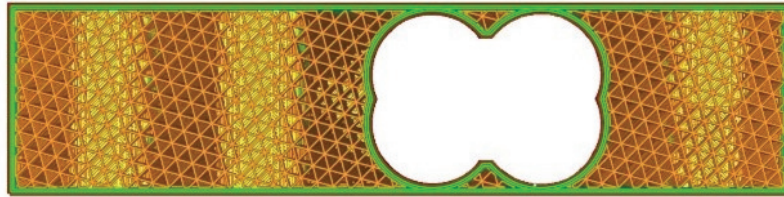


Fig. 1 Layout AM load cell.

Table 1 Dynamic characteristic of the AM load cell

Natural frequency [Hz]	Critical damping ratio (%)
817	0.1

The load cell has been modelled using linear brick elements (8-nodes, C3D8R where R is for reduced integration).

A preliminary FE free vibration analysis has been used to evaluate and compare the eigenfrequency of the first mode of the load cell to the experimental results obtained before [2], and so validate the FE model. The eigenfrequency of the first mode of the modelled load cell was found to be 839.7 Hz, which can be considered to be in good agreement with the experimental one.

Moreover, the FE model has been used to identify the regions of maximum stress and strain in the cell. The piezoelectric patches have to be placed in these areas in order to maximize their electromechanical coupling and, consequently, the damping effect. Figure 2 shows clearly that regions with maximum stress and strain are in the four narrow areas where the central hole leaves a thin thickness with external edges, while Figure 3 shows the level of the strain (ε_{11}) in the upper central area.

Starting from these results, piezoelectric patches of different sizes have been added to the FE model in the regions with the maximum strain. The geometrical configurations have been chosen between those from the catalogue purchased by PI company for the bending piezoceramic sensor. All the patches are made in modified Lead Zirconate-Lead Titanate (identified by the company as PIC151) whose properties can be found on their website [8]

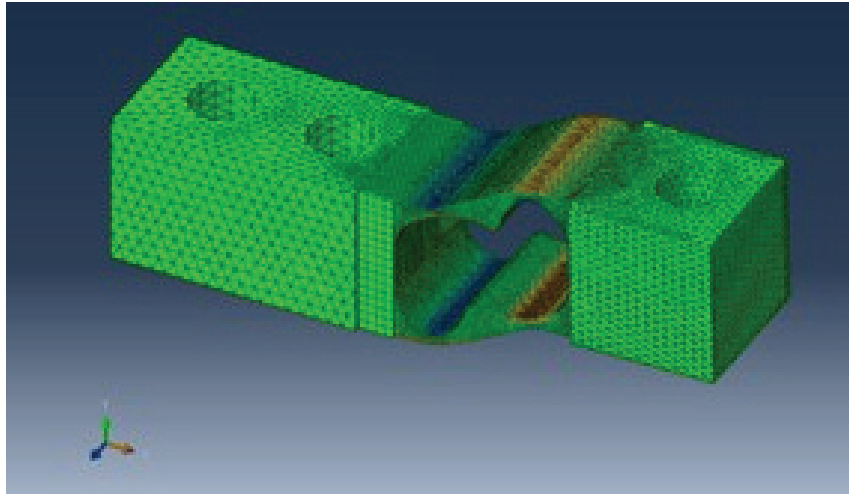


Fig. 2 FE model showing in red and blue the areas with maximum strain ϵ_{11} .

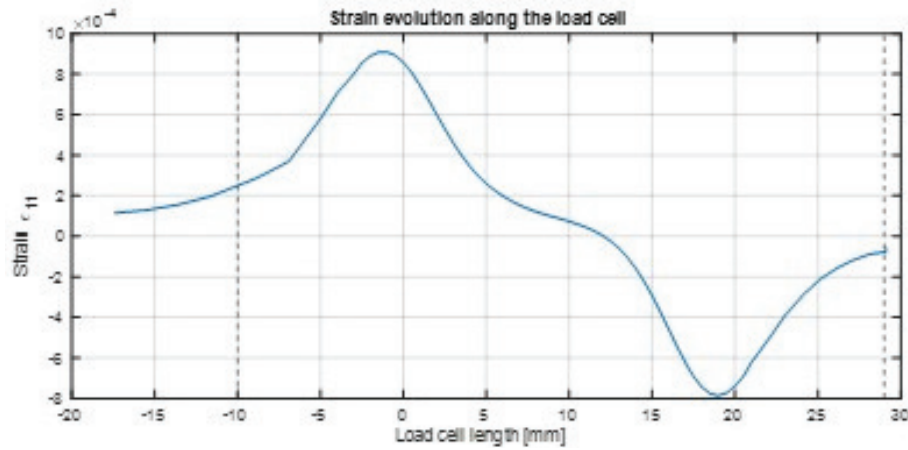


Fig. 3 Strain level along the central area of the load cell.

Every patch has been discretised using quadratic brick elements (20-nodes, C3D20RE where R is for reduced integration and E is for electric). Moreover, since the 4 main stressed areas show a symmetric deformation, two pairs of piezoceramic patches (PZT1-PZT3 and PZT2-PZT4) have been symmetrically bonded as shown in Figure 4

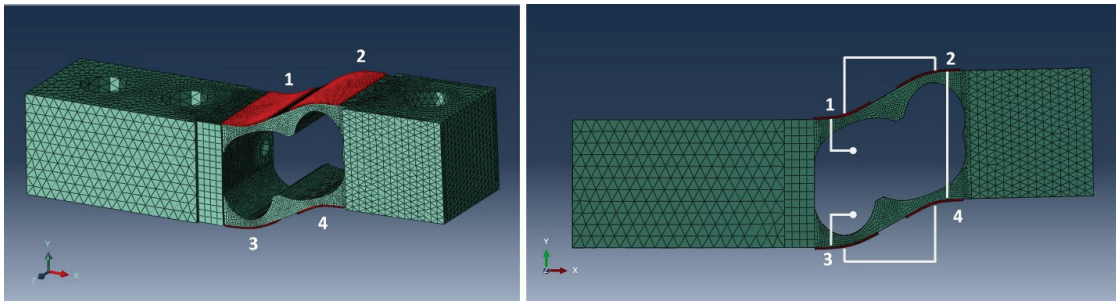


Fig. 4 FEM model with piezoelectric patches and their wiring connection.

The two pairs have the same poling direction, and so require electric parallel wiring. These connections produce opposite fields in the patches, so when the patch on the upper surface expands the one on the bottom surface contracts, creating a downward bending on the load cell. Then the two pairs are connected to each other with electric series wiring. The electric

wiring connection between the patches has been created using the kinematic linear relation constraint on the electric degree of freedom (DOF9) of the external node surfaces of each patch. With these connections, to simulate the SC configuration it is required to apply zero potential to the surfaces of PZT1 and PZT3, while for the OC case one face has zero potential and the other one is charge-free. This solution has been chosen to maximize the attenuation given by the patches. Finally, the contact between the patches and the load cell is defined by Tie Constraint surface contact which ties two separate surfaces together so that there is no relative motion including rotation between them.

Free vibration analyses of the EMS have been performed in SC and OC of the mode being analyzed, to evaluate the MEMCF and thus the best piezoelectric patches geometrical characteristics.

Finite Element Results and Negative Capacitance Implementation

As discussed in the previous paragraph various geometrical layouts of the piezoelectric patches have been simulated. Some of them are reported in Table 2. In all the configurations the length of the patches is 28 mm, as the width of the load cell. The focus was to maximize the coupling factor trying to reduce the height of the patch as possible, in order to reduce its cost.

Table 2 Results obtained from different layouts of piezoelectric patches

Width x Height	Short Circuit Frequency	Open Circuit Frequency	Coupling Factor k
8 mm x 1 mm	957.89 Hz	969.04 Hz	0.15
8 mm x 0.75 mm	925.76 Hz	937.01 Hz	0.16
10 mm x 1 mm	989.65 Hz	1004.3 Hz	0.17
10 mm x 0.75 mm	946.34 Hz	960.34 Hz	0.17
12 mm x 1 mm	1032.4 Hz	1050 Hz	0.18
12 mm x 0.75 mm	983.37 Hz	999.15 Hz	0.18
12 mm x 0.5 mm	932.8 Hz	945.2 Hz	0.16
12 mm x 0.4 mm	912.77 Hz	923.25 Hz	0.15
12 mm x 0.25 mm	885.17 Hz	892.57 Hz	0.13
14 mm x 0.75 mm	999.58 Hz	1017.6 Hz	0.19
14 mm x 0.5 mm	941.1 Hz	954.69 Hz	0.17
14 mm x 0.4 mm	918.53 Hz	929.90 Hz	0.16
14 mm x 0.25 mm	886.6 Hz	894 Hz	0.13

The best solution was found to be the one with the patches of 28x14x0.75mm which ensure a MEMCF equal to 0.19. The most similar patch available in the PI catalogue with wrapped electrodes was 15x15x0.5 mm. The FE simulation of this patch resulted in $\omega_{SC} = 955.34$ Hz, $\omega_{OC} = 955.34$ Hz and $k = 0.17$.

Shunt Circuit

To enhance the performance of the shunt a synthetic circuit such as a negative capacitance (NC) can be added. Since in this case the attenuation has to be done on the first mode of the EMS the best NC layout is given by a series connection [9], [10] as shown in Figure 5.

In this case, an analogous term to the MEMCF, the enhanced modal electromechanical coupling factor (EMEMCF, $\tilde{k}_i$) is introduced, which accounts for the presence of the NC ($\tilde{k}_i > k_i$). This parameter, in a NC series connection, can be evaluated as follows:

$$\tilde{k}_i = \frac{k_i}{\sqrt{1 - \frac{C_{pi}}{C_2}}}, \text{ where} \quad (2)$$

C_{pi} is the piezoelectric patch's capacitance at frequencies above ω_i and C_2 is the equivalent value of the NC.

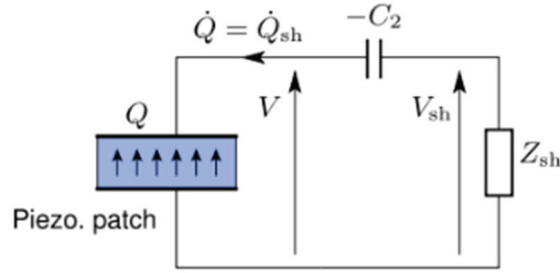


Fig. 5 Negative capacitance electrically connected in series to the piezo patch.

Negative Capacitance Optimisation

For the implementation and optimisation of the NC to be coupled to the circuit, the model developed by Thomas et al., Ducarne et al. and then improved by Berardengo et al. were used as reference [4], [9], [10]. First of all the capacitance of the piezoelectric patch C_{pi} , has to be evaluated.

The capacitance of the piezo patch was measured with a Hewlett-Packard 4284A LCR meter and resulted equal to 7.2 nF. To ensure the stability of the EMS the ideal value of C_2 must satisfy the condition: $C_2 > C_0$, where $C_0 = C_{pi} * (1 + k_i^2)$ for a series connection. The closer C_2 is to C_{pi} , thus to the stability limit, the higher EMEMCF is, and so the damping performance. To clarify further the effectiveness of the NC the index $\beta_2 = C_{pi} / C_2$ is introduced [9]. The EMEMCF $\tilde{k}_i$ enhanced by the NC in series so can be rewritten as:

$$\tilde{k}_i = \frac{k_i}{\sqrt{1 - \beta_2}} \quad (3)$$

Since the NCs do not exist in nature, they must be built synthetically, for example as in this work, through an OP-AMP. The circuit layout used in this case is shown in Figure 6.

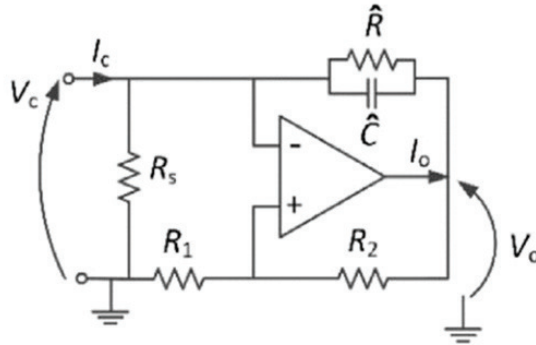


Fig. 6 Electrical layout of the NC implemented.

Notice that an additional resistance $\hat{R}$ is added to solve problems such as bias currents and also a compensation resistance R_s which improves the control performance and makes the circuit more robust to instability [9], [10].

The Optimisation results will be discussed in next section.

Experimental Tests

Starting from the results obtained from the FE simulations, the experimental tests have been carried out. The set-up consisted of the load cell mounted on a 1000 kg mass with two screws with a tightening torque of 40 Nm to avoid the influence of

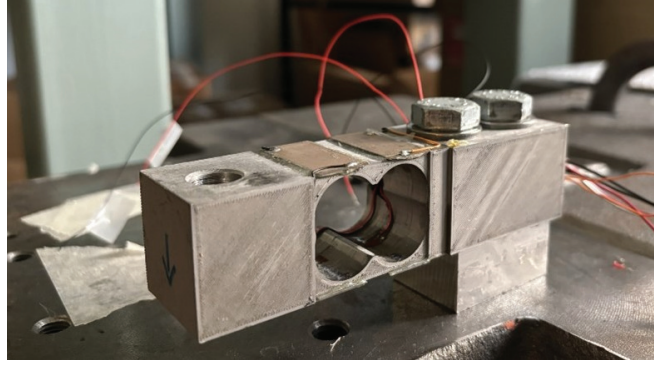


Fig. 7 Prototype of the load cell.

any external dynamics. Four piezoelectric patches with wrapped electrodes were bonded as shown in Figure 7 with an epoxy adhesive.

Unfortunately, the thin layer of adhesive used was not sufficient to electrically insulate the patches from the load cell. For this reason, the electrical circuit was modified as shown in Figure 8.

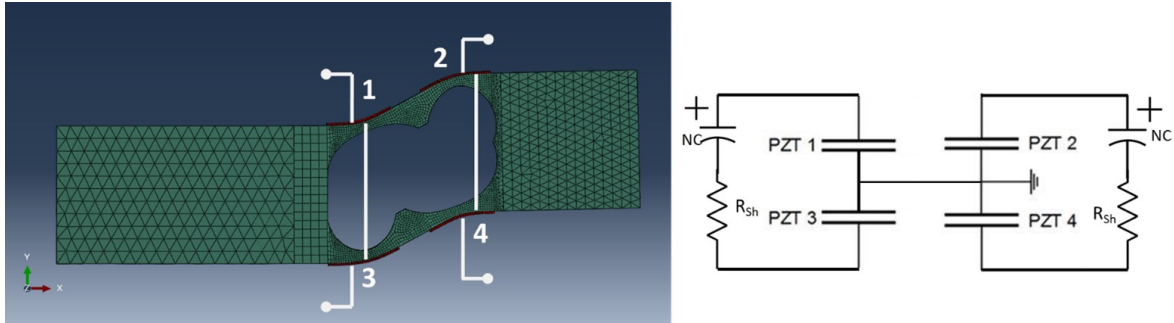


Fig. 8 New wiring connection and electrical layout.

The new layout consists of two pairs of piezoelectric patches (PZT1-PZT3 and PZT2-PZT4) with the same poling electrically connected in series. Then they are connected to the respective shunt circuit composed by the NC and the resistance R_{shunt} . Due to this, FE simulations with this new layout have been performed, to allow a correct construction of the NC.

The results of the two configurations are reported in Table 3 below:

Table 3 FEM results from the new piezoelectric layout

Parameter	PZT13 test	PZT24 test
$\omega_{SC}/(2\pi)$ [Hz]	957.02	958.71
$\omega_{OC}/(2\pi)$ [Hz]	964.38	964.38
k	0.124	0.109

In order to experimentally evaluate the dynamic parameters the load cell was excited with an impulsive method using an impact hammer. The FRF has been estimated with the H_1 estimator as

$$H_1 = \frac{G_{xy}(f)}{G_{xx}(f)}, \text{ where}$$

$G_{xy}(f)$ is the cross spectrum between the load cell response and the hammer's one, and $G_{xx}(f)$ is the power spectrum of the hammer.

Being the system response basically a 1 DOF one, the eigenfrequencies and non-dimensional damping ratios have been evaluated using the peak picking method and the half power method. The k_i and $\tilde{k}_i$ values have been estimated by measuring the natural frequencies with the piezoelectric patches in SC, OC and with the working NC using the equation (1) and (2).

Finally, the optimal value for R_{shunt} to be connected in series to the NC was computed considering that:

$$R_{shunt} = \frac{\tau_e * (1 - \beta_2)}{C_{pi}}, \tau_e = \frac{1}{\omega_F} \text{ with } \omega_F = \sqrt{\frac{\omega_{iOC}^2 + \omega_{iSC}^2}{2}} \quad (4)$$

The NC has been built using Texas Instruments OPA445 OP-AMPs suiting the electrical scheme shown in Figure 6. The OP-AMPs were supplied with a direct current (DC) voltage of ± 45 V. The values of the different electric elements are reported in Table 4.

Table 4 Values of the parameters for the NC construction

Parameter	Value
R_1 [k Ω]	9.262
R_2 [Ω]	9.906
$\hat{R}$ [M Ω]	0.800
$\tilde{R}$ [M Ω]	0.748
R_s [Ω]	0.756
$\hat{C}$ [nF]	7.480
C_2 [nF]	10.133
β_2	0.75

In the next section, the results of the experimental test will be discussed. Notice that, when one couple of piezoelectric patches was analysed, the other couple was in an OC condition. To be clearer, results will be presented as PZT13 for the couple PZT1-PZT3 and PZT24 for the couple PZT2-PZT4, but the test procedure is the same.

Experimental Results

The natural frequencies of the load cell in SC and OC conditions are first estimated. After that, the NC is added in series to the circuit inducing a shift of the SC frequency to the left, and so, a higher coupling factor. Starting from these results the optimal value of R_{shunt} can be evaluated with 4. The same procedure was used during the test PZT24. The critical damping has been evaluated in the SC configuration and then with the R NC.

In Figure 9 the magnitude of the FRFs in the various electrical conditions for the two tests are reported. It is possible to notice that the shunt circuit implemented provides a good attenuation of the resonance peak to which it is tuned.

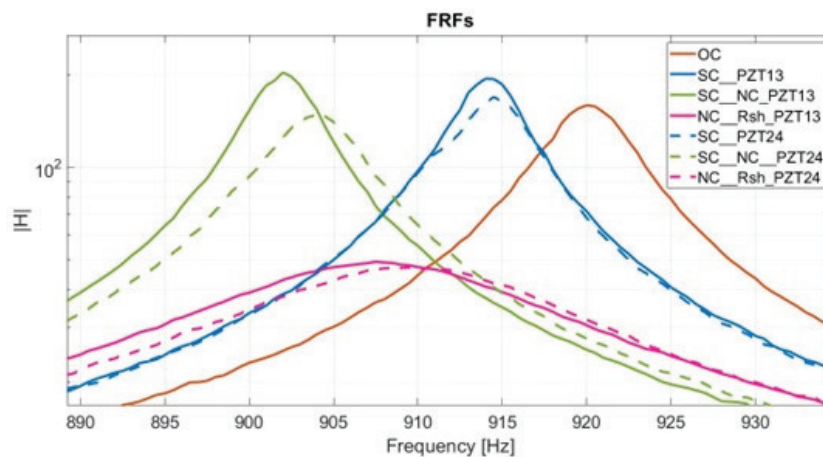


Fig. 9 RFs of the EMS obtained in different configurations (OC Open Circuit, SC Short circuit, NC Negative capacitance connected, NC-Rsh Negative capacitance with the resistive shunt connected).

Natural frequencies and damping ratios are reported in Table 5.

Table 5 Experimental results.

Parameter	Symbol	PZT13 test	PZT24 test
Natural Frequency in Short Circuit	$\omega_{SC}/(2\pi)$ [Hz]	914	915
Natural Frequency in Open Circuit	$\omega_{OC}/(2\pi)$ [Hz]	920	920
Modal electromechanical coupling factor	k	0.115	0.110
Natural Frequency in Short Circuit with NC connected	$\widehat{\omega_{SC}}/(2\pi)$ [Hz]	902	904
Natural Frequency in Short Circuit with NC and R_{shunt} connected	$\omega_{NCRSh}/(2\pi)$ [Hz]	909	910
Enhanced modal electromechanical coupling factor	$\tilde{k}$	0.201	0.189
Critical damping ratio in Short Circuit	ξ_{SC} (%)	0.27	0.29
Critical damping ratio with RNC connected	ξ_{RNC} (%)	1.29	1.12

All the tests have been re-done with the piezoelectric patches couple not investigated in SC instead of OC, but since they do not show significant variations compared to the results already obtained, they will not be reported. Moreover, comparing the results of the two tests it is noticeable that they are quite similar, confirming the symmetry of the EMS.

The addition of the Resistive Negative Capacitance shunt circuit results in the increment of the damping ratio ξ and of the natural frequency as shown in **Errore. L'origine riferimento non è stata trovata.**

Table 6 Natural frequencies and critical damping ratios comparison

	Natural frequency [Hz]	Critical damping ratio (%)
Additive Manufacturing	817	0.1
Additive Manufacturing and piezo in SC	914	0.27
Additive Manufacturing with shunt circuit	910	1.3

The improvement of the dynamic characteristic of the load cell can be established by both evaluating its bandwidth and settling time. In order to estimate the bandwidth, a single degree of freedom model was fitted with a least square method to experimental FRF. The bandwidth of transducers is computed starting from the magnitude value of the reconstructed FRF at 0 Hz, which represents the static sensitivity, with tolerance ranges equal to 1%, 5% and 10%. The reconstructed FRF is depicted in **Errore. L'origine riferimento non è stata trovata.**

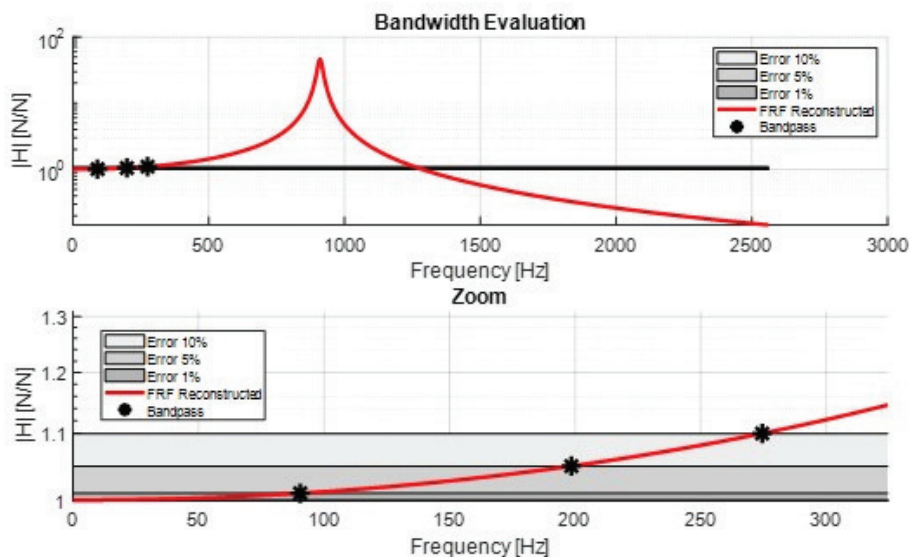


Fig. 10 Bandwidth evaluation for the load cell with shunt circuit connected.

Finally, the bandwidth was compared to the resulting one of the load cell without the shunt circuit, in order to evaluate the possible enhancement of the dynamic characteristics. Results are compared in Table 7.

Table 7 Bandwidth estimated from load cells

Tolerance	High Cutoff Frequencies [Hz]		
	1%	5%	10%
Additive Manufacturing	81	178	246
Additive Manufacturing with shunt circuit	91	199	275

Results show an increase in the bandwidth thanks to the shunt circuit of 10 Hz for 1% error, 21 Hz for 5% error and 31 Hz for 10% respectively. Thus, an average growth in bandwidth by about 12% is led by the shunt circuit.

Moreover the settling time of the load cell has been improved in the order of 1:10 as can be seen in the following graph (Figure 11).

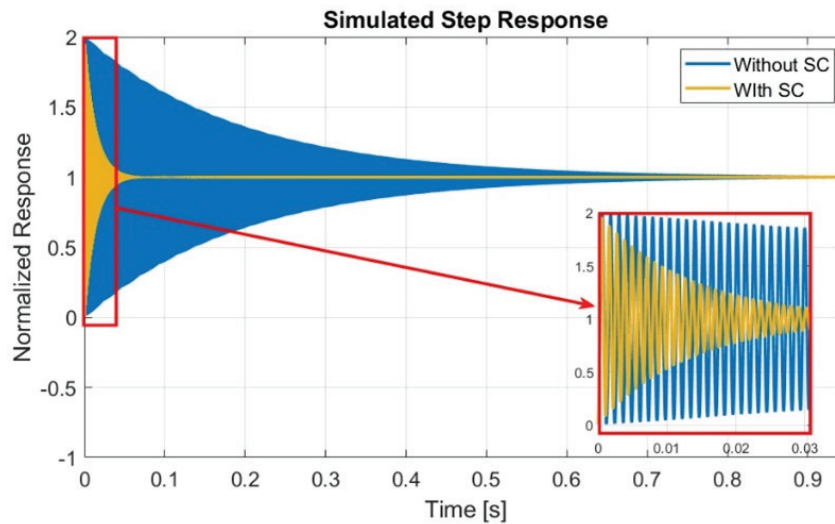


Fig. 11 Load cell settling time (Step response).

This improved settling time is meaningful in all those applications like filling machines where the step response is one of the driving parameters, and may lead to faster and more accurate measurements.

Conclusion

The aim of this work has been to enhance the dynamic characteristics of a single point load cell made by Additive Manufacturing through the implementation of piezoelectric shunts, precisely a resistive shunt coupled to negative capacitance. The FEM of the transducers was first developed in Abaqus software, to identify the areas with maximum stress and strain. Then, different piezoelectric patch layouts were simulated, with different sizes and electrical connections. The best configuration is the one which ensures the highest electromechanical coupling factor k . Starting from the numerical results, experimental tests have been carried out. To optimize the R NC layout the FRFs of the EMS in different electrical configurations have been implemented. Natural frequencies, MEMCFS, EMEMCFS and damping ratio have been evaluated. Experimental results are in good agreement with the numerical and theoretical ones. The damping ratio resulted in an enhancement from 0.27% to 1.3%. The resonance frequency increases too, from 817 Hz to 910 Hz. Finally, the load cell's bandwidth with the shunt circuit connected was evaluated. Bandwidth results prove an enhancement of dynamic characteristics, showing a 12% increment in bandwidth frequencies. Settling time was reduced in the order of 1:10, being this a real improvement in the cell performance.

Due to how the patches are bonded to the transducers, the two R NC shunts work separately and one at a time. So, in future works, the two R NC shunts working simultaneously will be tested. Finally, the patches will be rebounded to the transducers and the single electrical layout originally designed will be examined.

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Chapter 19

Monitoring the Interface Behaviour of Bolted Joints

T. Mace, S. Gilbert, T. Moghadam, and P. Pandey

Abstract Validating the localized nonlinear contact behaviour of a jointed structure is a challenging task. This is because monitoring the interface during a vibration cycle, to identify stick, slip and separation zones, without changing the interface, is very difficult. Total Internal Reflection has recently been used at Imperial College London to measure the interface conditions of simple, well controlled joints, where two transparent interfaces were illuminated via a laser and the reflected/transmitted light from the interface was recorded. The change in reflection/transmission can reveal localised contact zones and show their transition from stick to slip during a vibration cycle. This work applies the Total Internal Reflection (TIR) technique to more realistic lap-jointed structures. The structures investigated were based on the well-investigated Brake-Reu β beam but manufactured from transparent PMMA to facilitate the novel testing approach. Bolt loads and excitation were well controlled via the use of instrumented bolts, and rigorous modal testing, and TIR measured using a high-speed camera and expanded laser beam. For the first time, this has allowed for the production of a fuller picture of the contact conditions during the vibration cycle of a realistic jointed structure.

Keywords Optical · Contact · Joint · Vibration · Interface

Introduction

In most large engineering structures, such as turbomachinery and load bearing structures, elements of the structure will be fastened together via bolted joints. Here, an interface, consisting of two contact surfaces and one or more bolts exists, which drastically alters the vibrational characteristics of the system. Vibration results in High Cycle Fatigue (HCF), due to high vibration amplitudes and intense cyclic stresses. The friction in such joints provides a certain amount of damping to the structure, reducing vibration amplitudes, but the behaviour of the joints is unpredictable because the vibration response is often highly nonlinear, with the damping and resonances varying significantly with vibration amplitude [1]. There nevertheless exists the fact that it is very difficult to physically measure the contact conditions inside the contact interface during a vibration cycle without physically altering the interface, limiting the physical understanding that underpins the simulations, which must instead utilise smeared contact stiffness parameters obtained experimentally for an entire interface contact, without the consideration of local behaviour. There have been extensive attempts to improve upon the understanding of the contact interface behaviour using experimental techniques. These studies have typically involved high-speed cameras and Digital Image Correlation (DIC), which allows for the observation of the contact along the edges of an interface to impressive levels of detail, but still not within the interface [2]. Other novel measurement techniques have involved the use of two aligned LDVs on either side of a flange-like interface, but have been limited in spatial resolution[3]. Furthermore, alternative approaches

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using Acoustic Emission (AE) [4] are somewhat limited in measuring the stick-slip transition of an interface due to the loss of the AE signal to the noise floor of the experiment for certain test specimens and configurations. By comparison, a promising technique utilises Total Internal Reflection (TIR), whereby transparent specimens are used, and the contact interface is illuminated with an expanded laser beam. High-speed camera measurements can subsequently capture the change in luminance across the interface, with different greyscale values attributed to areas that are in full contact, slip and separation. This technique has been successfully used, in combination with hysteresis loop measurements, to assess the affect of frictional forces upon the real area of contact of a small, sliding interface [5]. The study details the application of this novel approach upon a larger, bolted interface.

Materials and Methods

The structure chosen for the application of TIR is the well-documented Brake-Reu β Beam (BRB), consisting of a lap joint with three bolts tightened to equal torque levels [6]. Unlike all BRBs thus far, the two structures examined were manufactured from cast PMMA, which was milled to produce the lap jointed region, but is otherwise of the same dimensions as a standard BRB. A monolithic beam, of the same dimensions but with no lap joint, was used to benchmark the effects of the test setup. Instrumented bolts with load calibrated strain gauges [7], were used to provide an accurate and consistent bolt tension of 3000N across the three bolts, improving upon the usual error margins of a torque wrench. A back-of-the-envelope calculation estimated a plastic limit torque of approximately 10Nm for the M8 bolts and PMMA. A laser source (Coherent – VLM 0.6mW 635nm) was connected to a laser beam expander (Edmund Optics 20x VIS-NIR Vega BB) with a 3D printed adaptor to increase the area of illumination and mounted at approximately 60° to the contact interface. A Photron Mini-UX100 high-speed camera, running at 4000FPS was placed colinearly with the laser, to capture the illuminated contact area. The vibratory motion of the beam, due to shaker excitation, was removed using the Photron PFV software's in-built stabilization algorithm. The technique takes advantage of the principle that the transmitted light intensity across each pixel is a proportional to the real area of contact, since at areas which are not in contact, light will be reflected, and at areas in contact, light will be transmitted directly through the interface. In reality, the resolution of each pixel is larger than that of a micro-contact region, so the effective contact area is assessed through the light intensity, rather than binary, 'light or dark' values. A DataPhysics Quattro and Shaker, with a PCB Accelerometer (352A21) and Dytran load cell (1051V2) were used to measure and control the vibration of the structure at three excitation levels (1N, 5N and 10N), and a manual trigger connected to the force channel of the DAQ and high-speed camera allowed for tachio-syncing of the two data streams allowing the position of each captured image within the vibration cycle to be tracked. The test specimens (monolithic, cast-jointed and 3D-printed) were suspended by a light, inextensible fishing wire through the nodal locations of the first bending mode, to minimise damping from the boundary conditions. The shaker and accelerometer were attached at the location marked in Figure 1b and the camera-laser setup directed to the region at the end of the lap joint, where motion is most expected to occur, since the expanded laser beam was not large enough to illuminate the entire length of the contact interface. Both in-plane (IP) configurations, with the

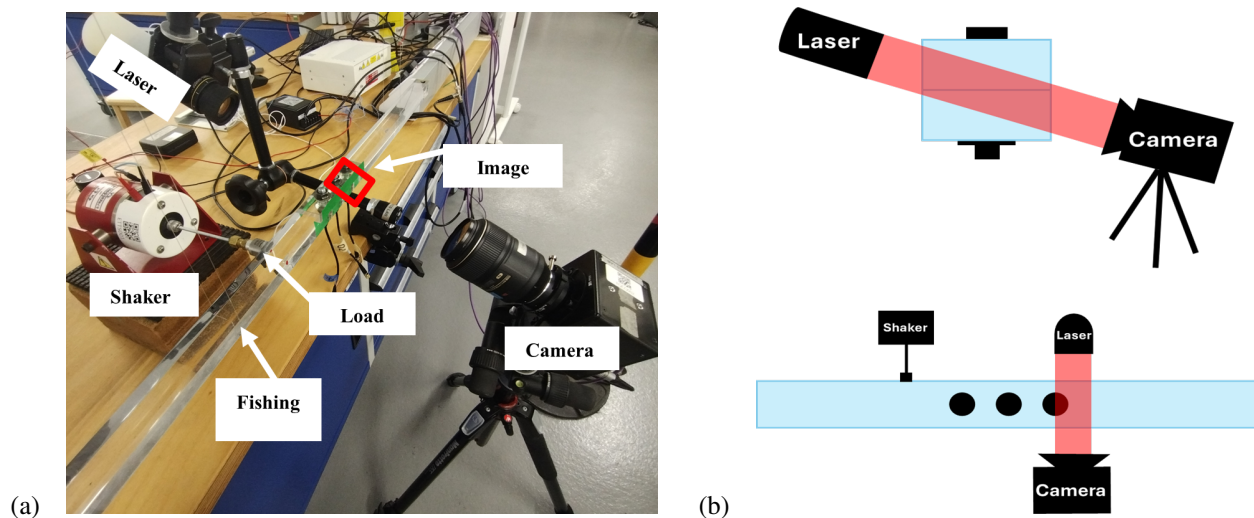


Fig. 1 Physical Test Setup (a) and Schematic (b) Detailing the Approximate Region of Interest.

bending motion in the plane of the contact interface, and out-of-plane (OP) configurations, with the beam rotated through 90° along its length, were examined for each test case. The first three bending modes were examined for each test configuration.

Results

Initially, roving impact tests on the monolithic beam and all BRB specimens allowed for the determination of the natural frequencies of each mode. Inherent material damping of the first mode found to be approximately 3% for the monolithic beam. Stepped-sine measurements of the monolithic beam at three different forcing levels (Figure 2) showed little significant shift in the natural frequency or damping of the first two modes. As such there is only marginal nonlinearity due to the viscoelastic material or shaker test set-up at forcing levels of up to 10N. Conversely, the BRB specimens showed more significant nonlinear behaviour at higher forcing amplitudes, which can likely be attributed to the frictional bolted joint behaviour, with a 2%, 0.5% and 1% shift in frequency of the first three out-of-plane bending modes respectively. This alludes to potential micro-slip in the contact interface and allows for further investigation with the novel optical technique.

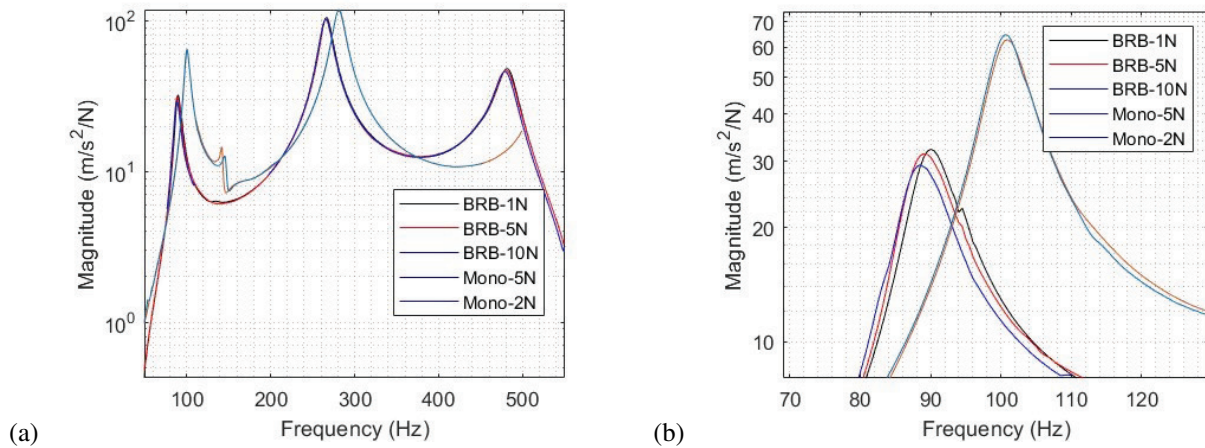


Fig. 2 Comparison of Nonlinearity for the Monolithic/PMMA BRB for the 1st 3 modes (a) and a Close-Up of the 1st Mode (b).

Clear images and high-speed videos were captured of the ‘end’ region of the BRB contact interface. As a result of the footage stabilization, the laser beam moves relative to the laser and the laser has a inconsistent intensity distribution with multiple visible ring patterns. This can be observed across the vibration cycle in the captured images, reducing the ease with which the contact behaviour can be assessed.

From Figure 3 the change in real area of contact can be observed, and this increases with a higher excitation load as might be expected. Additionally, it can be observed that the average contact area reduces as the excitation load increases. Whilst this suggests that the higher amplitude may results in greater slip, and therefore a reduced apparent contact, this may be an artifact of relative motion between the specimen and the illumination field, so requires further investigation.

Figure 4 demonstrates the ability of the test technique to capture the change in the real contact area across the vibration cycle. Both the IP and OP cases show an increase in the change in contact area across a vibration cycle, but in a different manner. The IP case clearly shows the maximum contact to exist in the middle of the cycle at the point of minimum displacement/acceleration, whereas the OP configuration shows the contact area to increase and decrease at either end of the cycle, showing the opening and closing of the interface as the specimen is vibrated out-of-plane with the contact interface. The overall change in contact area is greater for the OP configuration than the IP configuration (seen in Fig. 5a), presumably due to this mode of motion, in comparison to the shearing motion of the IP cases. Furthermore, Figure 5b highlights the regions of the interface that show the greatest change in contact area, showcasing the opening/closing region for the OP 10N 1st bending mode, and providing insight into the physical change of the interface for this mode, information that could be used to inform more representative modelling.

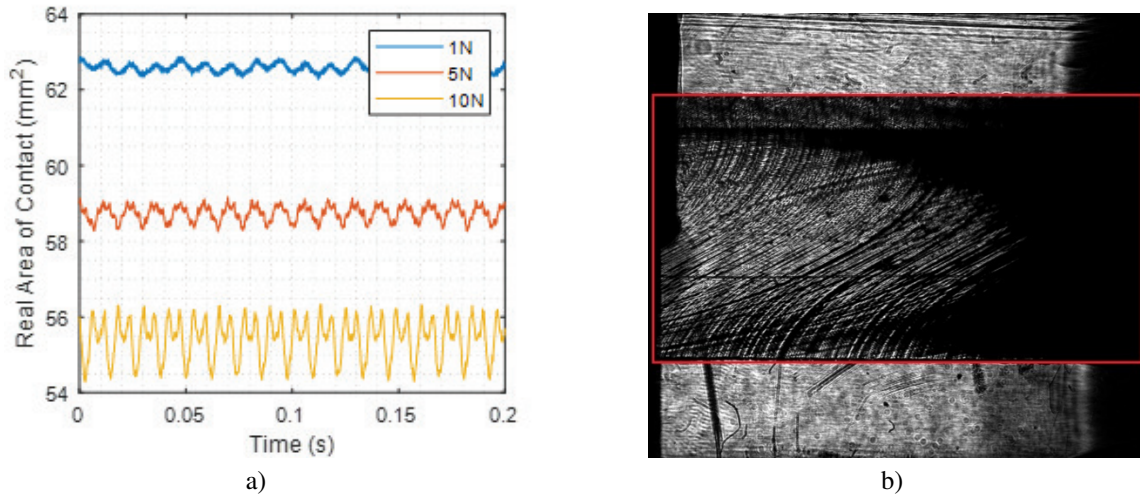


Fig. 3 Capture of the change in contact area for several harmonic excitation loads (a), and image of the illuminated contact interface (highlighted in red) (b).

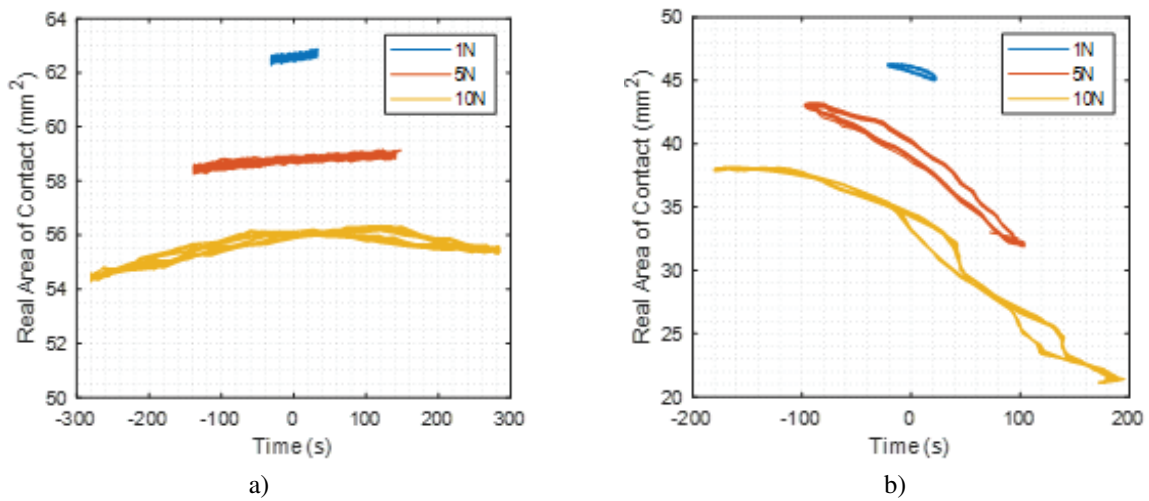


Fig. 4 Change in Real Contact Area with Respect to Acceleration for a) the IP case and b) the OP case.

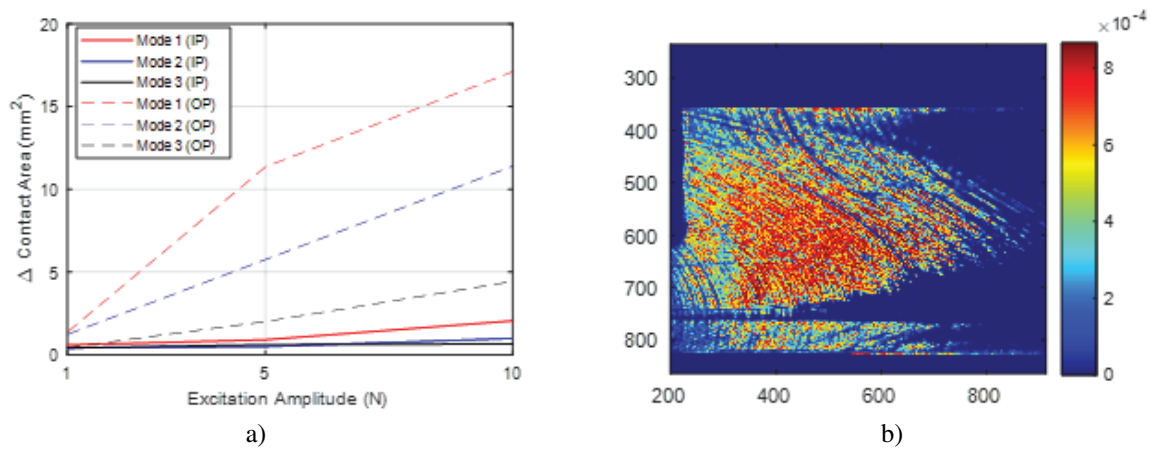


Fig. 5 a) Relationship between Excitation Amplitude and Change in Contact Area across 3 modes, both IP and OP b) the Distribution of Contact Area (mm²) Change for the 1st Bending Mode of the OP Configuration, at 10N Exc. Amplitude.

Conclusion

The TIR measurement technique has demonstrated great potential for the further analysis of bolted joints, allowing for the study of the changes of contact area in the contact interface. Whilst further test refinement and analyses are required to explore the full capabilities and limitations of the test technique, the technique has allowed for the observation of expected relationships, providing confidence in the data produced, as well as producing a novel insight into the change in contact area across a vibration cycle which has been thus far unobserved in literature without the use of supplementary material in the interface. As such, this study has highlighted the potential for the further development and refinement.

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