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POLYTECHNIQUE MONTRÉAL

affiliée à l'Université de Montréal

**Design of a Vertical Axis Rotating Machine for the Development of a Digital
Twin**

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Département de génie mécanique

Mémoire présenté en vue de l'obtention du diplôme de *Maîtrise ès sciences appliquées*
Génie mécanique

Août 2023

POLYTECHNIQUE MONTRÉAL

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présenté par **Souheil SERROUD**

en vue de l'obtention du diplôme de *Maîtrise ès sciences appliquées*
a été dûment accepté par le jury d'examen constitué de :

David MÉLANÇON, président

Frédéric GOSSELIN, membre et directeur de recherche

Oguzhan TUYSUZ, membre

DEDICATION

To my family who supported me unconditionally. To my friends, who push me to move forward. And finally, to myself, because why not.

À ma famille qui m'a supporté inconditionnellement. À mes amis, qui me poussent à avancer. Et finalement à moi, car pourquoi pas.

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I am particularly thankful for his consistent availability and the time he devoted to us, despite his demanding schedule. Professor Gosselin's unerring demands for high standards in our work have been a driving force, instilling in me a deep-rooted appreciation for academic excellence and rigor. His unique ability to push us towards perfection without compromising our intellectual curiosity and love for learning has been the epitome of effective mentorship.

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To my esteemed colleagues, Sima Rishmawi, Mohammed Abda, Maryam Boukor, and Lucas Berthet, I owe a debt of gratitude. The camaraderie we shared in the offices of Polytechnique Montreal made the journey enjoyable and fruitful. The positive environment they created not only fostered my academic growth but also fostered a sense of friendship that I will cherish for years to come. I consider myself fortunate to have worked alongside such outstanding individuals.

Lastly, I extend my sincere thanks to Hong Yan Miao, Benedict Besner, and Fabrice Danet. Their technical expertise and unwavering support over the last two years have been indispensable. Without them, the project would not have reached its current state of advancement. Their dedicated efforts have truly made a world of difference.

RÉSUMÉ

Cette étude aborde le besoin croissant de production d'énergie fiable et durable, en mettant l'accent sur l'énergie hydroélectrique. Alors que la population mondiale et les besoins en énergie augmentent de manière exponentielle, l'impact environnemental des combustibles fossiles traditionnels devient de plus en plus problématique et irrémédiable. L'énergie hydroélectrique, notamment dans des régions comme le Québec, au Canada, où elle représente 94% de la production d'électricité, offre une alternative plus propre. Cependant, le développement de nouveaux projets hydroélectriques est confronté à des défis liés aux préoccupations environnementales, à l'opposition du public et aux coûts initiaux élevés, tandis que la conservation des unités déjà existantes fait face à une demande croissante et à des opérations irrégulières de plus en plus importantes.

Ce projet vise à contribuer au développement d'un jumeau numérique (DT) d'une turbine hydroélectrique, en commençant par la conception, la construction et l'instrumentation d'une machine rotative à axe vertical (VARM). Cette machine sert de source de données empiriques pour les chercheurs travaillant sur les premières versions de modèles de DT de turbine hydroélectrique. La VARM fournit des données d'entraînement et de validation pour le développement de tels modèles en simulant des modes de défaillance contrôlés en laboratoire.

La VARM est conçue pour être modulaire et flexible, capable de fonctionner avec différents matériaux, arbres, rotors et paliers pour simuler de nombreux cas d'essais différents. Sa conception est d'abord basée sur le modèle de machine tournante de Jeffcott qui offre un équilibre entre simplicité et précision. Elle aussi est accompagnée d'un modèle par éléments finis (FEM) offrant une source de données de référence régie par des paramètres connus servant à l'entraînement précoce du DT et la comparaison avec des données empiriques enregistrées sur la VARM.

La conception et la construction complétées de la VARM sont ensuite accompagnées d'une série de tests préliminaires sous différentes configurations et cas de chargements. Des paramètres clés du système tels que la fréquence naturelle et le taux d'amortissement sont estimés par le biais de diverses techniques pour plusieurs cas. Le comportement de la VARM est également comparé aux résultats de simulation du modèle d'éléments finis permettant d'identifier un certain type de défaillance grâce à sa simplicité. De nombreuses améliorations pourraient encore être apportées à la VARM afin d'améliorer ses performances et ses capacités d'acquisition et de traitement de données, tout en offrant déjà une large gamme de mesures.

ABSTRACT

This study addresses the increasing need for reliable and sustainable energy production, with a focus on hydroelectric power. As the world's population and energy needs grow exponentially, the environmental impact of traditional fossil fuels becomes increasingly problematic and irremediable. Hydroelectric power, particularly in regions like Quebec, Canada, where it accounts for 94% of electricity generation, offers a cleaner alternative. However, the development of new hydroelectric projects is confronted with challenges due to environmental concerns, public opposition, and high upfront capital costs while the conservation of already existing units faces an evermore increasing demand and irregular operations.

This project aims to contribute to the development of a digital twin (DT) of a hydroelectric turbine, starting with the design, construction, and instrumentation of a Vertical-Axis Rotating Machine (VARM). This machine serves as an empirical data source for researchers working on early versions of hydroelectric turbine DT models. The VARM provides training and validation data for the development of such models by simulating lab-induced failure modes similar to those found in actual hydroelectric turbines and rotating machinery.

The VARM is designed to be modular and flexible, capable of operating under different conditions, shafts, rotors, and bearings to simulate numerous test cases. Its design is initially based on the Jeffcott rotating model, which provides a balance between simplicity and accuracy. It is then accompanied by a finite-element model (FEM) offering an accurate and versatile source of reference data governed by known parameters for early training of the DT and comparison with the empirical data recorded on the VARM.

The successful design and construction of the VARM, which will contribute to future researchers in need of experimental data of a vertical-axis rotating machine, is then accompanied by a series of preliminary tests under different configurations and loading schemes. Key parameters such as natural frequency and damping ratio are estimated through a variety of techniques for multiple cases. The behavior of the VARM is also compared to the simulation results from the finite-element model, which, in its simplicity allows the identification of a certain type of failure. Many improvements could still be applied to the VARM to improve its performance and data acquisition and treatment capabilities while still already offering a wide array of measurements.

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LIST OF SYMBOLS AND ACRONYMS

DT	Digital-twin
VARM	Vertical-axis rotating machine
AI	Artificial intelligence
IoT	Internet of things
PHM	Prognosis and health management
FEM	Finite-element model
FEA	Finite-element analysis
AMD	Active magnetic damper
MCSA	Motor current signature
MBD	Multi-body dynamics
CFD	Computational fluid dynamics

A	Amplitude
C	Damping coefficient
D	Diameter
E	Young's modulus
e	Eccentricity
F	Loading force
g	Gravity constant
I	Moment of inertia
K	Stiffness coefficient
L	Beam/Shaft effective length
M	System mass
m	Unbalance mass
T	Torque
t	Time
W	Weight

x	Lateral displacement
$\ddot{x}$	Lateral acceleration
z	Longitudinal position along the shaft
α	Non-dimensional acceleration
β_n	Modal coefficient
Δ	Non-dimensional displacement amplitude
δ	Non-dimensional displacement
ζ	Damping ratio
ρ	Mass density
σ_u	Ultimate strength
τ	Non-dimensional time
Ω	Non-dimensional speed
ω_e	Rotation speed
ω_n	Natural frequency

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CHAPTER 1 INTRODUCTION

In every and all machinery, the most critical performance aspect is arguably its operational reliability. Even for single use machines like F1 engines, the operator or driver, must be confident that the machine will run optimally throughout the whole designed operational duration. Otherwise, unplanned shutdowns of operations due to un-predicted failures are likely to amount to significant headaches for all users and personnel operating and maintaining the machine. These unexpected failures will prove to be significantly more expensive than for regular scheduled maintenance as they bring about a full or partial stop of operations which often results in a loss of revenue.

To effectively schedule a maintenance on a machine part, one must be in possession of important knowledge about a specific system's failure modes. This is necessary in order to correctly estimate the appropriate level of damage, and optimal course of action for repairs but most importantly, the damage frequency as to not over- or underestimate the maintenance rate (Bloch and Geitner, 1997). Miscalculations in the maintenance strategy can bring about heavy financial losses for its owner. This is all the more amplified when dealing with machines on which whole communities rely for power generation. Hydroelectric turbines are not immune to these technical and logistic problems. Unexpected failures on machines of this scale can prove to be critical for operations and hazardous for workers and communities. Thus, proper maintenance and failure prediction is imperative for continuous optimal operations.

No two hydroelectric turbines are identical as their specifications are custom designed for distinct operational conditions (Bideris-Davos and Vovos, 2023). However, they still exhibit similarities in their main architecture as they all require some basic components to be able to fulfill their function. Fig.1.1 presents the main components of a Francis turbine which is the most common type hydroelectric turbine in use today from which we identify critical components that are present in almost every kind of turbine. These include a generator to convert mechanical energy into electricity, a turbine to produce rotation and torque from the flowing fluid, a shaft to transfer the turbine's torque to the generator and a combination of bearings to support all of these rotating parts. HydroLeader (2023) revealed in an interview with Canyon Hydros vice president of engineering, Brett Bauer, that net head and water flow rate are the two main variables driving the design process. Moreover, decades of operations under different conditions for each individual unit has made it unique in the way fatigue and wear developed over time. Physical analytical and numerical models used to understand the

behavior of such units are not designed to take this history into account and are by nature imperfect as capturing the complete physics of a system can be very challenging. A computer model of a complex system, like a hydroelectric turbine, based on physics phenomena formulations as well as historical data could potentially help bridge the gap between reality and simulation. We refer to such a model as a Digital Twin (DT) (Jones et al., 2020). By being informed about the physics of a system as well as empirical data from in-site live sensors monitoring the machine, the DT could help flag abnormal system behaviors and predict failures.

A reliable and repeatable source of empirical data is essential in order to learn and navigate the intricacies that come with the design a DT model. This is where the vertical-axis rotating machine (VARM) will prove valuable. To streamline the design process of a hydroelectric turbine DT, a model experiment exhibiting similar dynamics would facilitate the DT scaling to a real-world running turbine. The VARM is a rotor-bearing system vertically oriented and designed to produce known rotor dynamics failure modes like unbalances and misalignments, but also hard-to-model non-linearities that arise from the bearings orientation. This assortment of known and unknown parameters makes it an excellent model experiment for a DT where part of the dynamics can be analytically modeled by conventional theories and the other part by machine-learning techniques combining physics based knowledge and sensors data.

This introduction will be continued by the motivations behind the development of a hydroelectric turbine DT, the scope of work of this project, the contribution to research and the thesis outline.

1.1 Motivations

1.1.1 The evolving place of hydro-power in energy production

As time passes, climate change and environmental degradation of our planet becomes more and more problematic and apparent as our energy needs grow exponentially. The vast majority of civilisation’s energy needs are fulfilled by fossil fuels: 84% of the world energy needs is satisfied by coal, oil, and natural gas alone (Ritchie et al., 2022). Evidently, regional power generation means are unique to regions and countries as available natural resources differ geographically. In Quebec, Canada, about 94% of our electricity generation comes from hydro-power units which puts it in the top world hydroelectricity producers (Ritchie et al., 2022). This kind of energy is clean in the sense that it does not produces significant greenhouse gases during operation. However, the development of new hydro-power projects

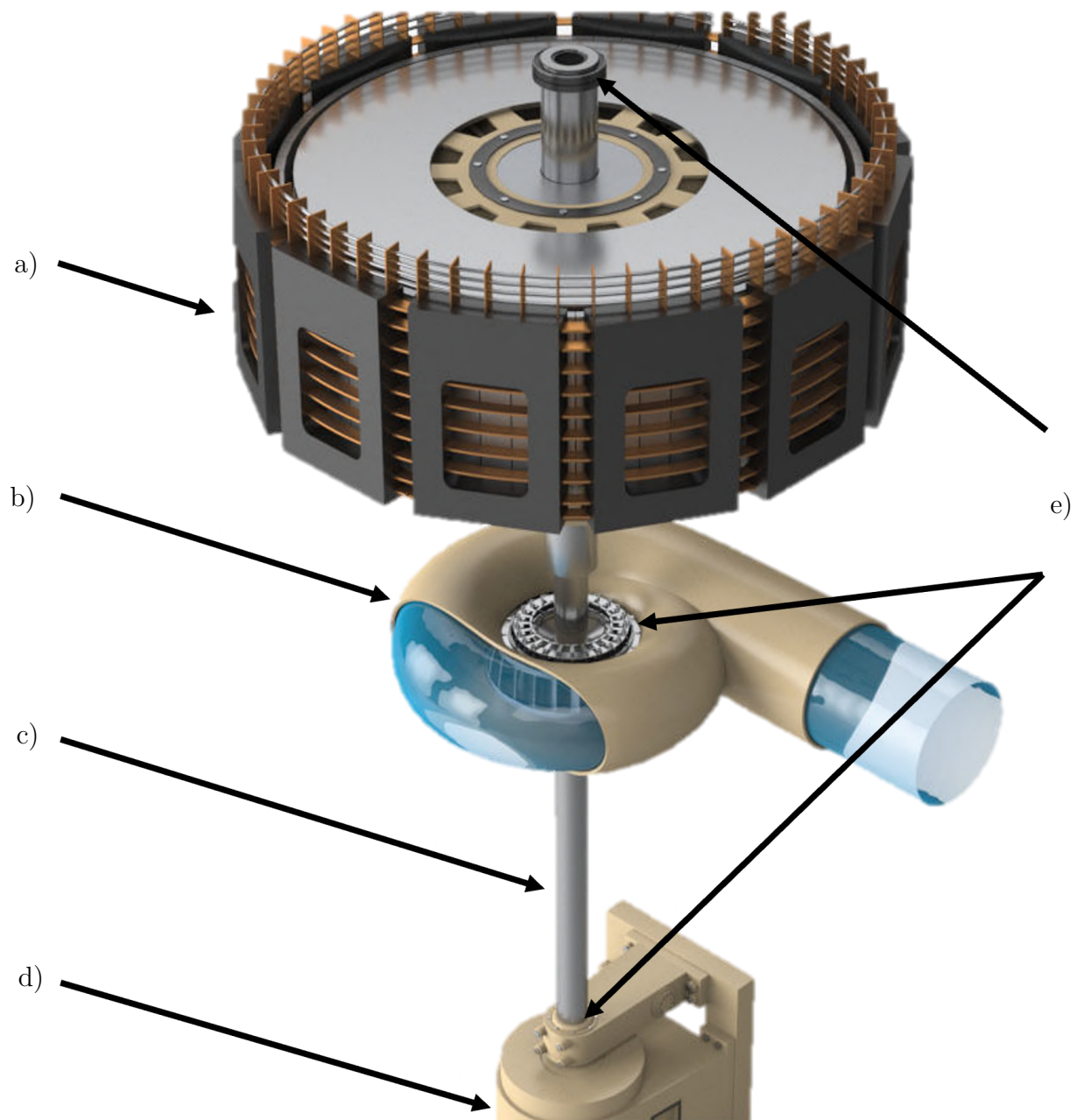


Figure 1.1 Schematic of a typical Francis turbine (MotionDana, 2021). (a) Alternator. (b) Spiral case. (c) Shaft. (d) Support frame. (e) Guide bearings.

can be challenging due to environmental concerns, public opposition, and very high upfront capital costs. As an alternative, wind and solar electricity generation are rapidly gaining traction as setting up a wind or solar farm is significantly less constraining and does not require the permanent flooding of large area (Csanyi, 2011). Yet, these power generation methods come with their own pros and cons. On one hand they are quick and cheap to install but on another, they still lack efficiency and consistency (Csanyi, 2011, Chamorro, 2020). This inconsistency is to be expected from weather and diurnal course dependant technologies, which in turn causes fluctuations in the power grid. Storage technologies such as batteries can help address the intermittency issue by storing excess energy generated during peak production and providing it during periods of low production. However, current storage technologies are still expensive and limited in their capacity, making it difficult to rely on them entirely. The battery technological industry is still relatively unchanged compared to 30 years ago with the advent of the first commercial lithium-ion battery by Sony (Reddy et al., 2020). Thus, to regulate the power grid, hydro-units are now used intermittently with frequent start/stop routines or running at lower-than-optimal capacity. As these units were designed to run at constant loads, these new methods of operation are causing accelerated wear and tear on hydroelectric turbines which require producers to revise and improve their failures prediction and maintenance programs.

1.1.2 Emergence of Industry 5.0 in engineering

Industry 5.0 is increasingly visible in many spheres of society. It is defined as the increased collaboration between humans and machines, physical as well as virtual (Adel, 2022). This occurrence is a result of the exponential amount of data being generated through new technologies in hardware and software capabilities. The emergence of big data and artificial intelligence (AI) has significantly influenced various sectors, including engineering-related industries. The proliferation of Internet of Things (IoT) devices, social media platforms, digital transactions, and other technological innovations have contributed to the exponential growth of data. This data, when harnessed correctly, can provide valuable insights and drive decision-making processes across various sectors, including engineering systems (Miotto et al., 2018).

Engineering systems, like hydroelectric turbines-alternators units, have become increasingly complex just like the need for efficient and effective monitoring and control. Sensors embedded in these systems generate a wealth of data that can be used for various purposes, such as monitoring system performance, detecting failures, and predicting future behavior. However, the sheer volume and complexity of this data present a significant challenge. Tradi-

tional data processing and analysis methods are often inadequate to handle such large-scale, high-dimensional data (Hosny et al., 2018). Furthermore, sensors on operating hydroelectric units rarely measure quantities of interest like flow rate and strain directly. They are inferred from other measurements.

AI models, particularly machine learning algorithms, are capable of processing and analyzing large volumes of data and extract meaningful insights. These models can learn from the data, identify patterns, and make predictions, making them appealing for engineering applications. For instance, in a mechanical system, sensors can monitor variables such as temperature, pressure, vibration, and so on. Any anomalies from baseline operations in these variables could indicate a potential failure. However, detecting such failures is not a straightforward task. It requires the analysis of large volumes of sensor data and the identification of subtle patterns (Harrer et al., 2019). The emergence of big data and AI has brought about a paradigm shift in the way we design, operate, and maintain engineering systems. As we continue to generate more and more data, the role of big data and AI in engineering systems is only set to increase (Dzau and Balatbat, 2019). According to Zhao et al., 2021, while there have been significant advancements in AI-driven Prognostics and Health Management (PHM), a comprehensive approach that seamlessly integrates monitoring, diagnosis, and prognosis is yet to be achieved, and the importance of open-source resources is still not fully recognized.

1.1.3 Goal and scope of work

To be able to create a DT of a hydroelectric turbine with all the diverse physics and phenomena that come with it, we need to start small, and then scale up until we reach a model applicable in real-world working conditions. This project is only the first step towards that purpose. By the design, building and instrumentation of a VARM, we aim to provide an empirical data source to researchers working on early versions of hydroelectric turbines DT models and DT models of mechanical systems in general. The test rig will provide training and validation data for the development of such models by monitoring multiple test cases of lab-simulated failure modes similar to what can be found on actual hydroelectric turbines. Furthermore, it will be accompanied by a finite-element model (FEM) offering a clean data source governed by known parameters for early training of the DT and comparison with the empirical data recorded on the VARM.

As it is the case with most mechanical systems, having quantifiable knowledge on all parts and components of a machine can be a challenging undertaking deserving of its own research project. For that reason, parameter identification is only slightly touched regarding

the VARM and its components as system identification techniques are numerous and very sophisticated. The objective is to provide an experimental setup of vertically oriented rotating systems mainly for the design of a DT but also for any research related to rotor dynamics which includes parameters and failures identification using system identification techniques.

1.1.4 Outline

This thesis presents an experimental test bench exhibiting different dynamics to provide data for a hydroelectric turbine DT. Following this introduction, a review of literature about existing test rigs of rotating machinery, common failures and monitoring methods and modeling techniques in rotor dynamics is presented, followed by the synthesised goal of this master's project. This is then followed by the methodology employed for the design, building and instrumentation of the VARM as well as experimental results from different configuration of the VARM. Then, we introduce the methodology used to build the FEM and discuss and analyse the results that are produced. Finally, the conclusion summarizes the project in its entirety, discusses its limitations and possible improvements as future work to be done.

CHAPTER 2 LITERATURE REVIEW

2.1 Existing experimental test-benches

In the study of rotor dynamics, experimental test benches are of great importance. They provide a platform for the validation of theoretical models against real-world data, helping to identify any discrepancies and improve the accuracy of these models. Beyond validation, test benches enable a deeper understanding of real-world behavior. Real-world systems often deviate from theoretical predictions due to factors such as manufacturing tolerances, material imperfections, and environmental conditions. By observing these behaviors in a controlled setting, it is possible to gain insights that are not likely through theory alone. Furthermore, test benches offer a controlled environment for testing and evaluating new technologies, designs, and control strategies, accelerating the development of innovative solutions in rotor dynamics. They also play a crucial role in fault diagnosis and prognosis. By simulating various fault conditions, it becomes feasible to develop and test new methods for detecting and predicting faults, leading to more reliable and resilient rotor systems (Schafroth et al., 2009). Lastly, test benches serve as educational tools, providing students and engineers with a practical understanding of rotor dynamics principles and hands-on experience. In the context of vertical rotor dynamics, a test apparatus can be particularly useful for studying failures, understanding the effects of gravity on rotor performance, and testing vertical rotor designs and technologies. Despite these advantages, there is a notable scarcity of vertical rotor test benches in academic research, indicating a need for more work in this area.

Most test benches are designed to be operated in a horizontal position where the effect of gravity plays a stabilisation role in the dynamics of the machine as can be seen on Fig.(2.1)(a). The dead weight of the rotating shaft acts as a stabilisation force which constrains the shaft to the lower part of the bearing even while rotating. This limits the effects of impact contacts between the shaft and the bearing inner-race as well as within the bearings different components on the behavior of the machine. In contrast, these impacts are much more present in a vertical configuration setup which in turn makes non-linearities much more present in the response monitoring of the machine. Furthermore, the vertical orientations of rotors also comes with the addition of a loading force and moment in the plane of rotation when lateral displacements amplitudes become important as shown on Fig.(2.1)(b). These non-linearities can prove to be very challenging to model when no experimental data is available for comparison. Finally, the type of non-linearity in the system will greatly depend on the type of bearings used as they are the principal source of non-linearities in rotating machinery.

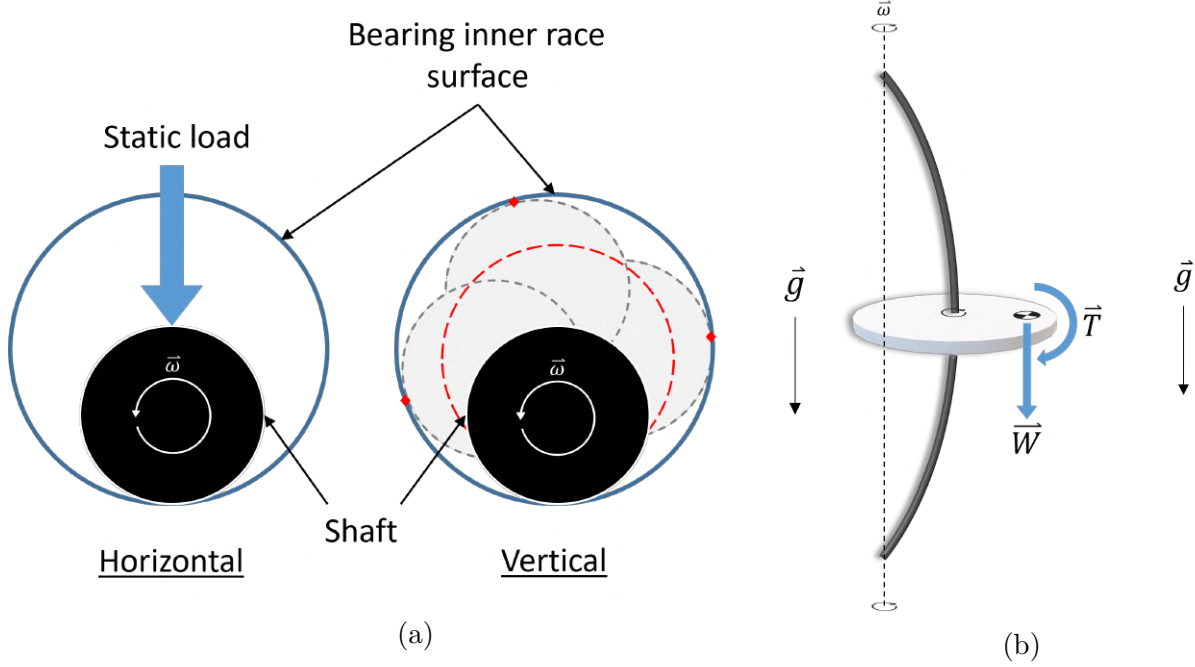


Figure 2.1 Effects of vertically oriented rotors where g is the gravity, W the disc's weight, ω the rotation velocity and T the induced moment: (a) on bearings behavior and (b) on rotor loads.

In the study of rotor dynamics, the design and orientation of the test-bench are critical factors that significantly influence the research objectives and methodologies. The literature reveals a clear distinction between studies employing horizontal test-benches and those utilizing vertical ones, each serving specific research needs.

Research employing horizontal test-benches often focuses on vibration reduction, misalignment, and sensor development. For instance, Kasarda et al. (2004) utilized a horizontal test-bench with Active Magnetic Dampers (AMD) attached at the mid-span and outboard quarter-span of the rotor to control vibrations. This design allowed for a targeted approach to vibration control, demonstrating the effectiveness of this orientation in vibration studies. Similarly, (Li et al., 2019) designed a test-bench with an "intermediate" part for easy alteration of the bearing structure, emphasizing the need for flexibility in test-bench design. In the study of misalignment, Hariharan and Srinivasan (2009) used a self-designed pin type flexible coupling in their rotor-coupling-bearing system. This unique design element allowed for a more accurate simulation of misalignment effects in rotor systems. Holl and Pfaffenbauer (2020) developed a horizontal rotor test bench designed to maintain high rotation speeds and temperatures for sensor device testing, highlighting the importance of environmental control in test-bench design. A notable example of horizontal test-bench can be found in

Ohio Northern University. The design configuration employed by Krak et al. (2011) and presented in Fig. 2.2 allows for great experimental versatility and is comprised of two subsystems: the vibration simulator, which is a mechanical system designed to demonstrate various vibration phenomena associated with common machine faults. These include rotating imbalance, shaft misalignment (both parallel and angular), impact and slider-crank vibration, and flow-induced vibration. The simulator uses two shafts coupled to a motor with a flexible spider coupling, which dampens vibrations from the motor and allows for quick changeover between configurations. The imbalance configuration uses two inertial disks which can be adjusted in position, and set screws on the disks serve as imbalance masses making the setup able to simulate different types of imbalance based on the location of the disks and the set screws.

On the other hand, vertical test-benches are predominantly employed in research on thrust bearings. Walvekar et al. (2018) designed a test-bench influenced by the five-ball fatigue tester developed by Zaretsky et al. (1982) with the test specimen being a thrust bearing with the lower race replaced with a flat specimen. This design allowed for a targeted application of load, demonstrating the importance of load application methods in test-bench design. Henry et al. (2014) designed a test-bench equipped with a 15 kW electrical motor, a loading system, and a support for the thrust bearing. This comprehensive design facilitated a wide range of data collection, emphasizing the importance of diverse data collection methods in the preliminary design considerations. Myers (2017) further extends this line of research by designing a vertical test-bench to study hydrodynamic thrust bearings under varying loads, showcasing the adaptability of vertical test-benches.

Beyond these specific areas, both horizontal and vertical test-benches have been employed in a variety of research objectives. For instance, Kawamura et al. (2012) used a horizontal model rotor system to diagnose abnormalities in rotating machinery. Their test-bench design includes a shaft with one or two disks supported by rolling bearings, highlighting the importance of modularity in test-bench design. Conversely, Nässelqvist et al. (2014) used a vertical test-bench with a mid-span rotor supported by two tilting-pad bearings. This design allowed for the accurate simulation of the unbalance response of the vertical test-bench based on a novel analytical model of tilting-pads bearings. Fig. (2.3) shows a schematic and photograph of the test-bench that was developed for their research. Particularly, their studies reveal that in vertical machines, cross-coupling stiffness can be as significant as the radial stiffness can impact dynamics and increase higher frequency components. It is therefore crucial to consider these factors in high-load simulations or detailed dynamic analyses.

Finally, the choice between horizontal and vertical test-benches is often influenced by the spe-

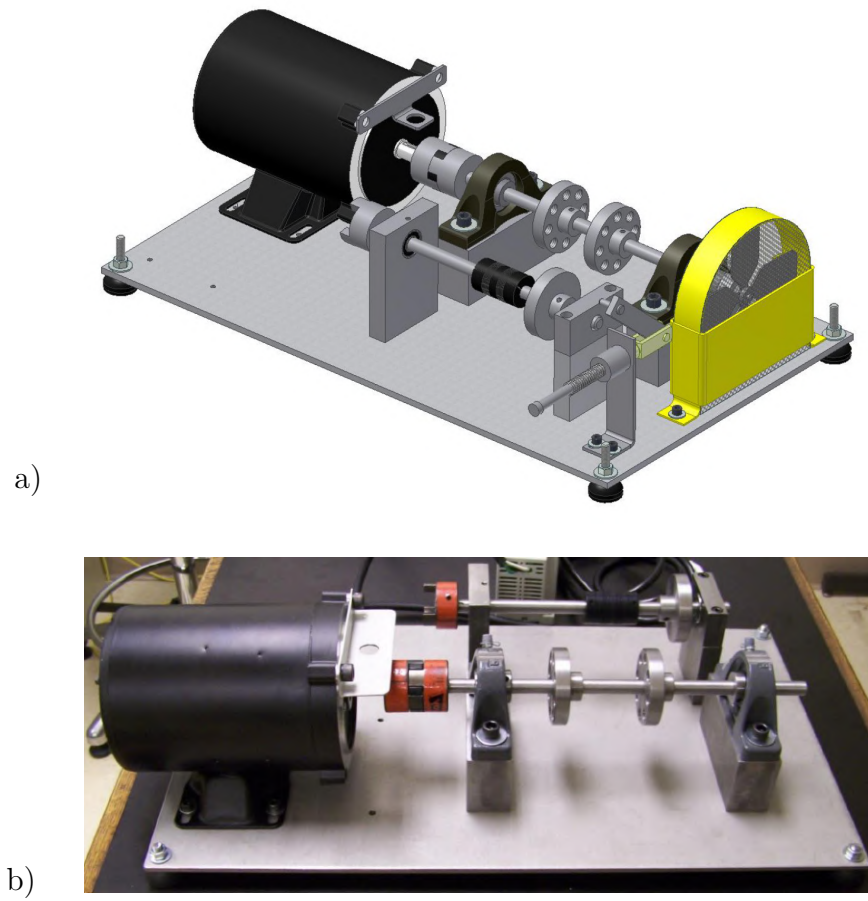


Figure 2.2 Modular horizontal test-bench by (Krak et al., 2011). (a) Schematic showing the flow-induced configuration. (b) A photograph of the unbalanced configuration.

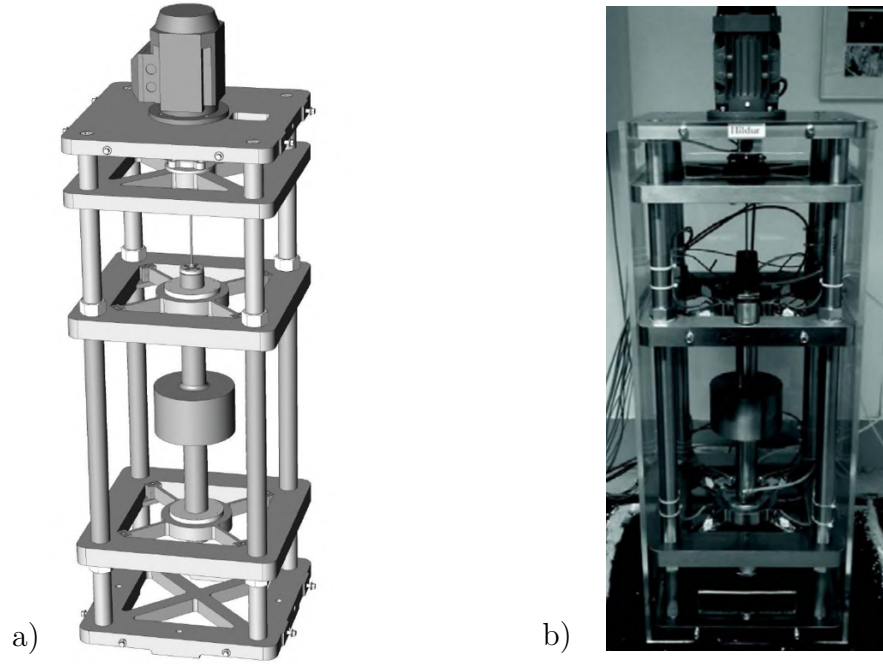


Figure 2.3 Vertical axis rotating machine test-bench (Nässelqvist et al., 2014). (a) Schematic (b) Photograph.

cific research objectives, with horizontal test-benches often employed for vibration reduction of unbalances and misalignment, and sensor development studies while vertical test-benches are being predominantly used for thrust bearing experimentation. Only Nässelqvist et al. (2014) produced a vertical-axis rotating machine with the goal of facilitating the development of analytical and numerical models to evaluate general rotor dynamic failures in a vertical orientation. This review served as a starting point in the design phase of this project by providing ideas for the development of a VARM that would facilitate the experimental simulation of different rotor dynamics phenomenon.

2.2 Monitoring failures in rotating machinery

2.2.1 Common failures

Rotating machinery, such as turbines, motors, and generators, are subject to various types of failures. These can be broadly classified into mechanical and electrical failures. Mechanical failures often involve components such as bearings, shafts, and seals, while electrical failures typically involve the stator, rotor, and insulation (Bloch and Geitner, 1997).

This discussion will span the most common mechanical failure modes in hydroelectric tur-

bines: static unbalance, dynamic unbalance, misalignment, mechanical looseness, cavitation, and bearing failure, while providing a comprehensive understanding of each type of failure.

Unbalance is a fundamental issue in rotating equipment stemming from the natural and to-be expected imperfection and tolerances of manufacturing processes. It is characterized by an uneven distribution of mass around the axis of rotation and is typically categorized into two main types: static unbalance and dynamic unbalance which are shown on Fig.(2.4) (Rao, 1996).

Static unbalance arises when there is an eccentricity between the center of gravity and the geometric center of the rotor. This mass imbalance results in a rotor inclination in the resting position and manifests as a simple pendulum-like swinging motion when the machinery is in operation. The effects of static unbalance, while seemingly minor, can have significant implications on the performance and lifespan of machinery. If left uncorrected, static unbalance can give rise to excessive vibration, leading to increased wear on bearings, premature fatigue in structural components, and ultimately a reduction in the machine's operational lifespan. This failure mode is usually relatively easy to detect and correct (Rao, 1996).

Dynamic unbalance, however, presents a more complex scenario. In dynamic unbalance, the mass eccentricity varies along the rotor's axis, resulting in a wobbling or precessing motion during rotation. Unlike static unbalance, dynamic unbalance is more difficult to detect and rectify due to its multi-plane nature. It requires the addition of correction mass in at least two separate planes perpendicular to the axis of rotation. The effects of dynamic unbalance parallel those of static unbalance, with the potential for more severe outcomes due to the complexity of the mass distribution. The vibrations caused by dynamic unbalance can also lead to high stress levels in various parts of the machinery, potentially causing structural damage if not addressed correctly (Rao, 1996).

Misalignment, another prevalent failure in rotating machinery, refers to a situation where the center lines of two interconnected rotating shafts fail to align. Misalignment can be broadly classified into two types: parallel, also known as offset, and angular. Parallel misalignment is a condition where the center lines of the shafts are parallel but offset from each other. In contrast, angular misalignment occurs when the shafts intersect at an angle. Saavedra and Ramirez (2004) studied the effect of misalignment between a horizontal shaft and the motor drive line while being coupled by a flexible jaw-spider coupling. The analytical model they developed and experimentally tested demonstrates the change in stiffness in the coupling depending on the angular position of the shaft which produces visible harmonics in the frequency domain. Regardless of the type, misalignment can induce increased vibration, accelerate bearing and seal failure, elevated power consumption, and potentially, catastrophic

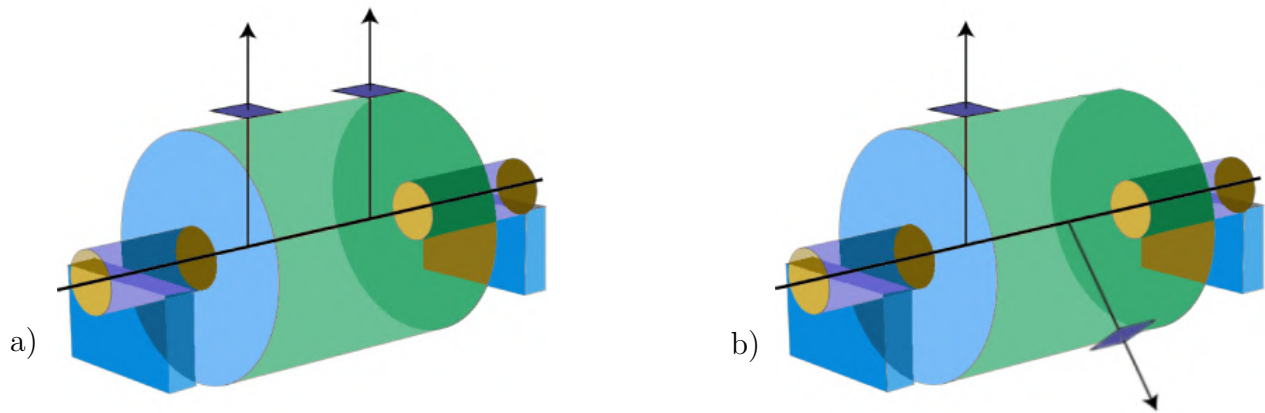


Figure 2.4 Different types of mass imbalances in rotors: (a) Static unbalance and (b) dynamic unbalance (Rokade, 2021).

machine failure (Rao, 1996).

Manufacturing imperfections in rotating components is a common occurrence in many types of shaft applications, such as crank shaft and roller shaft. This issue can significantly affect the overall dynamic performance of the shaft-bearing system. In a study conducted by Guo et al. (2016), a systematic modeling approach was developed to analyze the effect of shaft manufacturing bending deviation on the dynamic behavior of a geared rotor system. The study found that as the magnitude of the shaft bending deviation increases, the overall systematic vibration response also shows an ascending trend. The magnitudes of most of the frequency components in the vibration spectrum increase as the deviation increases. Furthermore, as the position of the deviation changes, the system shows different dynamic sensitivities to the interested frequency components. Finally, as the rotating speed increases, the overall system response also obviously increases and the frequency component of the unbalance force gradually dominates the spectra responses.

Cavitation, while particularly relevant to fluid-moving machinery such as pumps, propellers and turbines, also represents a significant failure mode. Cavitation arises when rapid changes in fluid pressure lead to the formation and collapse of vapor-filled cavities within the fluid. The constant formation and violent collapse of these cavities can cause severe damage to the machine parts, evident as pitting or erosion on the surfaces in contact with the fluid. This process usually occurs when a volume of liquid is subjected to a sufficiently low pressure, causing it to rupture and form a cavity. This can occur behind the blade of a rapidly rotating propeller or any surface vibrating in the fluid with sufficient amplitude and acceleration.

When these vapor-filled cavities or "bubbles" are subjected to higher pressure, they collapse, generating shock waves that may damage machinery. These shock waves are strong when they are very close to the imploded bubble, but rapidly weaken as they propagate away from the implosion. This repeated implosion causes surface fatigue of the metal, leading to a type of wear known as cavitation wear. This violent mechanism releases a significant amount of energy in the form of an acoustic shock wave and as visible light. At the point of total collapse, the temperature of the vapor within the bubble may be several thousand kelvin, and the pressure several hundred atmospheres (Liu et al., 2016).

Another failure commonly encountered in rotating machinery is bearing failure. Improper lubrication, corrosion and contamination, misalignment, and fatigue or spalling are among the most common causes of bearing failure. Improper lubrication or exposing the bearing to excessive temperatures can lead to bearing failures. Contaminants such as dirt, sand, water, and chemical compounds can degrade the lubricant or corrode/erode bearing surfaces, thus hastening wear and failure. Misalignments, particularly bent shafts, and improperly positioned shafts and locking nuts, can lead to uneven load distribution and excessive vibration generation, further hastening failure (Mohanta et al., 2017).

This comprehensive, however not exhaustive, summary of common mechanical failures in rotating machinery will help determine the kind of failures we want to replicate on the experimental setup. Unbalances can be the result of many different causes in hydroelectric turbines and is fairly easy to reproduce with an imbalance mass. Misalignment is also very common and could be achieved by simply enforcing an offset between bearing components. Finally, bearing failure could also be replicated depending on the type of bearing used.

2.2.2 Common monitoring procedures

Condition monitoring is a crucial aspect of maintaining the operational efficiency and longevity of rotating machinery. It involves the systematic collection and analysis of data to identify changes in the physical condition of machinery that could indicate impending failure (Jardine et al., 2006).

Condition monitoring techniques are employed to detect these failures at an early stage, thereby preventing catastrophic breakdowns and costly unplanned downtime. These techniques include vibration analysis, oil analysis, thermography, and motor current signature analysis, among others (Heng et al., 2009). Vibration analysis is one of the most widely used techniques for monitoring the condition of rotating machinery. It involves measuring the vibration signals produced by a machine during operation to detect any abnormalities. Changes in vibration patterns can indicate problems such as misalignment, imbalance, or

bearing faults (Jardine et al., 2006; Efthymiou et al., 2012). The norm *ISO Surveillance et diagnostic d'état des machines Groupes de production hydroélectrique* (2020) highlights the type of physical quantities that should be used to successfully detect specific types of failures. For mechanical failures, vibration monitoring is mostly comprised of displacement sensor, accelerometers and tachometers, all physical quantities related to displacements. Oil temperature and electrical quantities such as current and power are also monitored when applicable.

Oil analysis is employed for machinery with lubrication systems. It involves analyzing the physical and chemical properties of the oil, as well as any contaminants present, to detect issues such as wear, corrosion, and overheating (Heng et al., 2009). A common application of this can be found in motor vehicle engines where contaminants found in used oil can help diagnose a multitude of failures and correctly plan for oil replacement intervals as it has been investigated by Rostek and Babiak (2019).

Thermography uses infrared imaging to detect abnormal heat patterns in machinery. This can be particularly useful for identifying electrical faults, such as those in the stator windings of electric motors (Pusey, 2007). This technique was further investigated by Moussa (2014) for bearing faults detection and improved by Mehta et al. (2021) with the implementation of machine-learning models on infrared thermography.

Motor current signature analysis (MCSA) is a technique that analyzes the current waveform of an electric motor to detect faults. MCSA can be particularly effective for detecting rotor faults, such as broken rotor bars, air-gap eccentricities, stator faults and much more (Miljkovi, 2015).

While these techniques have proven effective, it's important to note that condition monitoring is not a one-size-fits-all solution. The choice of monitoring techniques depends on various factors, including the type of machinery, its operating conditions, and the specific failure modes that are of concern (Heng et al., 2009). Since this project aims to provide data for the development of a DT based on analytical formulations of rotor dynamics, the type of sensors used should provide data on physical quantities exploited by said formulations which are usually comprised of positions, displacements, speeds and accelerations (Rao, 1996).

2.3 Rotor dynamics modeling - State of the art

Rotor dynamics is a field of study in mechanical engineering that focuses on the dynamic behavior and analysis of rotating machinery. With the increasing complexity of modern rotating systems, the accurate modeling of rotor dynamics has become essential for ensuring their safe and reliable operation. Models involve the mathematical representation and analysis of the vibrations, stability, and critical speeds of rotating systems. In the field of rotor dynamics, the primary objective is to understand the dynamic response of rotating machinery under different operating conditions, such as varying speeds, unbalance, and external forces. By accurately modeling the system's behavior, engineers can assess and predict its performance, identify potential issues, and optimize the design to mitigate undesirable effects while promoting desired behaviors. The modeling process often incorporates various mathematical techniques and physical hypothesis to capture the complex interactions between the rotor, bearings, and surrounding structures. Accurate modeling of rotor dynamics could identify critical speeds, resonant frequencies, and potential instabilities that can lead to excessive vibrations as well as fatigue wear, and even catastrophic failures. Additionally, it enables the evaluation and optimization of different design parameters to enhance system performance, efficiency, and reliability and is essential in the design of a test-bench as the physical concepts relayed by these models will help predict the test-bench behavior under different designs.

2.3.1 The Jeffcott/de Laval model

The Jeffcott Rotor Model, often referred to as the Laval rotor model, has a rich history and relevance in the field of rotor dynamics. Introduced by the Swedish engineer Gustaf de Laval in the late 19th century, this model represents the simplest form of a rotating machine, providing a foundational understanding of the critical aspects of rotating machinery dynamics (Muszynska, 2005).

Gustaf de Laval (1845-1913), a prolific Swedish engineer and inventor, is often recognized for his contributions to the development of steam turbines. However, less known is his role in the field of rotor dynamics modeling, primarily through the introduction of the Jeffcott Rotor Model. The model was named after Dr. H. H. Jeffcott, who popularized it in the 20th century through his influential work on rotor dynamics (Muszynska, 2005).

The Jeffcott Rotor Model is a lumped-parameter model that simplifies a rotating piece of equipment to a mass-less shaft with a single mass at its center. Despite its simplicity, this model has proved effective in providing insights into the dynamics of rotating machinery. The single rotor mass rotates about a fixed axis and is supported by a simple spring and

damper system as shown in Fig.(2.5), allowing the investigation of critical speed, unbalance response, and other key phenomena in rotor dynamics (Rao (1996), Muszynska (2005)).

A simple force-balance method can be used on the stiff rotor to produce the systems equations of motions :

$$M\ddot{x} + C\dot{x} + Kx = F_x \quad (2.1)$$

$$M\ddot{y} + C\dot{y} + Ky = F_y \quad (2.2)$$

where x and y are perpendicular direction to the rotation axis and F_x and F_y refer to excitation forces in both direction (Rao (1996), Muszynska (2005)). As this system of equation is that of a simple damped oscillator, its solution is well known and many well established techniques such as logarithmic decrement can be used for its analysis and parameter identification.

The logarithmic decrement technique is a method used to estimate the damping ratio and damping coefficient of a linear damped oscillator, typically from experimental data of the system's free response. The logarithmic decrement is defined as the natural logarithm of the ratio of the amplitude of any peak to the amplitude of a following peak (Liao and Wells, 2011).

Let us denote:

- A_n as the amplitude of the nth peak,
- A_{n+1} as the amplitude of the next consecutive peak ($n + 1$),
- Δ as the logarithmic decrement.

The logarithmic decrement, Δ , is calculated as:

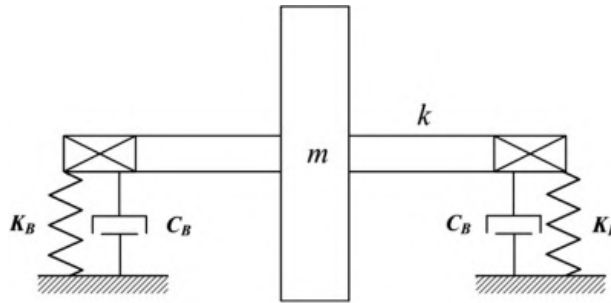


Figure 2.5 Side view of a Jeffcott rotor model schematic. The bearings are modeled as spring-damper pairs in directions orthogonal to the rotational axis (Wilkes et al., 2019).

$$\Delta = \ln \left(\frac{A_n}{A_{n+1}} \right) \quad (2.3)$$

For an under-damped system, the damping ratio, ζ , and the logarithmic decrement are related by:

$$\Delta = \frac{1}{n} \frac{2\pi\zeta}{\sqrt{1-\zeta^2}} \quad (2.4)$$

where n is the number of cycles between the two peaks used to calculate the logarithmic decrement. Often, n is taken as 1, meaning that consecutive peaks are used.

From the above equation, the damping ratio ζ can be estimated as:

$$\zeta = \frac{\Delta}{\sqrt{4\pi^2 + \Delta^2}} \quad (2.5)$$

Once the damping ratio is known, the damping coefficient, C , can be calculated using the relationship:

$$C = 2\zeta\sqrt{KM} \quad (2.6)$$

where K is the stiffness of the system, and M is the mass.

The damped natural frequency can also be obtain from experimental data by measuring the period of oscillation in the response making this method particularly useful for estimating the damping and natural frequency of a system based on measured vibration data.

The Jeffcott rotor model is usually used as an initial step in the analysis of more complex rotating machinery. It lays the groundwork for understanding the fundamental dynamics and instabilities that can occur during operation. The model's principal utility comes from its ability to facilitate the prediction of critical speeds of simpler rotating systems, one of the most fundamental aspects of rotor dynamics. Notably, the Jeffcott rotor model is the foundation for the linearized mathematical treatment of rotor vibration problems. It has facilitated the development of critical phenomena such as the gyroscopic effect, modal analysis, and stability analysis (Rao, 1996).

While the Jeffcott Rotor Model's simplicity is a strength, it also imposes limitations. The model assumes a simple geometry and does not account for complexities such as nonlinearities in the model, anisotropy of materials or bearings, and rotor-stator interactions in electrical components such as turbine-powered alternators. As a result, the Jeffcott Rotor Model is

often used a starting point for more complex models, including bearing models, asymmetric rotor models, and multistage rotor models. Several scholars have extended the Jeffcott Rotor Model to incorporate these complexities. For example, the influence of gyroscopic moments, due to rotating parts, has been integrated into the model, leading to more accurate predictions of system behavior. Additionally, improvements have been made to include the effects of damping, internal friction, and thermal effects (Rao, 1996).

2.3.2 Multibody Dynamics

Multibody Dynamics (MBD) is a powerful tool for modeling complex mechanical systems, such as rotating machinery. It allows for the inclusion of multiple bodies, each with its own mass, inertia, and geometry, connected by various types of joints or constraints. In the context of rotor dynamics, MBD allows for more accurate modeling of systems with complex geometries and multiple rotating components. It can account for effects such as gyroscopic moments, damping, and non-linear behavior of bearing components. MBD models can also include the effect of external forces, such as aerodynamic or hydrodynamic forces (Peng, 2010).

Advances are still being made in MBD methods. G  radin and Sonnevill   (2023) developed a new method using a dual-field floating frame-of-reference to efficiently analyze flexible body dynamics in rotating frames by separating motions into rigid-body modes and orthogonal velocity fields, with applications demonstrated in high-speed rotating systems.

In MBD, Euler parameters streamline rigid body rotation simulations. However, traditional integration schemes, such as Newmark and HHT, negatively impact Euler parameter descriptions. Sherif et al. (2019) addresses this by analytically demonstrating the issue and introducing a modified HHT method that substantially reduces the negative effects.

MBD is an effective tool in rotor dynamics, providing a comprehensive approach to modeling complex rotating systems. It complements simpler models like the Jeffcott rotor model, allowing for a more detailed and accurate analysis of complex rotating machinery. However, its utility has to be justified by the complexity of the model being studied. For very simple rotor-bearing assemblies with only few parts, a plain Jeffcott approach would be less time-consuming while still producing reliable results.

2.3.3 Beam theories in rotor dynamics

The behavior of rotating shafts is an integral part of rotor dynamics and the study of rotating machinery. Beam theories, primarily from Euler-Bernoulli and Timoshenko, are fundamental

tools in modeling shaft dynamics, providing insights into the behavior of shafts under various loading and operating conditions.

Euler-Bernoulli's beam theory is a simplification of the linear theory of elasticity. It provides a calculation of deformation and stress distribution in a beam under various load conditions (Lurie, 2010). This theory is widely established and gives the following equations to compute the deflection of a supported beam loaded at its mid-span, which will be useful in the design phase of the VARM:

$$\delta = \frac{FL^3}{48EI}, \quad (2.7)$$

for a simply supported beam and:

$$\delta = \frac{FL^3}{192EI} \quad (2.8)$$

for a beam fixed at both ends, with δ being the displacement, F the applied load, L the beam's supported length, E the material Young's modulus and I the cross section quadratic moment (Lurie, 2010).

This theory operates under the assumption that plane sections before deformation remain plane after deformation and perpendicular to the neutral axis of the beam. In the context of rotor dynamics, Euler-Bernoulli beam theory is used to model the bending behavior of slender rotating shafts and estimate natural frequencies also referred to as critical speeds (Rao, 1996). It is often incorporated into Finite Element Analysis (FEA) methodologies to model the rotor as a series of connected beam elements, each of which follows the Euler-Bernoulli beam theory (Adams and Askenazi, 1999). This approach allows for the detailed prediction of stresses, strains, and vibrations in the system. While this theory has provided significant insights into the dynamics of rotating shafts, it does have limitations. It does not account for shear deformation and rotary inertia, which can be significant in certain situations with specific shaft geometries.

To account for the limitations of Euler-Bernoulli theory, Timoshenko beam theory was developed. This theory extends the Euler-Bernoulli beam theory by accounting for shear deformation and rotary inertia. These factors become significant when dealing with short, thick beams, or at high rotational speeds, where the shear deformation and rotary inertia cannot be neglected (Beck and Silva Jr, 2011). In rotor dynamics, Timoshenko beam theory has been used to create more accurate models of rotor behavior. This theory allows for the analysis of more complex systems and a broader range of operating conditions than Euler-Bernoulli

beam theory.

Both Euler-Bernoulli and Timoshenko beam theories have been extensively applied to various problems in rotor dynamics. They have been used to model the behavior of various types of rotating machinery, including turbines, engines, and electric motors. Researchers have extended these models to include more complex behaviors. For example, these beam theories have been adapted to model cracked shafts (Rubio, 2009), enabling the detection and analysis of this type of failure, which is a critical aspect of machinery maintenance and failure prevention.

Euler-Bernoulli and Timoshenko theories are foundational tools in the field of rotor dynamics, enabling the modeling and analysis of the dynamic behavior of rotating shafts. While both theories have their limitations, they provide valuable insights that can be extended to more complex systems and operating conditions. They will likely continue to be fundamental components of the toolkit for researchers and engineers in the field of rotor dynamics. These theories will prove useful in the design of the VARM by providing analytical formulations for natural frequencies under different operating conditions and materials.

2.4 Bearings modeling

Bearings play a pivotal role in the performance, reliability, and efficiency of rotating machinery. As essential components in a wide array of applications, including automotive, aerospace, manufacturing, and energy industries, they enable smooth and controlled motion between moving parts. Therefore, a comprehensive understanding and accurate modeling of bearings are critical for optimizing the design, operation, and maintenance of such machinery. The importance of bearings understanding and modeling in rotating machinery cannot be overstated, as these components significantly influence the overall system behavior. Bearings are responsible for supporting the load and reducing friction between moving parts as well as maintaining the rotating parts in their axis of rotation allowing for optimal energy transfer. Consequently, they have a direct impact on the efficiency, lifespan, and noise levels of the machinery in which they are employed.

One of the primary challenges in bearings modeling is the complex interactions between various forces, materials, and lubricants. Advances in computational methods, such as finite element analysis and computational fluid dynamics (CFD), have facilitated the development of more accurate and sophisticated models. These models enable engineers to predict the behavior of bearings under various operating conditions. This information is crucial for selecting appropriate bearing types and materials, as well as designing effective maintenance

strategies to ensure the longevity and reliability of the machinery.

Furthermore, a thorough understanding of bearings enables engineers to identify and mitigate potential failure modes, such as fatigue, wear, and corrosion. This is particularly relevant in the context of condition monitoring and predictive maintenance of heavy machinery, where early detection of bearing faults can significantly reduce the risk of catastrophic failures and associated downtime costs. Many different methods were developed to model bearing dynamics such as lumped-parameters, static and dynamic formulations, and finite-elements models.

2.4.1 The lumped-parameter method

Lumped-parameter models can be utilized in the analysis of bearing dynamics, considering only the planar movements of the bearing components. The connections between the rolling element, raceway, and the bearing housing's influence are represented as linear or nonlinear springs and dampers as shown in Fig.(2.6). In these models, the bearing component experiences solely translational motion, and friction is disregarded based on the pure-rolling assumption (Harris and Creceius, 1986). There are two main categories of lumped-parameter models, depending on whether the motion of the rolling element is considered or not. In the first category, where the rolling element's motion is significant, each bearing component has individual dynamic equations, derived using Lagrange's equations (Harsha, 2005). These lumped-parameter models have been employed to investigate bearing defects (Ahmadi et al., 2015) and structural vibrations in rolling element bearings (Datta and Farhang, 1997).

In the second category, the bearing's degree of freedom is entirely focused on a raceway. The force acting on the ring is calculated by aggregating the force from each rolling element (Gao et al., 2008). This type of lumped-parameter model is particularly useful for studying the nonlinear dynamics of rotor-bearing systems.

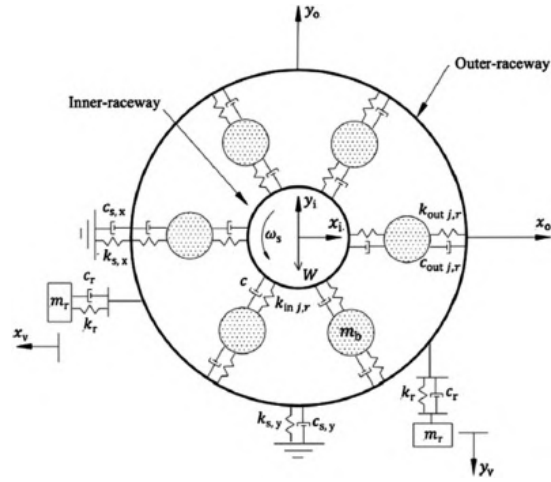


Figure 2.6 Bearing lumped-parameter model where each rolling element is modeled with a spring-damper element (Ahmadi et al., 2015).

2.4.2 The finite-element method

One of the early applications of FEM in bearing analysis focused on the evaluation of stress and deformation in rolling element bearings under various loading conditions. Since then, several researchers have developed advanced and refined FEM models to study more complex bearing geometries and configurations (Xiang et al., 2009; Deng et al., 2013). These studies have contributed significantly to a deeper understanding of the mechanics and failure modes of various bearing types and have guided the development of more robust and efficient bearing designs.

In addition to analyzing the mechanical behavior of bearings, FEM has also been employed by Rehman et al. (2016) to investigate the thermal performance of bearings, which is critical in high-speed applications. These thermal models have been used to predict temperature distribution within bearings and assess the influence of parameters such as rotational speed, load, and lubrication on bearing temperature.

FEM has further been utilized to study the effects of manufacturing and assembly errors on bearing performance. These analyses have helped identify critical tolerances and quality control measures necessary for maintaining optimal bearing performance and longevity. Furthermore, researchers have employed FEM to simulate bearing wear and predict the onset of fatigue, which is crucial for determining bearing service life and maintenance intervals (Deng et al., 2013).

The integration of FEM with other modeling approaches, such as MBD and fluid-structure interaction, has expanded the scope of bearing analysis to encompass more comprehensive and realistic simulations. These coupled models have been particularly useful for investigating complex phenomena, such as elastohydrodynamic lubrication and cage dynamics (Cao et al., 2016).

CHAPTER 3 SYNTHESIS OF THE LITERATURE REVIEW AND RESEARCH OBJECTIVES

3.1 Synthesis of the literature review

In order to develop a DT of a hydroelectric turbine, an experimental test-bench is required to generate data that differentiates itself from common analytical and numerical formulations in rotor dynamics. The vertical orientation of the rotating shaft in the VARM is its main feature as vertical rotors tend to exhibit different dynamics than horizontal ones. Many horizontal test-benches exist in academia but very few are vertically oriented. In order to produce rich data, the VARM needs to replicate multiple types of failures such as unbalance and misalignment. Bearing failure could also be investigated by having a modular setup that allows an efficient change of components. Vibration monitoring of rotating machines is usually done by displacement and acceleration data. Many modeling techniques exist in rotor dynamics with the most appealing being the Jeffcott rotor model for its simplicity, and the finite element approach for its flexibility and accuracy. Bearing modeling is another realm by itself. Fluid bearings employ fluid-interactions models which diverge from mechanical modeling techniques, and solid bearings have an astonishing number of moving parts making their models and analysis laborious.

3.2 Objectives

Three objectives are proposed for this Master's project which will facilitate the work of future researchers in need of experimental data of a vertical-axis rotating machine.

1. Design and build an experimental setup of a vertical-axis rotating machine (VARM) to simulate multiple failure modes in rotating machinery. The VARM needs to be modular and flexible in its capacity to operate different materials, shafts, rotors, and bearings as to simulate numerous test cases.
2. Instrument the VARM with an array of sensors to record data during experiments and document the phenomenon being captured. The quantities measured should be useful for the development of numerical and analytical methods in rotor dynamics.
3. Build a numerical model of the VARM as a secondary data source. This will be achieved using a finite-element model which allows for adaptability in the different configurations the VARM can take.

The following Fig.3.1 summarizes the role and place of this thesis in the Digital-Twin project. Data sources are designed in the form of a FEM and a VARM which is instrumented accordingly to the needs of the algorithms. The operational data and starting parameters are then used to train the digital-twin models and their output is finally compared to a final set of data from a high-speed camera recordings.

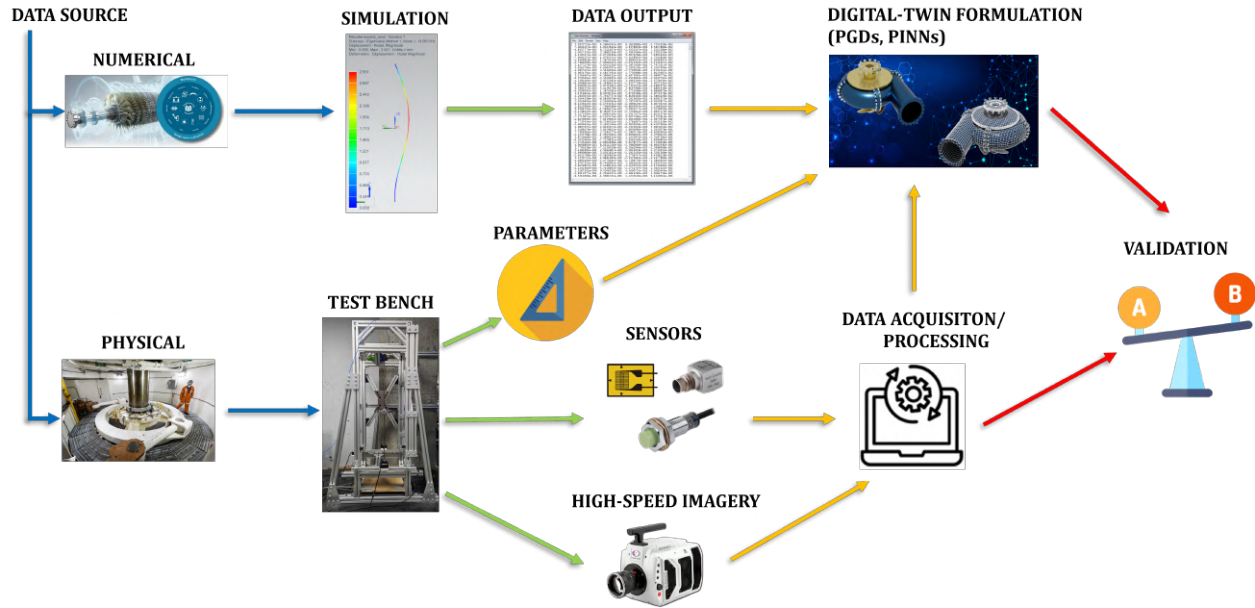


Figure 3.1 Overview of the VARM design project as a part of the Digital-twin project

CHAPTER 4 METHODOLOGY I: THE VARM

The VARM test bench needs to be highly versatile to accommodate different configurations for different experiments. It needs to be modular enough to be able to apply important performance and operational upgrades to simulate different types of behaviour and failures found in common rotating machinery, especially Francis hydraulic turbines. In addition, because the VARM is designed to mechanically display the failures of rotating machinery, it must be safely operable with minimum training. In the case of catastrophic, unplanned failure, it would be ideal for safety concerns that the machine discharges the lowest possible mechanical energy to limit potential damage under the most critical conditions. To achieve these requirements, a highly modular design and a highly deformable shaft material with high ultimate tensile strength are needed. It is desirable to be able to easily produce readable responses from the shaft under minimal load, thus reducing mechanical energy and potential catastrophic damage. An early schematic of such a machine showing its main functionalities is presented in Fig.(4.1).

4.1 Material selection and design

It is possible to approximate a material potential to handle high deflections under low loads while remaining structurally safe by determining the amount of strain energy absorbed by the object up to the elastic limit so that the material is not permanently deformed. This quantity is defined by:

$$R = \frac{R_e}{2E}, \quad (4.1)$$

where R_e is the material yield strength and E is its Young's modulus. A high ratio could indicate a lower deformability under load but a high capacity to handle stress before plastic deformations start to appear. Appendix A presents a list of the potential materials and their corresponding properties. From these, the most suitable one was identified as bronze 544, a phosphorous bronze alloy of whose main characteristics and properties are listed in Table 5.1 below. Even though other materials presented a better elastic energy strain like 9310 steel and bronze 630, bronze 544 was chosen for its lower Young's modulus and availability.

In our case, the Euler-Bernoulli beam theory can be effectively used to predict the shaft deflection and modes instead of the Timoshenko beam theory. The shafts elongated aspect ratio necessary for high deflections under low loads makes it flexure-dominated and enables

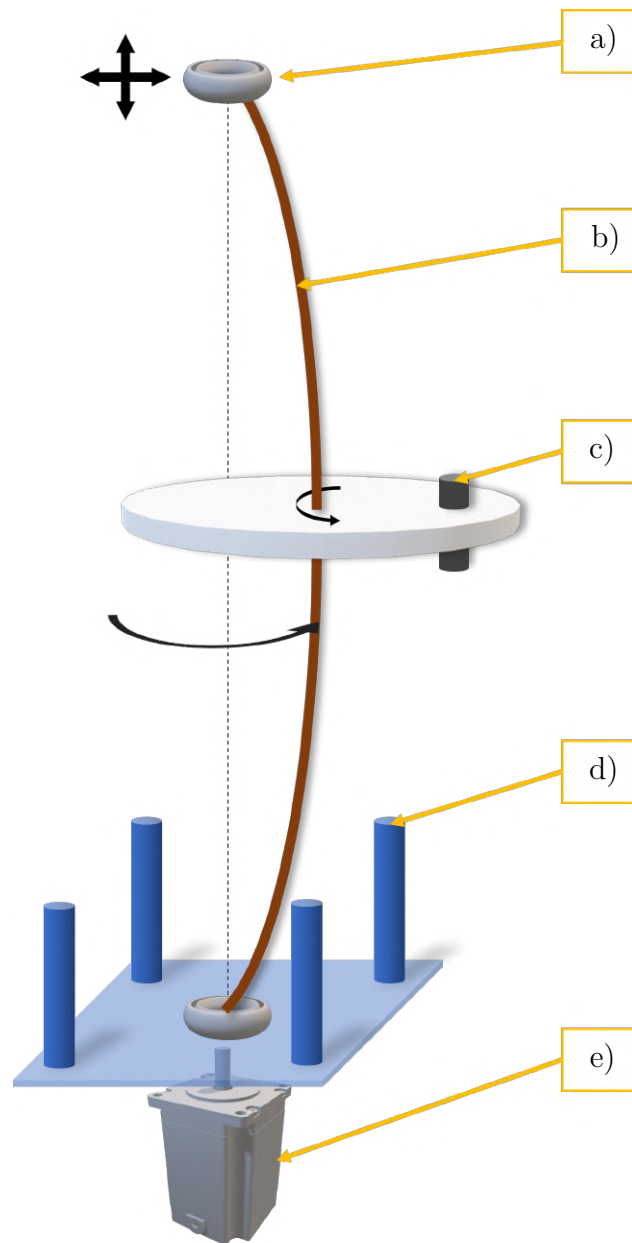


Figure 4.1 Schematic of the desired experimental setup: (a) Upper bearing which is movable in 3 dimensions; (b) flexible shaft; (c) disk with unbalance mass; (d) prismatic support structure; (e) driving motor.

us to discard the effects of shear deformation during deformations (Beck and Silva Jr, 2011). Deflection estimate under unbalanced conditions was first realized using the Jeffcott rotor analytical model (2.1,2.2) with a rotating mass unbalance load where the mass eccentricity remains constant under low deflection hypothesis such as:

$$F_u = me\omega^2, \quad (4.2)$$

where me represents the unbalance in kgm and ω is the rotational speed in rad/s. This was done only to simplify the design process and obtain a general idea of the shafts potential length and diameter based on the maximum deflection value of 30 mm. Once the shaft characteristics were fixed, these deflection values were then precisely established using a finite-element numerical simulation on Simcenter 3D with exact dimensions to ensure that the nonlinear form computed by the software would not exceed the safe maximum deflection value.

Furthermore, the shafts geometric shape, its length and diameter, will affect its deflection under a specified load even more than the materials properties. This can be observed in Eqs.(2.7- 2.8), where the maximum deflection experienced by a supported beam is inversely proportional to its polar moment, which is a function of the diameter raised to the fourth power. In our case, we hypothesize that the shaft will only be loaded by an added imbalance at its middle height, which can be equated to a point-force-loaded beam supported at both extremities by either a fixed or simply supported boundary conditions. In addition, the geometrical characteristics and boundary conditions will greatly affect the shafts natural frequencies, and because we would like to produce experimental data at, or very close to, the first critical speed, we must take it into account during the design phase. This would be very impractical and dangerous to accomplish on an actual hydro-turbine-alternator unit because of the high vibrations exhibited in this speed range, which makes this type of data more valuable. For safety concerns, a maximum rotational speed of 20 Hz or about 1200 RPM must be respected, which means that our shafts first critical speed must not exceed this value. Estimates for this quantity can be derived from Euler-Bernoulli beam theory by applying an eigenvalues analysis to the coupled space-time differential equation. These calculations do not consider the effect of damping on the system as it is usually very low for metallic structures in their elastic domain with a damping-ratio in the order of 10^{-2} . Such low damping would only change the results of this analysis by about 0.05% (Adams and Askenazi, 1999). The complete classical derivation of modal analysis from the Euler-Bernoulli beam equation is widely established and can be found in most book treating continuous medium vibrations. From this analysis, we are left with the following expression to compute a beam

natural frequencies:

$$\omega_n = \beta_n^2 \sqrt{\frac{EI}{\rho A}}. \quad (4.3)$$

Where β_n is a coefficient proper to each mode. In some cases, the values of β_n must be numerically calculated for cases where at least one of the beams extremities is fixed (Mureithi, 2021). Since we aim to operate the VARM around the first critical speed, only the first values of β_n were considered for the fixed-fixed and pinned-pinned cases which will depend on the bearing choice.

Thus, a thin and elongated circular beam is preferred as the rotating shaft of the VARM. And since it is possible to change a beams natural frequencies by modifying its boundary conditions and mass, this could be done on the VARM by interchanging bearings and adding or removing a mid-span disk to the shaft to change the systems overall rotating mass. We could then reduce or increase the first critical speed to our liking in order to acquire explicit experimental data at, or very close to, its corresponding excitation frequency.

For a 1000 mm long and 6.35 mm thick rod of bronze 544, Tab.(4.1) shows the first critical speed for such a shaft for two types of bearing supports.

Finally, the shaft is to be supported by a structure made of 1.5" aluminum extrusions. These are readily available in many shops and manufacturers due to their assembling versatility which makes them an excellent choice for building materials. In addition, add-on parts are diverse and easy to come by allowing for great customisation of the structure.

4.2 Final design

The final design of the VARM is shown in Fig.(4.2). Its prismatic frame is 1.2 m by 0.4 m by 0.4 m, and its overall weight is approximately 30 kg. Fig.(4.2)(a) is a bearing support frame that allows the bearing housing to be moved in the lateral direction to simulate shaft misalignment between the two bearing positions. Most of the brackets used to link the different aluminum extrusion beams can easily be removed or translated in their respective beams to adjust distances or add/remove components in the VARM. For example, adjusting the vertical position of Fig.(4.2)(a) will change the effective length of the shaft and thus modify its equivalent rigidity and critical speeds. The shaft is connected to the motor by a spider jaw-coupling, shown in Fig.(4.2)(e), that provides flexibility and damping to mitigate the vibrations transmitted to the motor. The motor is fixed underneath the medium-density fiberboard (MDF). This wooden part is the only one in the VARM that is non-metallic as

Table 4.1 Properties of a 1 m long, 6.35 mm thick bronze 544 rod and its associated first natural frequencies when supported at its extremities.

Proprieties		1 st Natural Frequency (Pin - Pin) (Hz)	1 st Natural Frequency (Fixed - Fixed) (Hz)
Yield Strength (MPa)	289	8.8	20.0
Young's Modulus (MPa)	110 300		
Ultimate Tensile Strength (MPa)	393		
Ratio R	0.00178		
Effective Length (mm)	1000		
Diameter (mm)	6.35		

the MDF will also damp the vibrations of the structure and help preserve the motor. When required, a steel disk of 1400 g can be added to the shaft and fixed at the desired height by tightening two clamp-like fixtures bolted on it. Individual mass in the form of nuts and screws can be added on the equally spaced bore holes of the disk which are located around its circumference at 40 mm from the rotation axis to simulate mass unbalance. The bronze 544 rod shaft in Fig.(4.2)(b) is supported by two ball-bearings, model P2BL 012-FM (YET 204) made by SKF which are equipped with an eccentric-lock mechanism that is easy to dismount to allow for setup modifications while providing safe and reliable locking between the shaft and the bearings. Detailed bearing specifications are available in Appendix B. Smaller aluminum extrusions in sub assembly (4.2)(c), made from smaller 3/4 thick aluminum extrusions, are fixed to the VARM frame to serve as a support for the displacement sensors monitoring the experiments. Finally, the entire frame is bolted to a heavy steel table, Fig.(4.2)(d), rated for vibration experiments.

4.2.1 Bearing considerations

Throughout the development of the VARM, the process of selecting the most suitable bearings for our specific type of rotating machinery was an essential undertaking. Various factors played a significant role in our decision-making, and numerous trial-and-error attempts were necessary to arrive at a functional type of bearing. One of the key considerations is the type of locking mechanism between the bearing and shaft. Our goal was to employ a bearing that would facilitate easy interchangeability and enable the removal and re-installation of the shaft without the need for specialized tools. This would streamline the process of modifying the machinery as needed.

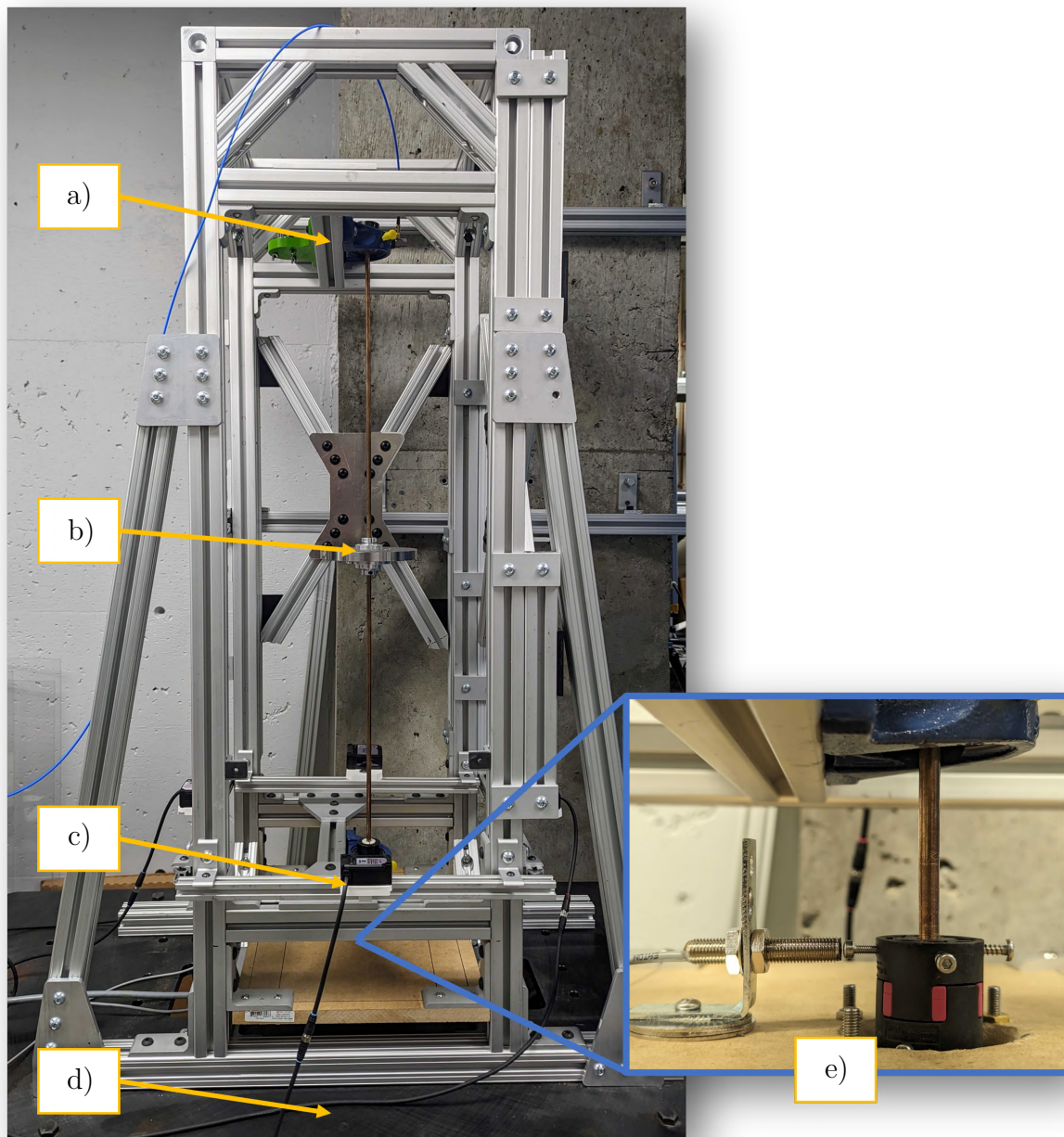


Figure 4.2 Photograph of the vertical-axis rotating machine (VARM). (a) Bearing assembly which can slide 3 spatial directions. (b) Shaft-disc. (c) Lasers support frame which can be moved up and down for measurements at different locations. (d) Steel vibration table. (e) Proximity sensor and motor-shaft jaw-coupling.

We considered factors such as the load capacity, operating conditions, compatibility with the machinery's overall design, and overall stiffness. After several iterations, we were able to identify a bearing that met our criteria and allowed for easy modification and maintenance of the VARM. This proved to be invaluable in ensuring the efficient operation of the apparatus while providing flexibility to adapt to any necessary changes in the machinery's configuration.

Because of our low-load design philosophy, most off-the-shelf bearings are easily capable of handling the expected radial loads from the VARM which are in the order of a 100 N, versus most bearings having radial limitations in the order of a few kilo-newtons. Since the only types of failures replicated in this experiment are from mass-unbalance and bearings misalignment, only the radial load needs to be evaluated to ensure safety of operations as no thrust bearings are used in the setup.

Choosing an appropriate locking mechanism proved to be more difficult and crucial to the system's smooth operations than originally believed. Three types of mechanisms were tested, of which only one was maintained for long-term operation.

The first option consisted of roller-bearings with only slide-in installation requirements which would greatly ease the installation and dismantling requirements. However, this method proved to be highly ineffective after only a few experiments as the rolling-elements lubrication did not last and critical wear on the shafts surface located at the bearing interface was observed as presented on Fig.(4.3). This indicated important friction caused by sliding between the shaft and the rollers which resulted in an important loss of performance in the form of amplified noise amplitude material degradation.

The second option involved the use of set screws. Usually in sets of 2, these screws are to be tightened into the shafts surface to ensure no relative motion between the shaft and the bearings inner race. Owing to the very small diameter of the shaft, the only available bearing models with this type of locking mechanism are made for shafts that are at least two to three times thicker than ours. To ensure adequate fitting, bushings were used to provide support and better compatibility between the shaft and bearings. Yet, this type of tightening system comes with its own downsides as both screws are usually set at 30 to 45 degrees to each other. This positioning of the set screws causes visible offset between the shafts rotating axis and the bearings inner-race center, as demonstrated in Fig.(4.4), and complicates the bushing design.

Lastly, the eccentric-lock mechanism proved to be the best option. The bearing has an inner ring with a protruding eccentric collar that features an off-centre hole. The collar is slid onto the shaft, aligning the off-centre hole with the shaft. A set screw is tightened through a threaded hole in the collar, clamping it onto the shaft and locking the bearing in place.



Figure 4.3 Photograph of surface damage on the shaft caused by open-needle bearing friction

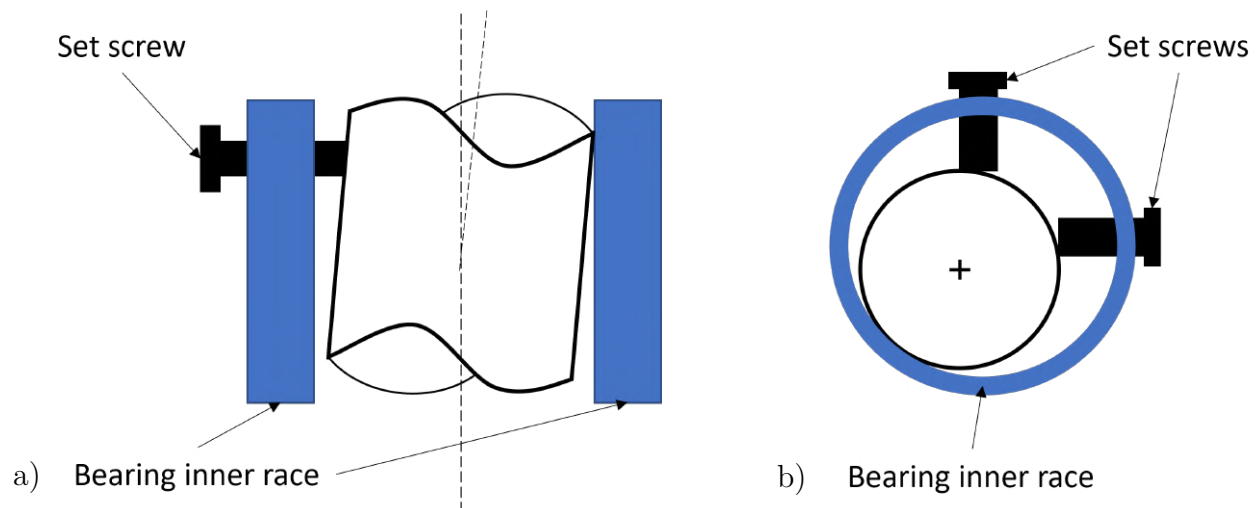


Figure 4.4 Schematic of misalignment problems in rotor-bearing assemblies caused by the use of set screws for mechanical locking of the shaft in the bearing: (a) side view of shaft angular misalignment caused by a set screw; (b) view from above of shaft offset caused by set screws.

The locked collar and inner race of the bearing produce opposing forces on the shaft because of the collars small eccentric bore keeping the shaft firmly fixed to the bearings inner race and spreading the loads onto a bigger surface area as shown on Fig.(4.5). Furthermore, as its locking direction is the same as the rotating direction of the shaft, there is no risk for the collar to loosen up during operations.

Comparing the eccentric locking collar with other locking mechanisms, the former stands out as the best choice for our application. Given the VARM's need for versatility and frequent adjustments, the ease of installation, disassembly, and safety provided by the eccentric locking collar are significant advantages. In contrast, other locking methods may be more challenging to adjust or may require additional components.

Finally, operating conditions and compatibility with the apparatus overall design can simply be synthesised as having a bearing housing that is base-mounted, meaning that the mounting holes for screws are situated on a side-face perpendicular to the bore axis. However, this type of housing is asymmetrical and could cause an anisotropy in stiffness orientations. Bearings with high angular misalignment capabilities were excluded since they do not come with adequate locking mechanism. This limits the use of the VARM in a configuration with increased first critical speed as rigid bearings with no angular misalignment can be equated to fixed boundary conditions. The laboratory where experiments on the VARM take place is free of environmental pollutants like very fine dusts or chemicals. Basic common housings and lubricant seals are sufficient to preserve the bearings rolling elements purity.

4.2.2 Modal analysis and stiffening of the VARMs structural frame

Conducting a modal frequency analysis on the VARM structural frame will inform us on the natural frequencies of the frame. These should be much higher than the studied shaft frequencies and rotational speeds to avoid coupling between the shaft and the frame.

Thus, modal frequency analysis of a complete CAD model of the VARM was performed using Simcenter 3D, a finite element simulation software (Siemens, n.d.). We initially started with only 1D beam elements to get a general idea of the vibration modes. Each element was 1mm in length for a total of 16 762 elements. We then used a CAD model built on CATIA V5 that was then imported in Simcenter 3D to run a complete 3D simulation with a more accurate geometry than what can be achieved with only 1D elements. A total of 1 480 543 CTETRA(4) elements were used in this run with an element size of 8 mm for aluminum beams and curved brackets and 10 mm for plate brackets. In contrast with the first 1D model, this 3D model also accounts for each liaison between beams which are only possible with brackets. These brackets have their own stiffness and weight which should make

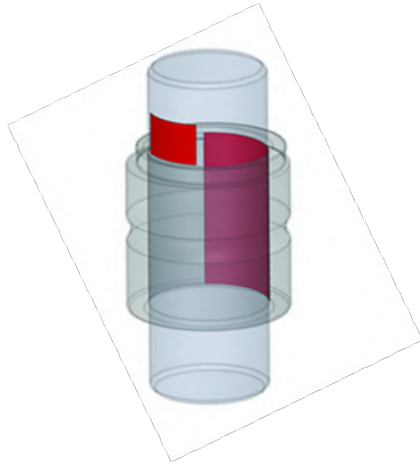


Figure 4.5 Pressured surfaces on the shaft caused by eccentric locking collar (Ansley, 2017). The areas in red mark the pressure distribution on the shaft's surface.

the analysis more accurate by adding them to the FEM. The connection between a bracket and its adjacent beam is modeled using the GLUE function which creates RBE3 elements between face nodes of 2 different bodies for a total of 568 different connections. The following Fig.(4.6) shows the first 3 modal shapes of the initial design of the VARM structural frame and their corresponding frequencies. As it is to be expected with prism-like structures like this one, lateral and torsional bending are easily excited and represent the principal modes of deformation of the frame.

The outcomes of the modal frequency analysis were used to improve the VARM's design by modifying structural elements and boundary conditions. This optimization resulted in a more robust and reliable system capable of accommodating a diverse range of experiments. Fig.(4.7) shows the new mode shapes and associated frequencies with the addition of braces. Details about the modal analysis are available in Appendix C.

However, even with a geometrically accurate model of the frame, it was found that actual natural frequencies were lower than the FEMs estimations. This was investigated through a free-response experiment where the frame of the VARM was impact-excited, and its response recorded by an accelerometer which provided us with peaks at natural frequencies in the FFT analysis of the data which is shown in Fig.(4.8). Similarly shaped and located peaks were obtained multiple times by monitoring the response and exciting the frame at different locations. This disparity between the model and empirical data can be explained by simplified and inaccurate modeling hypotheses meant to lighten the FEM which is comprised of millions of elements. These hypotheses include idealised contact modeling between brackets and extrusions, and the omission of hundreds of nuts and screws in the model, significantly

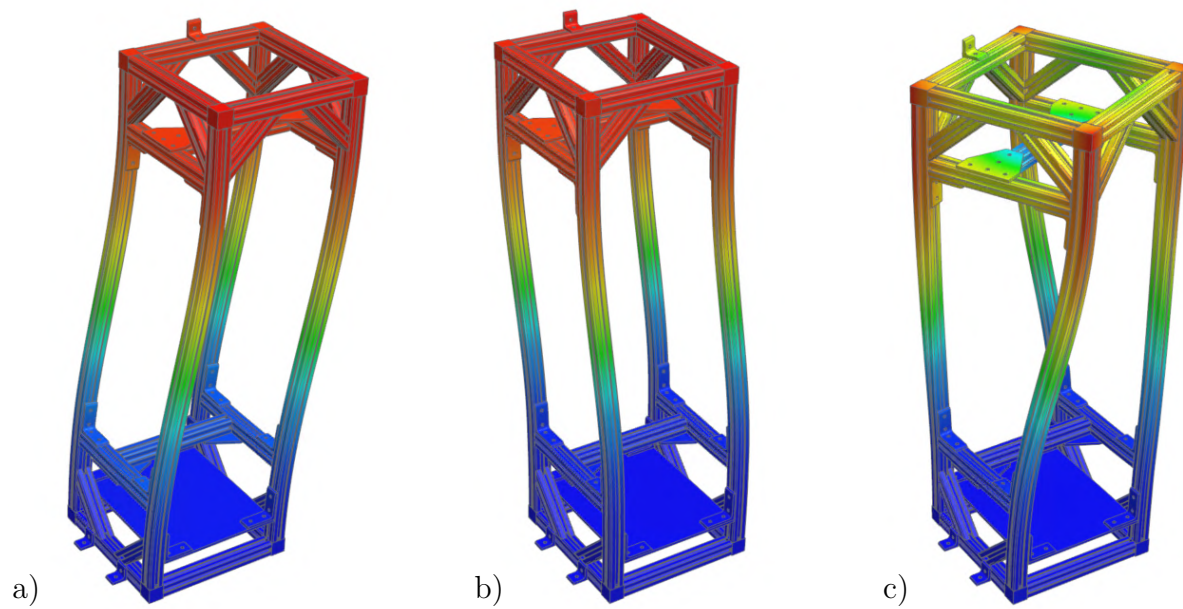


Figure 4.6 Modal shapes of the VARM's initial frame from FEM modal analysis: (a) first mode of vibration at 30 Hz showing lateral bending; (b) second mode of vibration at 42 Hz showing lateral bending; (c) third mode of vibration at 46 Hz showing torsional bending.

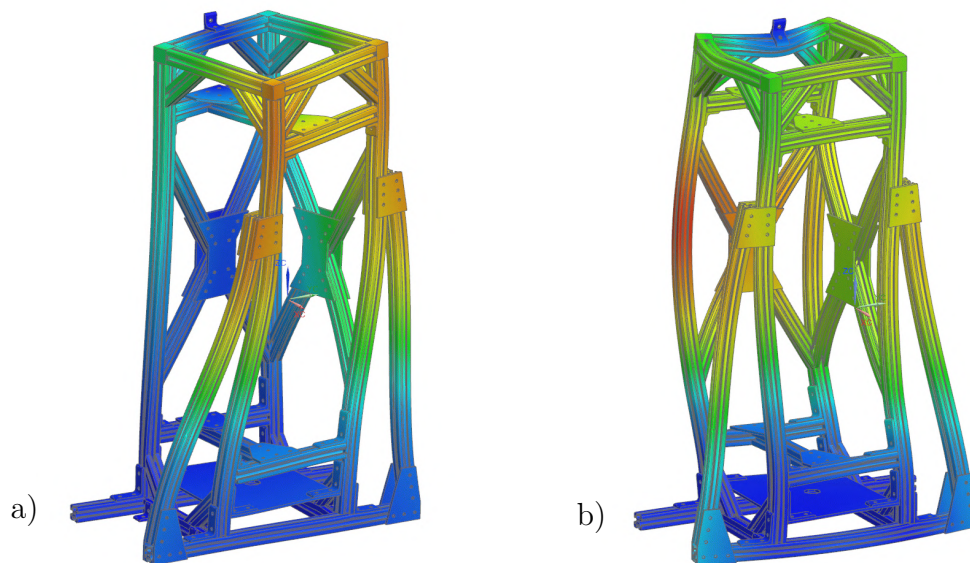


Figure 4.7 Modal shapes of the VARM's final frame with added braces for increased rigidity: (a) first mode of vibration at 89 Hz showing non-symmetrical torsion; (b) second mode of vibration at 138 Hz showing lateral bending.

reducing its overall mass, thus contributing to increased natural frequency values. Nevertheless, the predicted mode shapes presented on Fig.(4.6) were still relied upon to understand the frames displacement modes and add the adequate hardware such as cross-braces and wall fixtures to stiffen the first few bending modes and add fixed boundary conditions to the frames upper section by fixing it to the adjacent wall. For this final, stiffer, version of the VARM, the effective shaft length had to be reduced from the initial 1000 mm to 805 mm to accommodate the added structural components. This will cause the first critical speed to be higher than previously estimated. However, it can be brought down again by adding a mass-rotor to the shaft as shown by the disk presented in Fig.(4.2)(b).

4.3 Monitoring systems

As this project is part of a bigger one aiming to create a digital twin of hydroelectric turbine, measurement methods for data acquisition follows the same principles and try to collect the same kind of physical quantities that can be taken from an actual, running hydraulic-turbine unit. These measurements are comprised of, but not limited to, components temperature, production current and tension, operating speeds, phases, and vibrations by means of accelerations and displacements. Due to the very small scale of the VARM test apparatus and its uniqueness in design, most instruments used to monitor its dynamics are different than those typically found on operating hydroelectric turbines. However, they still enable the sampling of key physical quantities for the development of a digital twin.

Of these quantities, temperature and current would not be practical and relevant for monitoring in our experiments. First, the temperatures of critical parts such as bearings and the shaft beam are not expected to vary significantly owing to the small loads generated from the experiments. Then, the VARM does not generate any electrical power, as its main objective is to duplicate mechanical phenomena; therefore, it is not equipped with an alternator whose output would normally be monitored in a hydroelectric generation unit.

4.3.1 Acceleration monitoring

Because the shaft is extremely thin, one way to obtain vibration data during operation is to use accelerometers directly on the bearing housing, as is often done in the industry according to *ISO Surveillance et diagnostic d'état des machines Groupes de production hydroélectrique* 2020. Even though the bearings are very stiff relative to the shaft and should not experience much absolute displacement, this location method will still provide us with response frequencies through FFT analysis during data post-treatment.

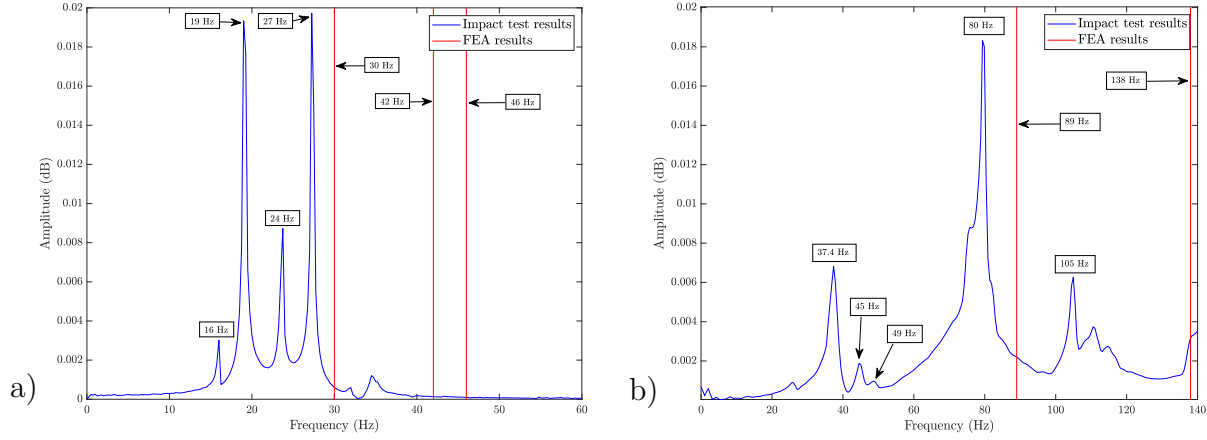


Figure 4.8 Comparison of the experimental impact test response (in blue) and the FEA-computed natural frequencies (in red) of the VARM's frame: (a) for the original design; (b) after the addition of braces to increase stiffness.

To achieve this, a PBC-352C33 uniaxial accelerometer is placed on the bearing outer housing as presented on Fig.(4.9) by the yellow arrows indicating the orientation of measurement. This accelerometer is very sensitive, $10.2 \text{ mVs}^2/\text{m}$, and has a high measurement frequency of up to 10 kHz, which is more than enough to capture the system responses in the low frequency range targeted which are expected to be at least two orders of magnitude lower.

4.3.2 Displacement monitoring

There are two ways to monitor beam displacement. The first one would be to place strain gauges at strategic locations where the bending stress is the highest in order to obtain the greatest signal output and then use beam theory to recreate the complete deformed shape of the beam. This also requires prior knowledge of the beam's expected bending modes, but this information can easily be obtained in our case from modal FEA of the shaft while knowing its excitation.

Two problems arise with this method. The first is again due to the shaft having a very small diameter. It requires an even smaller strain gauge and makes its installation a difficult process. The smallest sized gauges we could procure were still about half of the shafts diameter in width. This would still make any measurement readings sensitive to strains over a large portion of the shaft's perimeter. The second difficulty arises from the electrical connections between the gauges and the data acquisition apparatus (DAQ). Connecting the gauges to a DAQ would require slip rings. This type of connector allows electrical components to rotate with the machinery without entangling its connective wires. However, these slip

rings would add non-negligible noise to any signals they transmit.

The second method for measuring beam displacement is to directly use displacement sensors, such as linear variable differential transformers (LVDT) or ultrasonic probes. This would again be impractical for the VARM as the shaft would move in and out of the probes range while rotating because its displacement values are of the order of its own diameter, making the readings irregular. To overcome this issue, a laser scanning micrometer, typically used for the dimensional monitoring of manufactured parts, was used instead. This allows for touch-less measurements and extended freedom of installation, as the sensor positions are not limited by the movement of the shaft. Furthermore, laser technology is typically more precise and accurate than the other types of sensors.

Finally, the numerical integration of acceleration data from the accelerometers was also attempted to generate displacement values of the bearings housing. However, estimating the displacement from accelerometer data using numerical integration is challenging due to noise amplification, drift, error propagation, uncertain initial conditions, and integration technique selection. High-frequency noise and drift can compromise the accuracy, whereas errors propagate through integration. The initial conditions may be unknown or uncertain, and the choice of the integration method significantly impacts the results (Brown and Hwang, 1997).

A variant of the second option was chosen as the solution to these problems. This solution comprises a set of two IG-028 laser micrometers from Keyence, which are installed perpendicular to one another to gather data from the shafts displacement in both planar directions, as shown in Fig.(4.9)(c). The gate size is 28 mm and their precision varies from 0.03 mm to 0.17 mm depending on the setting distance between the transmitter and receiver, which can go as high as 1500 mm. With a sampling rate of up to 1 kHz, these sensors are sure to capture every relevant displacement response from the shaft.

4.4 Speed monitoring

To monitor the rotational speed in real time we use a proximity hall sensor that outputs an analog signal every time one of the four screws fixed to the jaw-coupling passes in front of it. Knowing that each peak in the output signal corresponds to a quarter-revolution and the associated timestamp, we can compute the rotational speed simply by counting the number of peaks over time. This computation is done every second which means that the speed is averaged over this time period. This calculation is done on MATLAB which is a matrix-based programming language for data computations (MathWorks, n.d.).

4.5 Data acquisition and treatment

Measurement data from the sensors presented above is collected and processed by a Lab-View program and treated and analyzed using MATLAB scripts. For fixed-rotational-speed experiments, both displacement and acceleration data are inspected via timeseries and FFT plots. For linear variable speed variation experiments where the rotational speed is slowly and steadily increased over time, a waterfall plot is used to inspect the data as it will show vibrational amplitude in the frequency domain with varying excitation frequency.

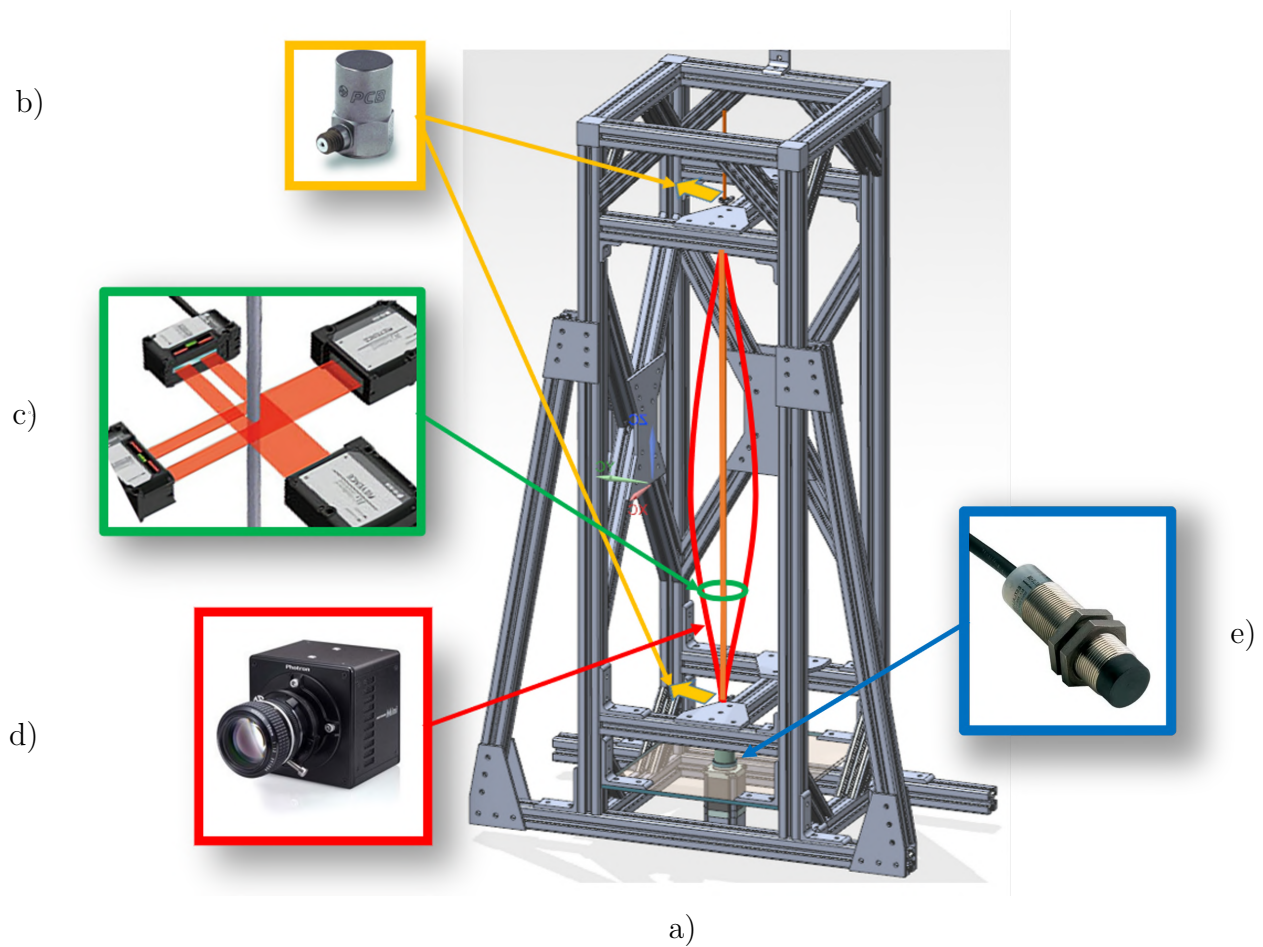


Figure 4.9 Schematic of the VARM with sensors and measurements: (a) CAD of the VARM. In red is a representation of the vibration envelop captured by the camera, in green the shaft's orbital captured by the lasers and in yellow the measurement direction of the accelerometer; (b) PBC-352C33 accelerometer placed on the bearing housing (PCB Piezotronics, 2023); (c) Keyence IG-028 laser micrometers positioned above the lower bearing to monitor the shaft's lateral displacements (Keyence, 2023); (d); High-speed camera monitoring the shaft vibration envelop; (e) Omega E57-08GE03-C proximity sensor.

4.6 High-speed camera

As previously shown on Fig.(4.9), one camera is used to record the shaft's displacement during experiments. When a mid-span disk is mounted, only the bottom or upper portion of the shaft can be recorded for processing. The recording speed in frames per second must be precisely documented from the cameras control software SpinView, and a picture of the shaft at rest is taken to evaluate the pixel size by measuring the length of the visible portion of the shaft in the cameras frame. The video processing script, running on Python 3 and using using OpenCV library, performs a series of operations to analyze the recorded footage of the rotating shaft. The main objectives of this analysis are to calculate the rotational speed of the shaft and to determine the maximum displacement of the shaft against its height position. The processing steps can be outlined as follows:

1. Reading the input video: The script reads the input video file and retrieves its properties, such as frame rate and dimensions, which are required for further processing and analysis.
2. Preprocessing: Each frame of the input video is preprocessed by converting it to grayscale, applying Gaussian blur to reduce noise, thresholding to create a binary image, and performing edge detection using the Canny algorithm. This series of operations isolates the shaft's contours from the background, enabling accurate displacement and speed calculations.
3. Contour extraction: The script identifies the contours of the shaft in each frame. This information is used to determine the position of the shaft axis over time.
4. Rotational speed calculation: The rotational speed of the shaft is computed based on the displacement of the shaft axis between consecutive frames. The speed is updated throughout the analysis, and its value is displayed on the processed video.
5. Processed video output: The processed video, which includes the calculated rotational speed displayed in the lower right corner, is saved to a file for further examination and validation of the results.
6. Maximum displacement analysis: By analyzing the position of the shaft axis in each frame, the script determines the maximum displacement of the shaft against its height position. This information is plotted to visualize the shaft's behavior under mass-unbalance conditions.

The video processing script provides a comprehensive analysis of the shaft's vibrations and behavior, enabling the validation of experimental results with computational simulations and facilitating further investigation into potential improvements and applications for such systems.

Lastly, Fig.(4.6) presents an image example of an output processed video file.

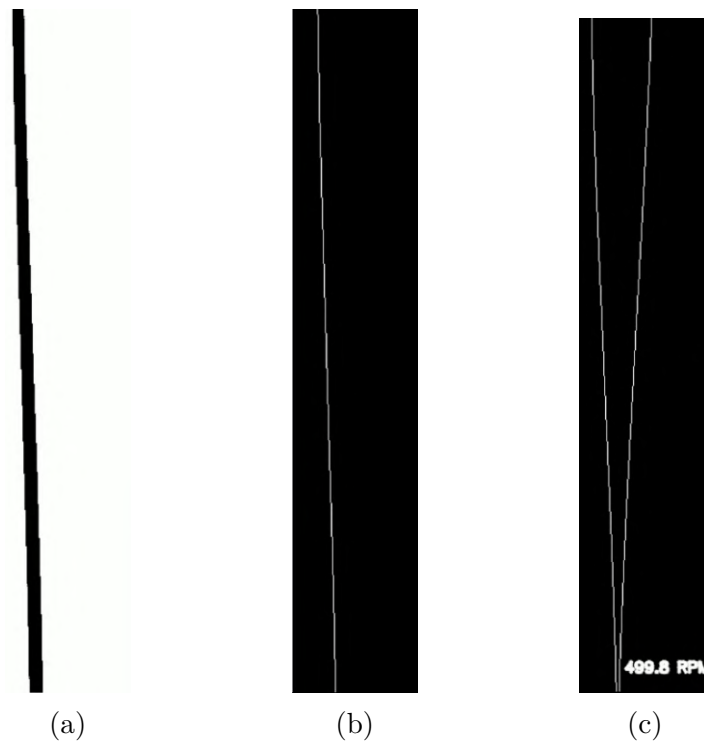


Figure 4.10 Frame samples of the video processing output from a high-speed camera monitoring the lower portion an unbalanced test of the VARM.(a) Binary color frame of the shaft.(b) Moving center-line of the shaft. (c) Vibration envelop and calculated rotational speed from displacement.

CHAPTER 5 METHODOLOGY II:: FINITE ELEMENT SIMULATIONS

Numerical data generation is indispensable for the success of the digital twin project, as it serves as a vital reference data source. Furthermore, numerical simulations played a pivotal role in the design of the VARM, serving as the primary source for estimating quantities such as expected loads and natural frequencies. Finite Element Analyses (FEAs) were instrumental in making informed decisions concerning various aspects of the VARM's design, with a particular focus on shaft dimensions, materials, and the vertical positioning of bearings. Throughout the course of this project, Finite Element Method (FEM) simulations were conducted using NX Simcenter 3D 2022 software.

5.1 Finite element model properties

5.1.1 Element selection

Choosing the right elements for the VARM shaft's FEM is crucial for efficient and accurate modelling. The choice primarily boils down to beam or solid elements.

Beam elements, being uni-dimensional, are apt for modelling long, slender structures like the shaft. Characterized by their simplicity, they only require two nodes at the ends and are defined by the shaft's cross-sectional attributes. They allow for efficient computation and accurately capture the shaft's bending, shearing, and torsional behavior. However, they fall short in representing variations in cross-sectional or material properties across the shaft's length.

On the other hand, solid elements, being three-dimensional, provide a detailed representation, capturing potential variations in material properties or stress distributions, but at a higher computational cost.

Given the VARM shaft's uniformity and consistent cross-section, beam elements offer an optimal balance between efficiency and precision. But for simulations with an added disk, 3D solid elements are preferred. Therefore, a hybrid model using 1D CBEAM for the shaft and 3D CTETRA for the disk is employed. Tab. (5.1) below shows the materials used for each component and their properties.

Table 5.1 Material Properties of the different components of the FEM

Part	Material	Density (Kg/m ³)	Young's Modulus (GPa)	Poisson ratio
Rod	Bronze 544	8885	110.3	0.34
Disk	Steel	7810	206.9	0.33
Shaft Collars	Aluminum 2024	2780	73.1	0.33

5.1.2 Modeling of boundary conditions and constraints

The idea behind the stiffening of the frame discussed in the last chapter was to make it as rigid as possible, firstly to mitigate the possibility of resonance with the vibrating shaft but also to simplify the FEMs by having to model only the bearing-shaft system. This would greatly simplify the models but would only work under the assumption that the bearings housings could be deemed fixed in relation to the frames high stiffness.

Screws are again left out of the model for simplicity as their added mass is negligible in this case. In the actual VARM, they are used to fasten the disc to the shaft collars which also use screws to grip around the shaft at the desired height. In the FEM, the contacts between the disk and collar and between the disk-collars and the shaft are modeled with RBE3 elements as shown on Fig.(5.1).

5.1.3 Modeling bearings

Simcenters rotor Dynamics module offers a great deal of flexibility for modeling bearing components. However, due to time, complexity constraints and lack of data, the VARM bearings will be modeled as lumped-parameters in the FEM. This can be done by creating a mesh-point at the centerline of the shaft and constraining its closest nodes to this new point by CBEAR2 1-dimensional connections which is automatically identified as bearing components by the solver. This allows to manually input bearing proprieties such a stiffness and damping in all 6 degrees of freedom. In our case, since the shafts equivalent stiffness is relatively low and the bearings used on the VARM do not allow for substantial rotation in the planes perpendicular to the rotation axis, the stiffness and damping parameters of the bearing are set to very high values in the order 10^6 N/m. This is done to model them as being perfectly rigid in relation to the soft shaft. Actual bearing stiffness and damping values are, in most cases, not available from manufacturers and can be very challenging to experimentally evaluate. In our case, for a shaft installed in long bearings being 30.5 mm

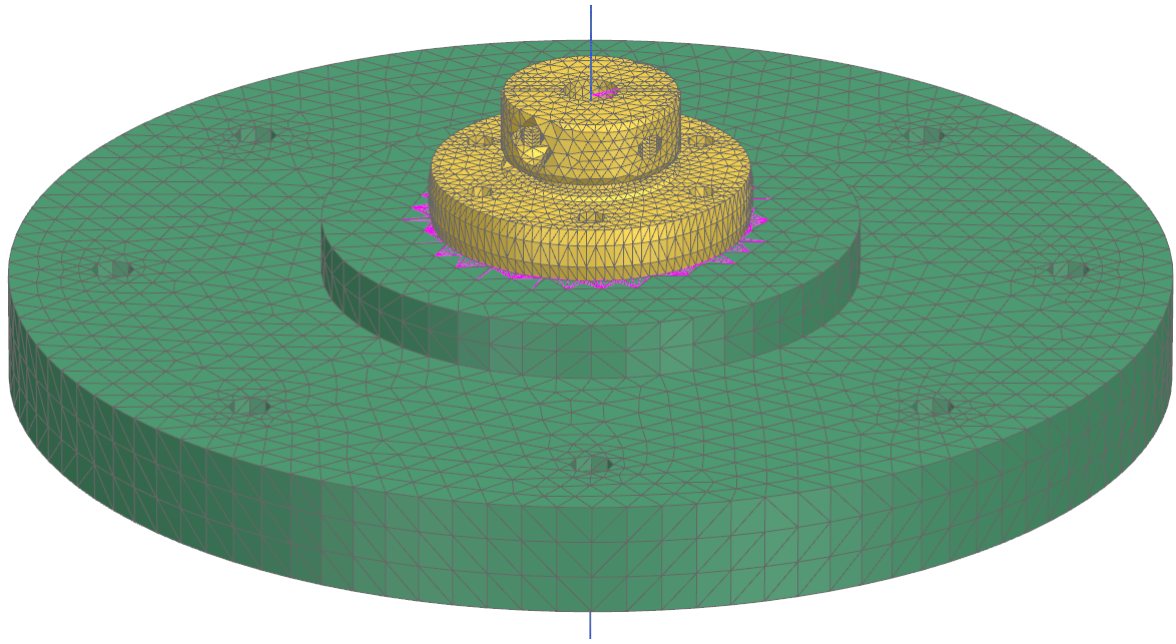


Figure 5.1 Finite element model of the shaft-disk assembly. In green is the disk mesh with 21468 CTETRA(4) elements, in gold is the upper shaft collar mesh with 25351 CTETRA(4) elements and in blue is the 1D mesh of the shaft composed of 805 CBEAM elements. The RBE3 elements modeling the contact connections between these meshes are shown here in purple.

high, such a modeling hypothesis is well acceptable as a first estimation (Kurowski, 2022).

5.2 Convergence study

Here we perform a preliminary convergence study to determine the optimal elements sizes for the model.

This is done in two parts. the first one refers only to the element size of a FEM of the shaft fixed at both ends by infinitely stiff bearings. Tab. (5.2) presents the results of the iterative procedure where the number of elements is gradually increased by refining the mesh. The error calculated refers to the difference between the FEA results and the analytical result for the fixed-fixed beam with a length of 805mm using Eq. (4.3). Beyond 101 elements, the error drops below 1 % and even though it is almost twice as small for the case with 101 elements than with 805 elements, the later will be preferred for the rest of the FEAs as this smaller compartmentalisation of the beam allows for more accurate modeling of contacts between nodes of different parts when add-ons are included such as a steel disk.

Table 5.2 Convergence analysis of a beam modeled by 1D CBEAM elements by comparing the first natural frequency computed and its the analytical formulation counter-part.

Number of Elements	First Natural Frequency (Hz)	Error (%)
2	27.03	12.069
101	30.86	0.390
805	30.94	0.651

For this hybrid model, we establish the mesh convergence according to Fig. (5.2) where the error seems to be converging to a minimum. Here, the error designates the difference between the resulting natural frequency values and the true value which is defined as the natural frequency of the model comprised of the most elements. During this analysis, a FEM of the shaft made of 805 elements (1mm elements) was used and only the CTETRA(4) elements modeling the disk and collars were refined. The first natural frequency converges to 7.5 Hz after 157 056 total 3D elements.

Consequently, the final FEM used for analysis contains 805 CBEAM elements for the shaft, 71 295 CTETRA(4) elements for the disk and collars and 1201 RBE3 elements to model the connections. This lower number of elements is preferred to lighten the computational time of the solver while still providing adequate accuracy.

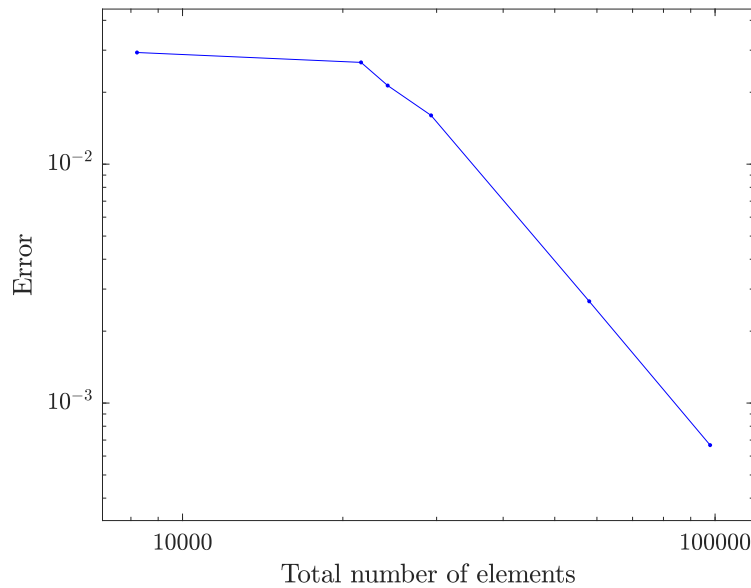


Figure 5.2 Convergence analysis of a hybrid 1D-3D FEM of the shaft with a disk mounted at its mid-span using shaft collars. The natural frequency converges to 7.5 Hz after 157 056 elements in total.

5.3 Finite-elements analysis

For each type of analysis, modal and harmonic, the same FEM can be used which includes the mesh and bearings modeling. What differs the most are the types of loading required for each analysis and the type of output they provide.

5.3.1 Complex Modal Simulation

Modal analysis, particularly in the context of rotating machinery such as the VARM, gains critical importance due to its ability to predict system behavior. While the mode shapes of a simple beam are straightforward to calculate manually, the addition of complex shaped disks introduces complexities that call for the use of FEA. The gyroscopic effects from the disk's rotation and the possibility of resonance with operational speeds make a more sophisticated approach most welcome. Complex modal FEA allows us to accurately determine the shafts natural frequencies and mode shapes. This is vital to avoid, or engage when desired, critical speeds or resonance conditions thus ensuring safer and more efficient operations. In the VARM setup, finite element-based modal analysis provides a deeper understanding of the dynamic interactions between the shaft and the disk. By revealing the system's vibration characteristics, it aids in devising effective vibration control strategies during experimenta-

tion. The outputs of this simulation are natural frequencies and their respective mode shapes displacement in unit length.

To be able to run properly, this analysis only needs the boundary conditions to be determined which will be the same for the other types of analysis. To do so, we define the added mesh points, which were connected to the shaft, as fixed in all degrees of freedom except in the z direction which represents the axis of rotation.

5.3.2 Harmonic Response Simulation

This analysis technique simulates the system's behavior when subjected to sinusoidal loads of varying frequencies by computing the steady-state solution of the system. In the context of the VARM, harmonic response analysis can be instrumental in evaluating the shaft's vibration response at different operating rotational speeds. This analysis provides us with displacements, accelerations, and stress values at every meshed point/element of the shaft.

To run, this analysis needs loading parameters. Two types of loading will be used: mass unbalance with specific mass, eccentricity, and rotating speed and misalignment in the form of an offset between the bearings where the upper bearing of the VARM is shifted in the lateral direction by a fixed amount.

CHAPTER 6 EXPERIMENTAL RESULTS

Multiple series of tests in different configurations were performed to demonstrate the data acquisition and treatment, and experimental capabilities of this VARM project. All were performed using the same bronze shaft and same ball-bearings under different load conditions. The two configurations experimented on are (1): when the VARM is only equipped with the shaft and (2): when the steel disk previously discussed is installed at the mid-span of the shaft. The laser micrometers are installed at a height of 160 mm and 60 mm above the lower bearing for the shaft-only and shaft-disk configurations respectively. The recording frequency of the DAQ device NI-9234 is set to 2000 Hz for all tests and sensors. Even though the micrometer lasers are rated for 1000 Hz, they were set to output a continuous analog signal which can be sampled at the preferred DAQ device's frequency. The motor output speed monitored by the proximity sensor is constant throughout the experiments with slight deviations of $\pm 3\%$. For each configuration, the following series of test are conducted:

1. Free response test: This is done by pulling the shaft in the transverse direction and releasing it to record its free response while responding to an initial displacement condition. The data is then used to extract the natural frequency by computing the average period between oscillations and the structural material damping using logarithmic decrement of the reducing vibration amplitude.
2. Ramping speed test: This test consists in gradually increasing the rotational speed from 0 to a fixed value which depends on the VARM configuration being tested.
3. Constant speed test: This final experiment is comprised of multiple tests each carried out at a constant speed over a duration of 60 seconds. Some cases require a longer duration of data acquisition to be able to clearly observe all the phenomena at play.

This protocol is performed for a straight and a misaligned arrangement of the bearings for both the shaft and the shaft-disk configurations.

Except for the free response tests and the bearings reaction force denoted by F_B , the measured data is normalised by the shaft characteristics and the experiment parameters yielding the following quantities for acceleration from the accelerometer, displacement from lasers, time and frequency respectively:

$$\alpha = \frac{\ddot{x}_B}{\omega_n^2 D}, \quad \delta = \frac{x}{D}, \quad \tau = \omega_n t, \quad \Omega = \frac{\omega_e}{\omega_n}, \quad (6.1)$$

where D is the shaft's diameter, ω_n rotor's first natural frequency, ω_e the excitation frequency, x_B the acceleration at the bearing and x the lateral displacement.

6.1 Shaft-only configuration

6.1.1 Free response test

This test was performed by pulling the non-rotating shaft by its mid-span point in the transverse direction towards one of the laser to make its displacement amplitude maximal for the other laser. The data recorded in time domain, Fig.(6.1), shows an exponentially decreasing amplitude in displacement, as we would expect from a minimally damped system such as this. The logarithmic decrement method described in section 2.3.1 is used here to compute the natural frequency of the response and its damping coefficient using 2 points from the time-series separated by n periods. The following parameters presented in Tab. (6.1) used to perform the calculation with (t_1, x_1) and (t_2, x_2) being the first and second data points, respectively, where $t_2 > t_1$. This resulted in a damping ratio of 1.2% and a damped natural frequency of 24.9 Hz.

The natural frequency obtained can also be observed from a frequency domain plot of the free response passed in an FFT algorithm. Fig.(6.2) shows such a plot for both the displacement and acceleration recorded during the test. The first peaks for both signals are in alignment with the damped natural frequency value previously computed with a clearly dominant peak at 25 Hz for both signals. Only the acceleration signal shows a second peak at 26.5 Hz which could originate from the structural components of the bearings or the frame. However, comparing these values with the numerical estimation obtained in section 5.2 for the 805 mm long shaft indicates a much bigger discrepancy of about 30 %. This is probably due to the fact that the bearings are not as rigid as we assumed. The values previously computed supposed Fixed - Fixed boundary conditions which could be matched to infinite stiffness. This is not the case in practice where the lower-than-infinite stiffness of the bearings causes a reduction in the natural frequency of the shaft. Furthermore, bearings are optimized to support rotating machinery and are designed with a minimal radial loading force which is

Table 6.1 Parameters used for, and results of the logarithmic decrement computation of the free response of the shaft.

t_1 (s)	x_1 (mm)	t_2 (s)	x_2 (mm)	n	ζ	ω_d (Hz)
0.060	0.288	0.981	0.053	23	0.012	24.9

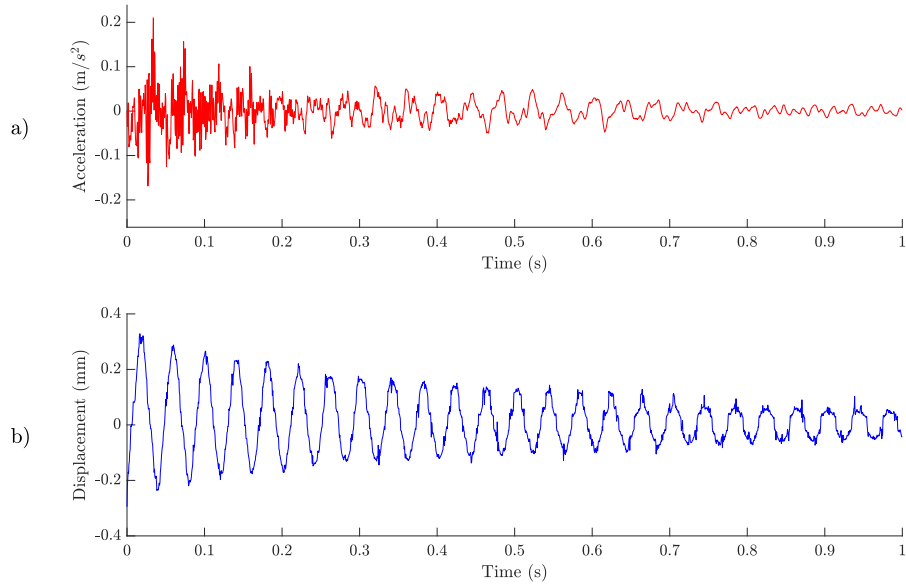


Figure 6.1 Timeseries plot of the recorded data from the free response of the shaft: (a) acceleration at the upper bearing housing measured by the accelerometer; (b) displacement at 160 mm above the lower bearing measured by the laser micrometer.

not attained at rest causing gaps and free-play in the bearings inner components which could explain the peak recorded at 26.5 Hz from the accelerometer placed directly on the upper bearing. These bearings have a minimal radial load of 127 N which is usually achieved by the shaft's own weight in a horizontal setup. The effects of these non-rigid bearings were not further explored as this endeavor would be outside the scope of this project that focuses on the design of a VARM.

6.1.2 Ramping speed test

For this test, the speed ramp lasted for 97 s and went from 0 RPM to 1170 RPM (19.5 Hz). Fig.(6.3) shows the waterfall plot resulting from the acceleration data recorded on the bearing housing. Many harmonics (diagonal lines) and resonant frequencies (horizontal lines) are visible throughout the spectrum. These harmonics represent sinusoidal behavior at frequencies equal to multiples of the excitation frequency. They are usually the result of defaults in the rotor line or bearings from manufacturing imperfections. Due to the high order of these harmonics, they probably stem from the bearings which have 8 rolling elements and many individual parts which could all interact to produce multiple vibrations during a single rotor cycle. The high number of resonant frequencies could also originate from bearings

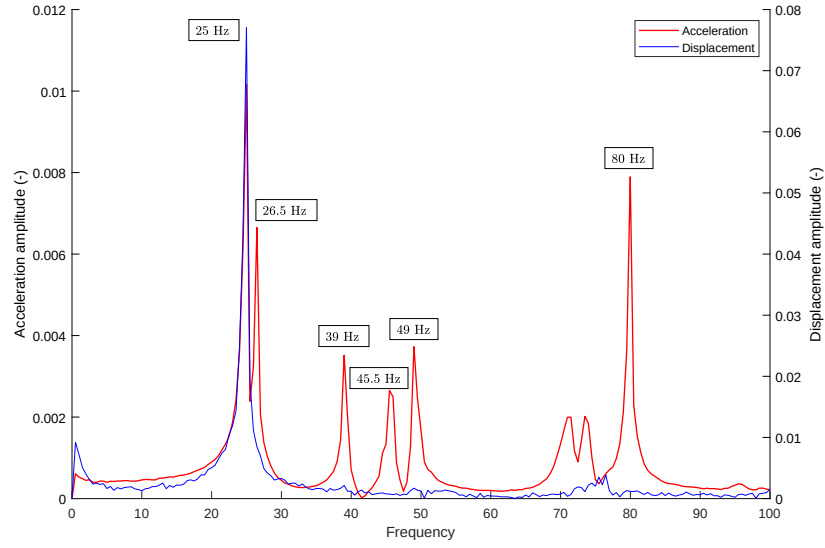


Figure 6.2 Frequency domain plot of the recorded data from the free response of the shaft. The red curve represents the acceleration signal monitored on the upper bearing housing external surface with a first main peak at 25 Hz and many secondary peaks indicating the presence of other resonance frequencies.

defects introducing non-linearities to the rotor. As previously mentioned in section 2.1, the vertical orientation of the experiment causes impacts between the bearing inner components during operations which could explain these harmonics.

Furthermore, shaft defects can also play an important role in determining these frequencies. Visual inspection of the drive-line did not uncover any cracks but rather an irregular shape of the previously thought straight shaft. This could lead to anisotropic properties causing harmonics. The previously obtained natural frequency of 25 Hz is only slightly visible at higher speeds. Yet, other resonant components at 45 Hz and 80 Hz are clearly visible from the acceleration waterfall plot.

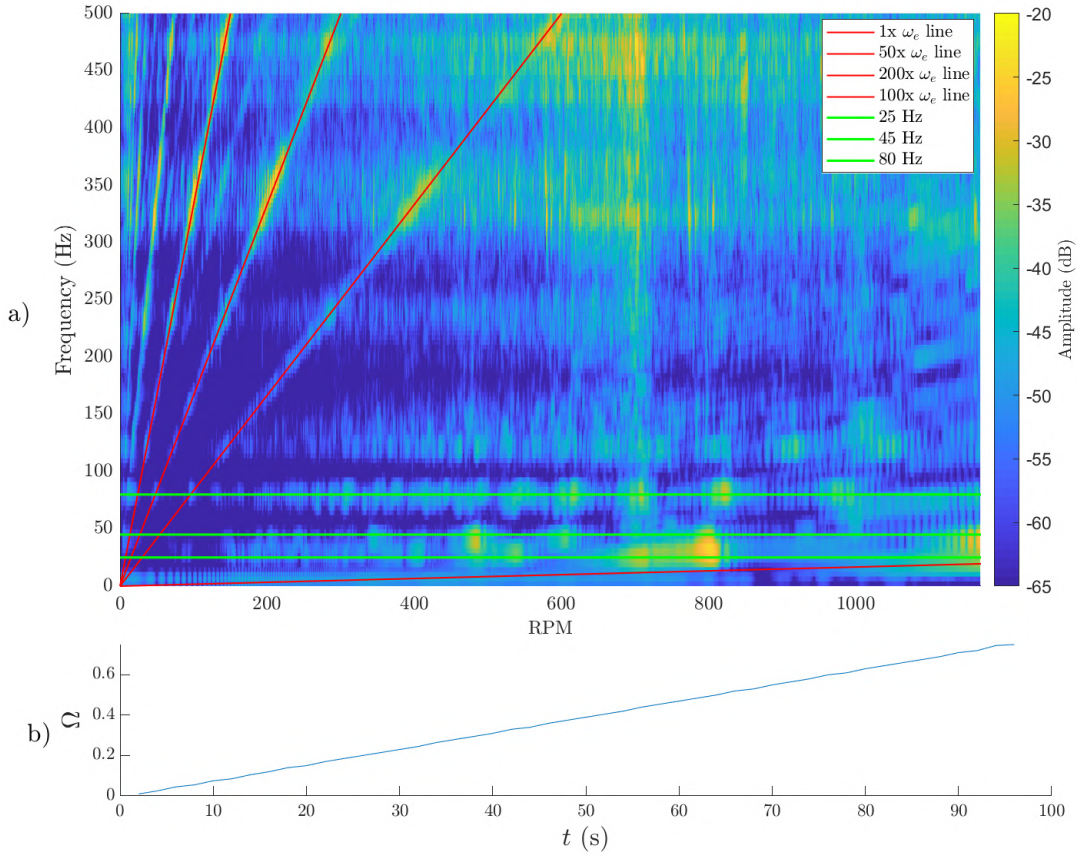


Figure 6.3 Ramping test frequency analysis. (a) Waterfall plot from accelerometer data. The red lines highlight a few excitation harmonics at 1x, 50x, 100x and 200x while the green ones highlight some resonant frequencies at 25 Hz, 45 Hz and 80 Hz. (b) Ramping speed profile over time.

6.1.3 Constant speed tests

In this series of tests, we rotate the shaft at rates between 180 (3 Hz) and 1200 RPM (20 Hz). Fig.(6.4) shows a sample of this series taken at 180 RPM (3 Hz). The orbit is obtained by plotting both displacements against one another. Its elliptical shape is typical of anisotropic stiffness stemming from imperfections in the system, and its orientation is determined by the bearing stiffness values in orthogonal directions (White, 2017).

We also use an FFT algorithm to plot the frequency domain response of the experiment. Fig.(6.5) shows such a transform from Fig.(6.4) and reveals many harmonics and sidebands in the response. Sidebands are additional frequency components that appear on either side

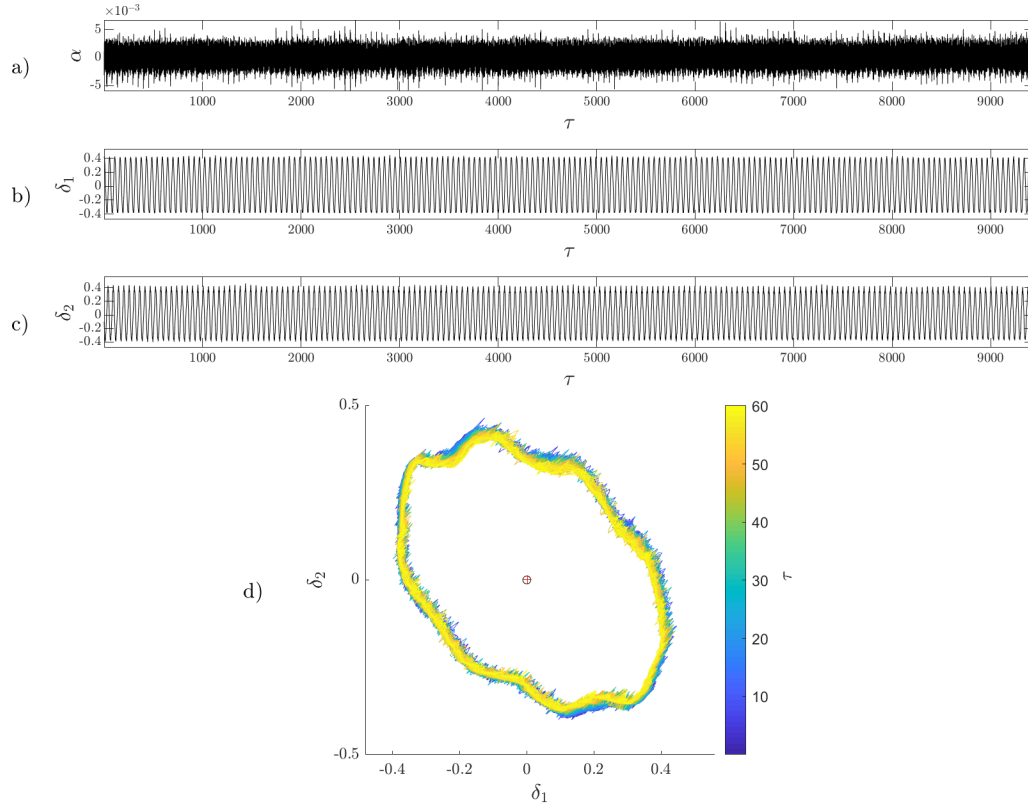


Figure 6.4 Acceleration and displacement data recorded at 180 RPM (3 Hz) over 60 seconds: (a) acceleration at the upper bearing; (b) displacement at 160 mm above the lower bearing in the x direction; (c) displacement at 160 mm above the lower bearing in the y direction; (d) shaft's orbit over the complete duration.

of the primary spectral peak. These sidebands are typically at frequencies that are slightly offset from the central peak and are the result of modulation or beating phenomena within the system being analyzed. This is present in the data from all experiments which points to mechanical imperfections in the VARM. Ball bearings are known to produce this effect but this could also be coming from structural looseness in the assembly, misalignment with the drive line or varying torque from the motor. Furthermore, harmonic peaks close to the natural frequency tend to have the biggest amplitudes. In this case, the 9x harmonic at 27 Hz is clearly the dominant frequency due to its close proximity to the natural frequency.

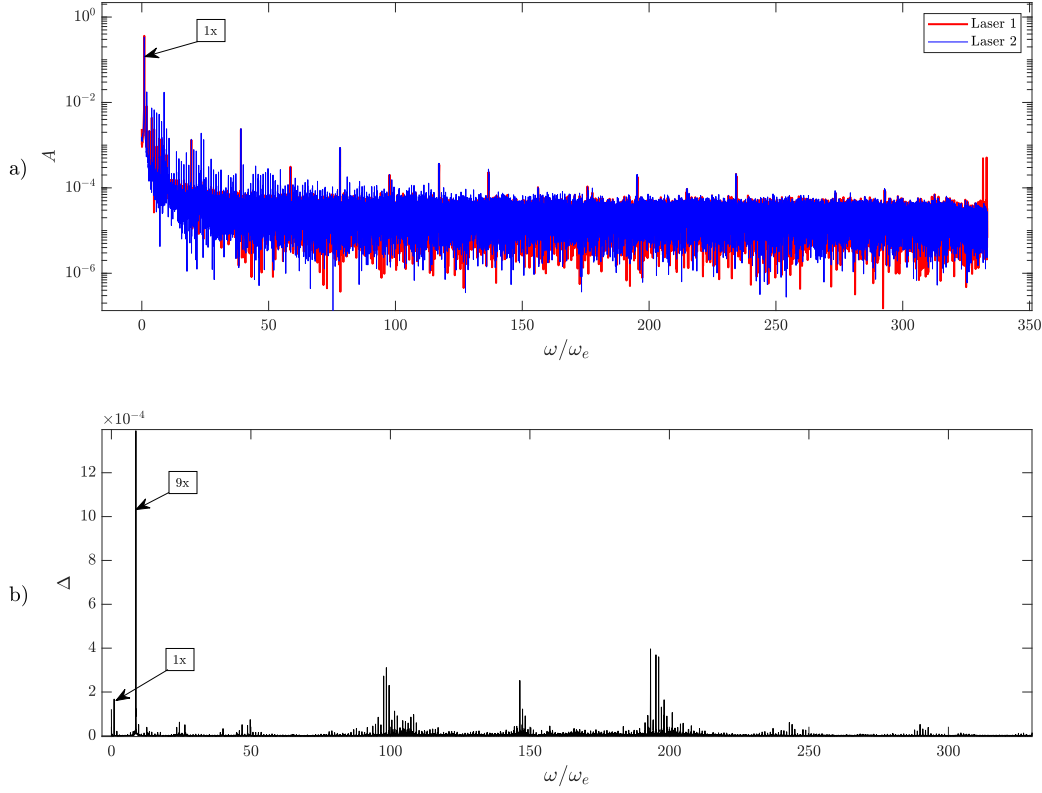


Figure 6.5 Frequency domain plots of the recorded data at 180 RPM (3 Hz) over 60 seconds: (a) displacements amplitudes in both directions; (b) acceleration amplitudes at the upper bearing.

Furthermore, the orbital paths gets smoother due to the bearing reaction loads increasing with the speed and getting closer and closer to the designed minimal load, thus reducing impact behaviors and gaps in the bearings internal mechanisms. This is all the more evident when comparing the orbits at 180 RPM (3 Hz), 420 RPM (7 Hz), and 1200 RPM (20 Hz) in Figs. (6.4,6.6-6.7) and by approximating the bearings reaction force for each experiments as shown in Fig.(6.8). This is done only as a first approximation by using Euler-Bernoulli beam theory to compute the value of a centered force applied on a fixed-fixed beam. Knowing the deflection at a specific location on the beam, we can use the following formulation to obtain the acting force:

$$F_B = \frac{48EI\delta(z)}{z^2(3L - 4z)}, \quad (6.2)$$

with E being the Young's modulus, I the surface polar moment, L the distance between the two bearings, δ the displacement and z the displacement's location along the shaft. However, this method assumes that the displacement is mostly due to a point-force from a potential static unbalance at the shaft's midspan and does not take into account its deformed shape. Actual modeling of the shaft's deviant profile would be more appropriate to estimate the loads and deflections but this would be outside the scope of this project as relevant techniques to do so are still being researched and developed.

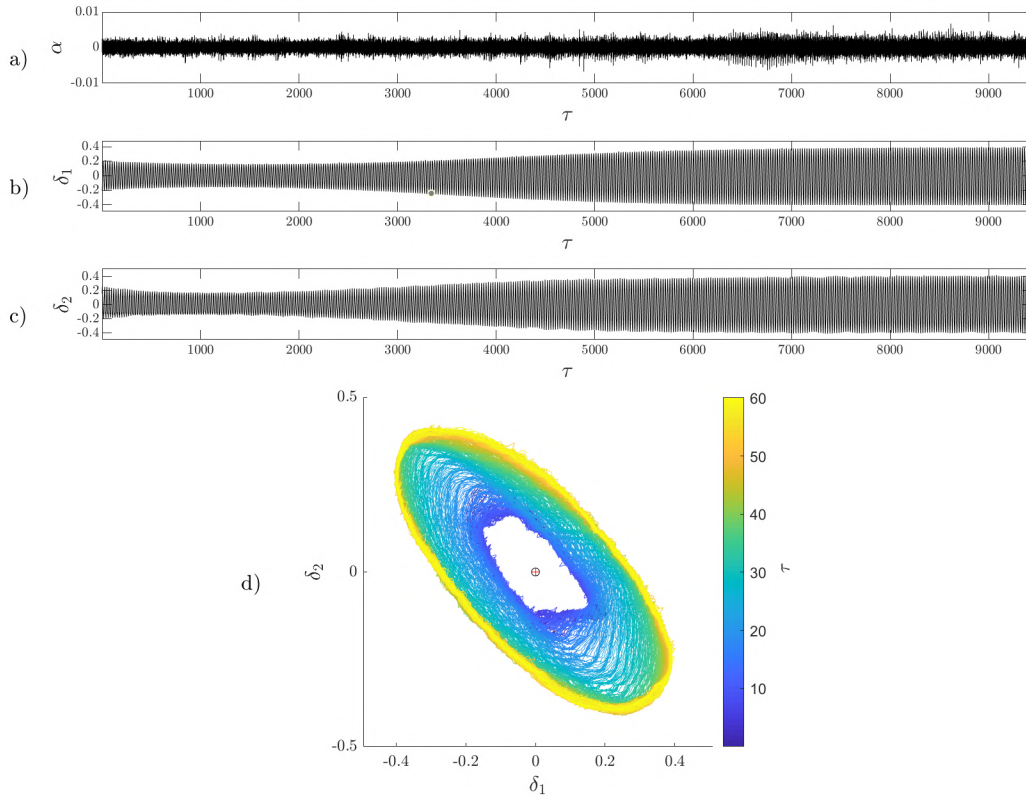


Figure 6.6 Acceleration and displacement data recorded at 420 RPM (6 Hz) over 60 seconds: (a) acceleration at the upper bearing; (b) displacement at 160 mm above the lower bearing in the x direction; (c) displacement at 160 mm above the lower bearing in the y direction; (d) shaft's orbit over the complete duration.

As we get closer to the natural frequency, the orbit becomes more and more elliptical as shown in Fig.(6.7) taken at 1200 RPM (20 Hz). These figures also hint at another phenomenon taking place. The displacement signals seem to exhibit amplitude modulation. By repeating the experiments over a longer duration, 180 s instead of 60 s, it becomes clear that this is in fact

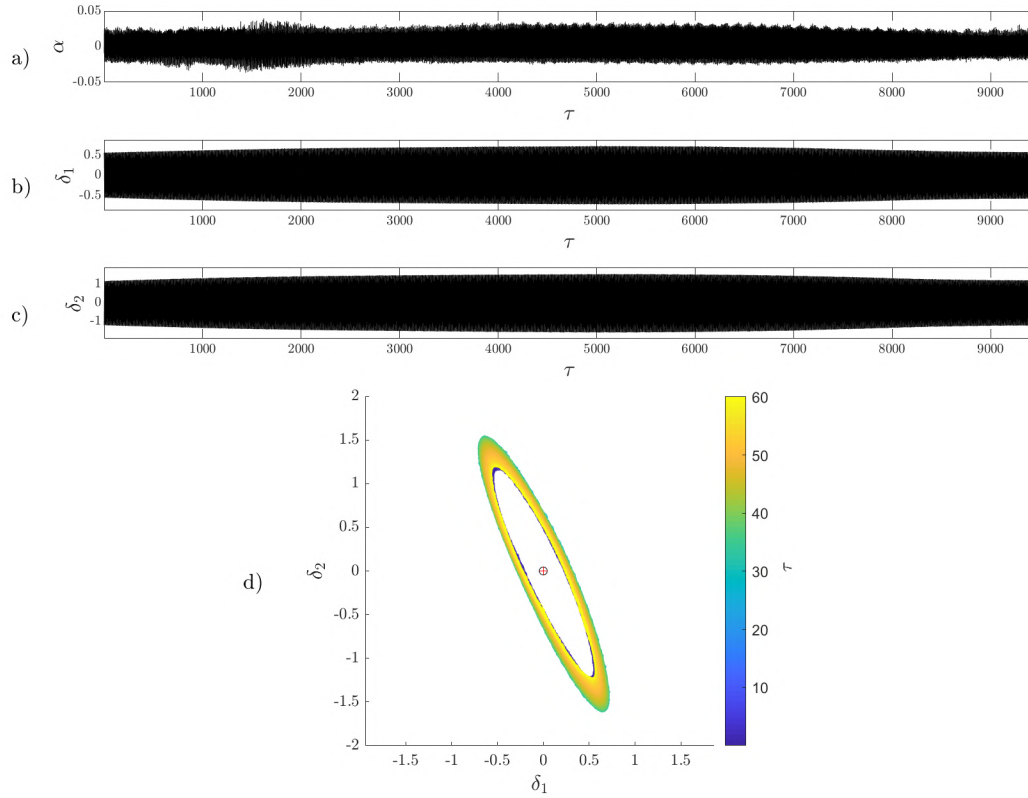


Figure 6.7 Acceleration and displacement data recorded at 1200 RPM (20 Hz) over 60 seconds: (a) acceleration at the upper bearing; (b) displacement at 160 mm above the lower bearing in the x direction; (c) displacement at 160 mm above the lower bearing in the y direction; (d) shaft's orbit over the complete duration.

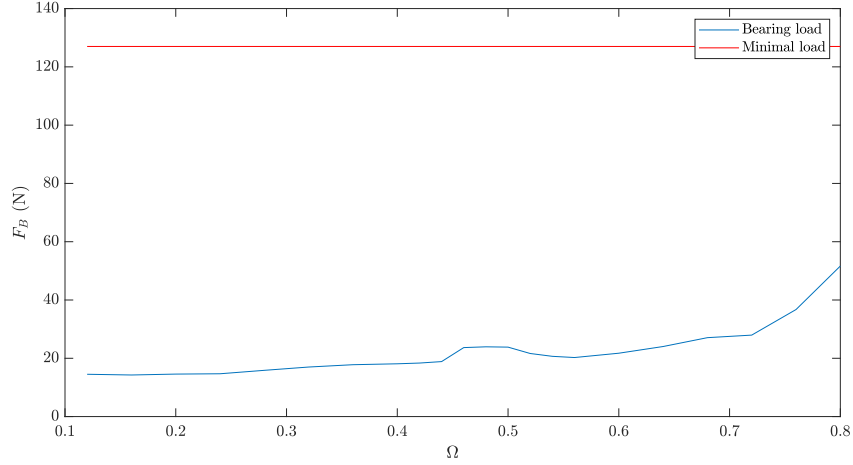


Figure 6.8 Bearings reaction force over increasing excitation speeds in blue and the required loading force in red.

beating occurring in the system at certain speeds. Beating in rotating machinery vibrations results from slight frequency differences between components. It's visible in vibration data as periodic amplitude variations in time series. Fig.(6.9) shows such results but only at 420 RPM (6 Hz) as to not overload the motor during a long period.

Fig.(6.10) shows the evolution in the root-mean square amplitude (RMS) of both acceleration and displacement measurements for each experiment. As expected, the closer the rotation speed gets to the natural frequency, the faster the amplitude increases. However, a rather visible increase in amplitude around half the natural frequency seems to be present in the acceleration data. In non-linear systems, secondary resonant frequencies can be observed below the first natural frequency depending on the order of the non-linearity. These frequencies would be equal to ω_n/O with O being the order of the non-linearity and ω_n the first natural frequency of the linear components (Nayfeh and Mook, 2008). In our case, peaks in amplitude at $\omega_n/2$ could indicate non-linearities of second order. This can also be seen in the bearings reaction force in Fig.(6.8). This non-linear behavior could be coming from multiple source: the ball bearings due to factors such as clearance, contact deformations, and damages; large shaft deformations; intermittent or unsteady contacts from the jaw-coupling components.

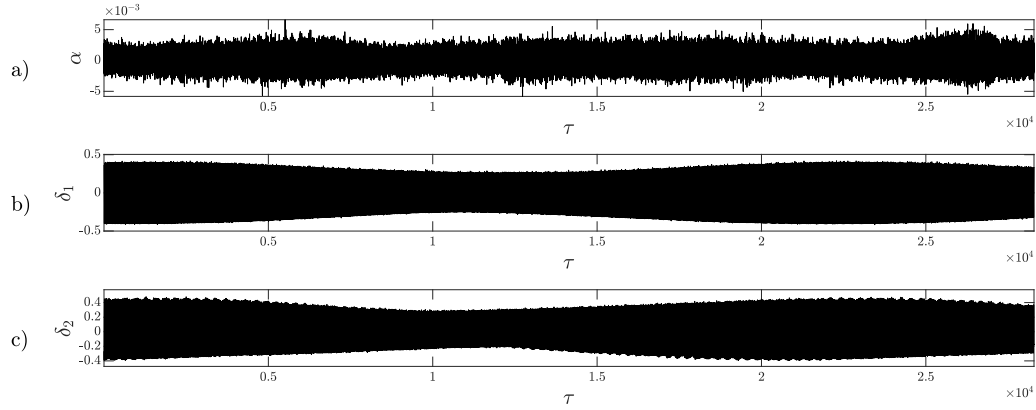


Figure 6.9 Acceleration and displacement data recorded at 420 RPM (6 Hz) over 180 seconds showing beating in the response: (a) acceleration at the upper bearing; (b) displacement at 160 mm above the lower bearing in the x direction; (c) displacement at 160 mm above the lower bearing in the y direction.

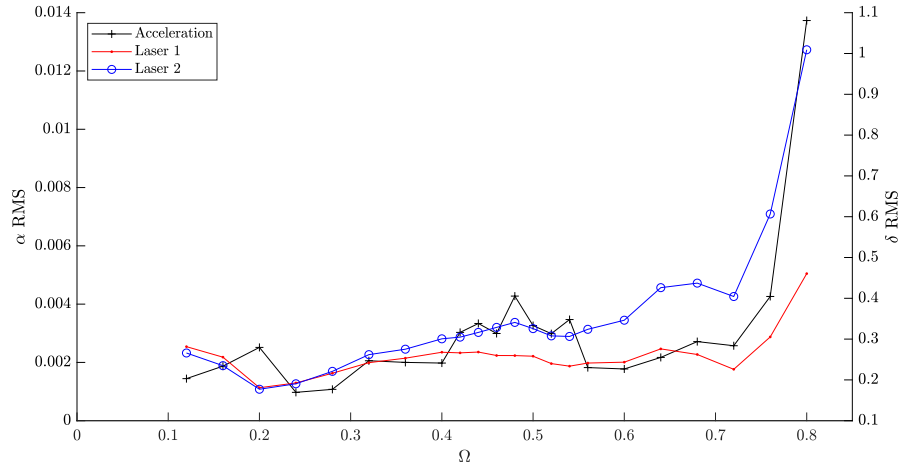


Figure 6.10 RMS amplitude of acceleration and displacement measurements over multiple experiments from 180 RPM to 1200 RPM. The red and blue lines represent displacement RMS value in orthogonal directions. The black line represents acceleration RMS on the bearing housing.

6.1.4 Constant speed tests - Misalignment

Here we present experimental results pertaining to a configuration of the shaft where the upper bearing is shifted by 18 mm in the x direction. Data recorded at specific speeds is similar to the previous case, displaying beating, harmonics and side bands. However, the motor could only reach 1140 RPM (19 Hz) in this configuration due to the additional load applied on the motor's drive line caused by the misalignment generating a static load on the shaft.

The following Fig.(6.11) shows the misaligned shaft's response at 780 RPM (13 Hz). The shaft's orbit does not follow an elliptical path but rather a knot-like shape. Muñoz-Abella et al., 2022 obtained similar results from experimental data of a cracked shaft causing significant drop in stiffness in one of the lateral direction. As mentioned before, no visible cracks are visible on our shaft yet this anisotropy in stiffness directions could be derived from its bent geometry. Looking closer at the timeseries, for a 10 seconds range, from the same experiment in Fig. (6.12)(b-c) we see that this is period doubling, clearly visible from the second laser signal while Fig. (6.12)(d-e) clearly shows dominant peaks at 2x harmonics.

Remarkable responses can be observed at high speeds where the displacements and acceleration amplitudes seems to be collapsing and increasing at different rates throughout the duration of the experiments. This is particularly present at 1140 RPM as showed on Fig.(6.13). More research and testing is necessary to explain this uncommon behavior.

The RMS amplitudes of the the acceleration and displacement signals in Fig.(6.14) are not as regular as with the aligned configuration. While still exponentially increasing close to the natural frequency, both acceleration and displacements data also show peaks around half of the natural frequency before settling down only to increase again.

Finally, an estimate of the bearings reaction force is computed and presented in Fig.(6.15) which clearly shows that they still are not properly loaded to their design value. The bearings reaction force is superior here than with the aligned configuration which is in line with Saavedra and Ramirez (2004), who modeled misalignment as an additional constant static load on the shaft.

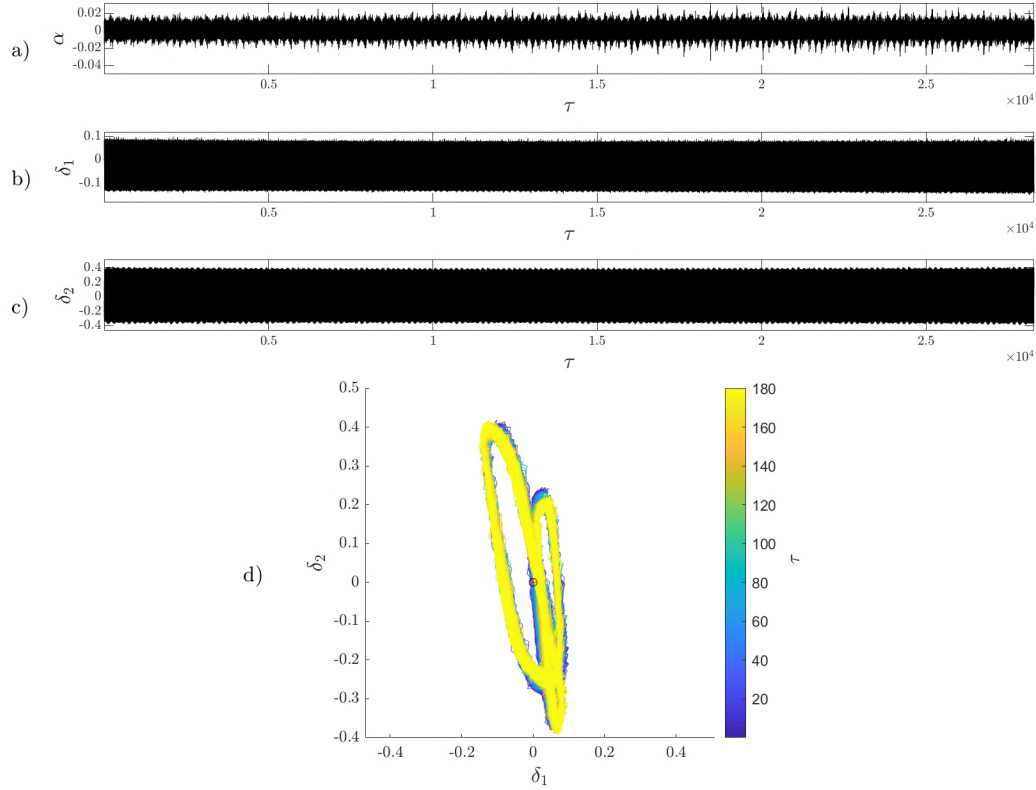


Figure 6.11 Acceleration and displacement data recorded at 780 RPM (13 Hz) over 60 seconds: (a) acceleration at the upper bearing; (b) displacement at 160 mm above the lower bearing in the x direction; (c) displacement at 160 mm above the lower bearing in the y direction; (d) shaft's orbit over the complete duration.

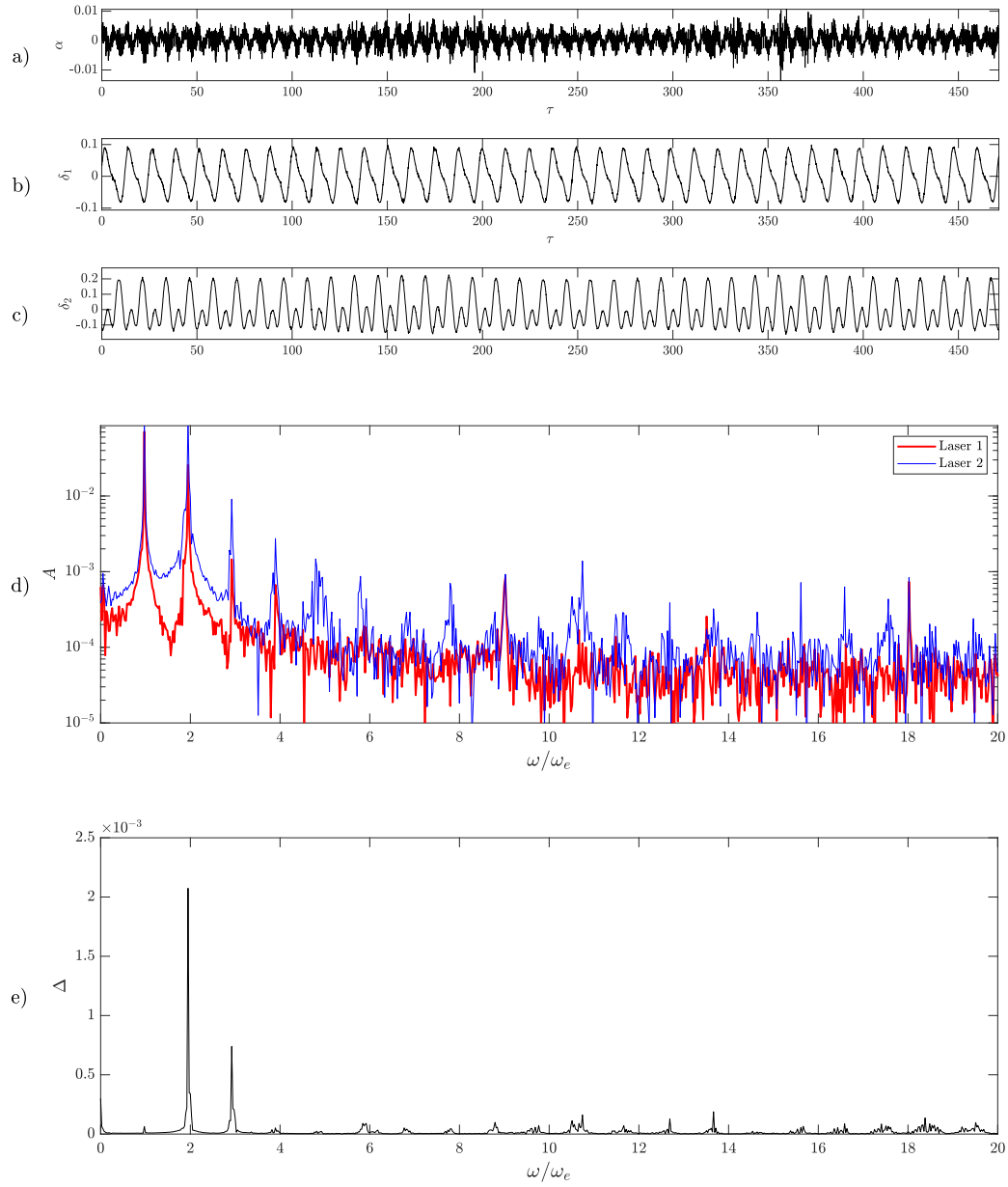


Figure 6.12 Acceleration and displacement data recorded at 780 RPM (13 Hz) for 10 seconds: (a) acceleration at the upper bearing; (b) displacement at 160 mm above the lower bearing in the x direction; (c) displacement at 160 mm above the lower bearing in the y direction; (d) frequency domain plot of the displacement signals; (e) frequency domain plot of the acceleration signal.

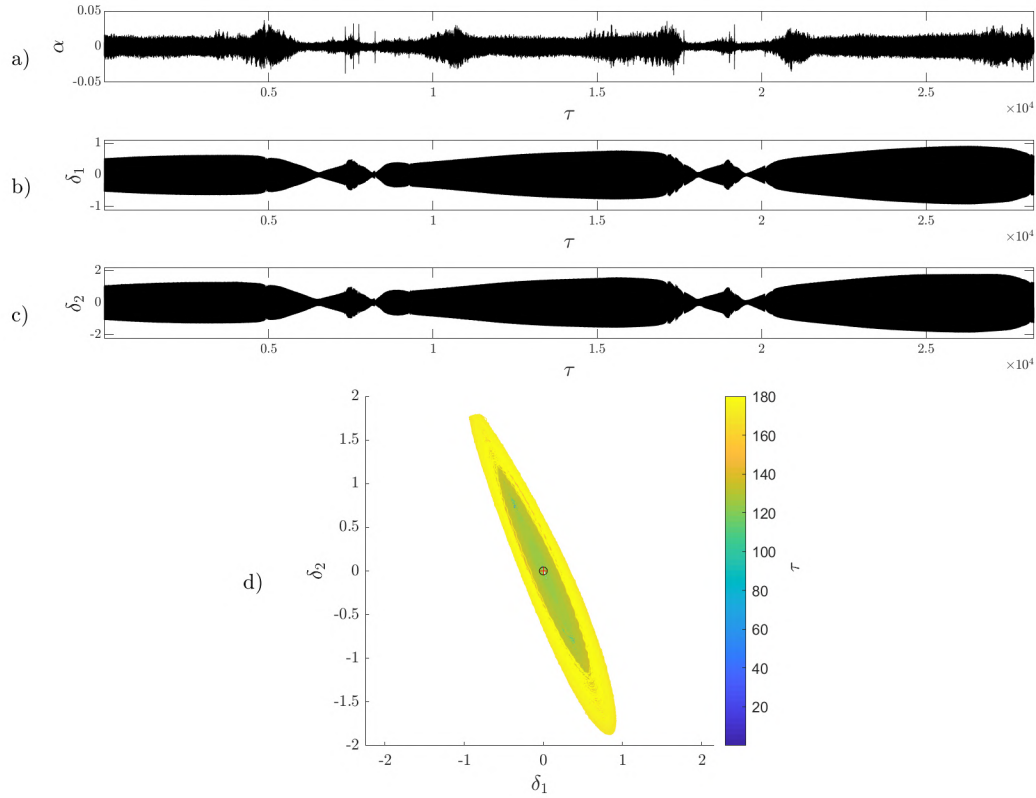


Figure 6.13 Acceleration and displacement data recorded at 1140 RPM (19 Hz) over 180 seconds. (a) Acceleration at the upper bearing. (b) Displacement at 160 mm above the lower bearing in the x direction. (c) Displacement at 160 mm above the lower bearing in the y direction. (d) Shaft's orbit over the complete duration.

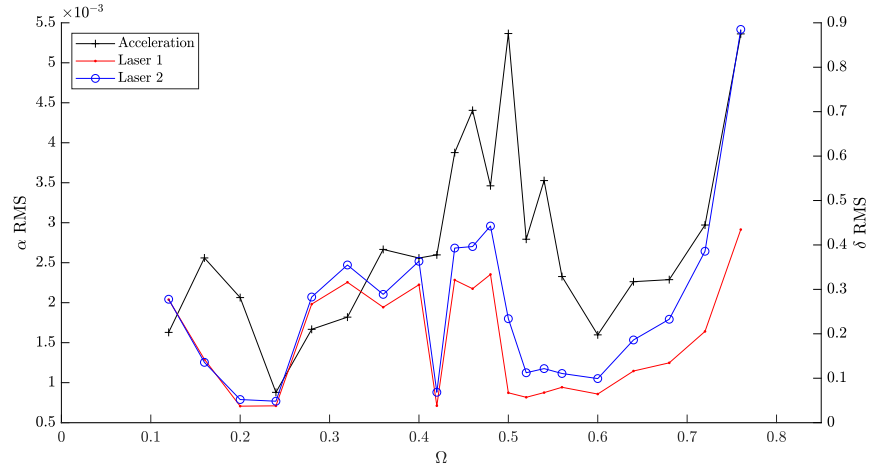


Figure 6.14 RMS amplitude of acceleration and displacement measurements over multiple experiments from 300 RPM to 1140 RPM. The red and blue lines represent displacement RMS value in orthogonal directions. The black line represents acceleration RMS on the bearing housing.

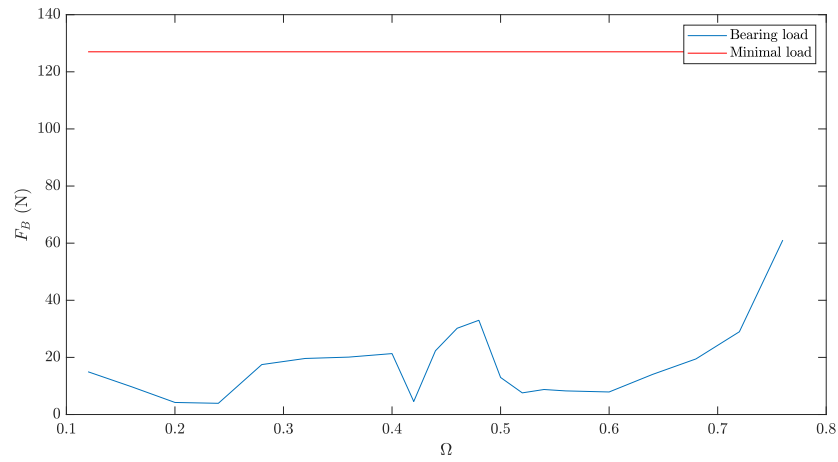


Figure 6.15 Bearings reaction force over increasing excitation speeds in blue and the required loading force in red.

6.2 Shaft - Disk

6.2.1 Free response test

As opposed to the shaft-only configuration, natural frequencies and damping are evaluated by impact tests on the shaft-disk configuration. This method allows to capture richer dynamics in the free response following the impact as it can excite multiple modes simultaneously. Tab. (6.2) shows the parameters and results from the application of logarithmic decrement on the free response as it was done in section 6.1.1 for the shaft-only configuration. The natural frequency is significantly lower than previously as it is to be expect after having installed a heavy rotor of 1.45 kg on the 0.281 kg shaft. Displacement measurements were made at a height of 60 mm above the lower bearing as the displacements at 160 mm in this configuration are too large to stay within the lasers measuring range.

Fig.(6.16) shows the decaying vibration amplitudes of this configuration on which the logarithmic decrement technique was used. It becomes clear that this method might not be suitable in this case since the free response clearly shows beating which could indicate a difference in stiffness, and thus in natural frequency, in both orthogonal directions for which the logarithmic decrement technique becomes inappropriate as it supposes only one frequency. A frequency domain analysis using FFT, with a resolution of 0.25 Hz, in Fig.(6.17) reveals two dominant peaks at 6.25 Hz and 7.25 Hz which could explain the beating behavior and, from Rao (1996), the beat frequency should be equal to the difference between these two peaks. That is to say beats should occur every 1 s, at 1 Hz. However, Fig.(6.16)(b) shows an approximate beating period of 1.33 s (0.75 Hz) which could indicate that more frequencies undetected by the lasers might be involved. Acceleration signal from the upper bearing in Fig.(6.16)(a) also shows beating with an approximate period of 0.17 s. This could be coming from structural components resonant frequencies as the associated frequency is about 6 Hz which is the difference between the two main peaks coming from the frame at 39.25 Hz and 45.25 Hz. Furthermore, a peak at 26.29 Hz from the acceleration data is still present in this new configuration. This reinforces the hypothesis that this resonance comes from the bearings inner parts when unloaded as this peak was also present in section 6.1.1.

Table 6.2 Parameters used for, and results of the logarithmic decrement computation of the free response of the shaft-disk configuration.

t_1 (s)	x_1 (mm)	t_2 (s)	x_2 (mm)	n	ζ	ω_d (Hz)
0.374	-0.24171	3.3335	-0.1247	21	0.005	7.1

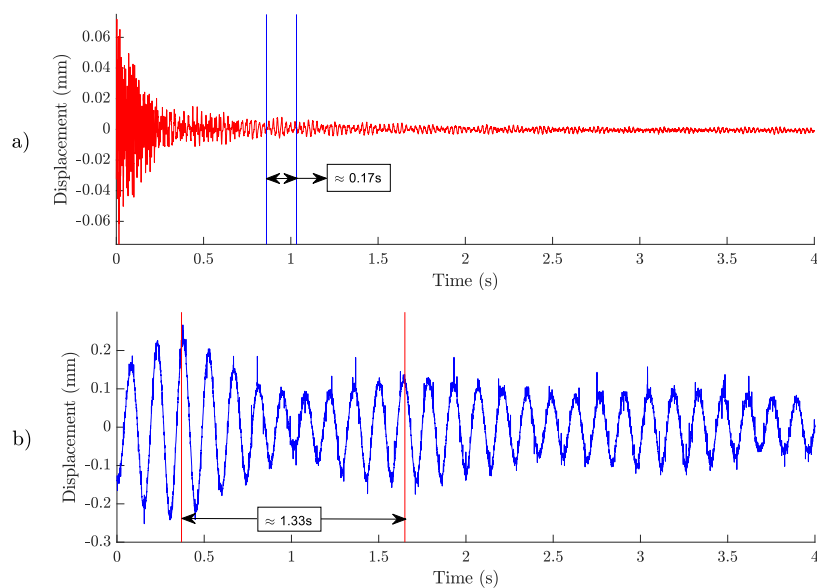


Figure 6.16 Timeseries plot of the recorded data from the free response of the shaft. (a) Recorded acceleration at the upper bearing housing showing low period beats. (b) Recorded displacement at 160 mm above the lower bearing showing high period beats.

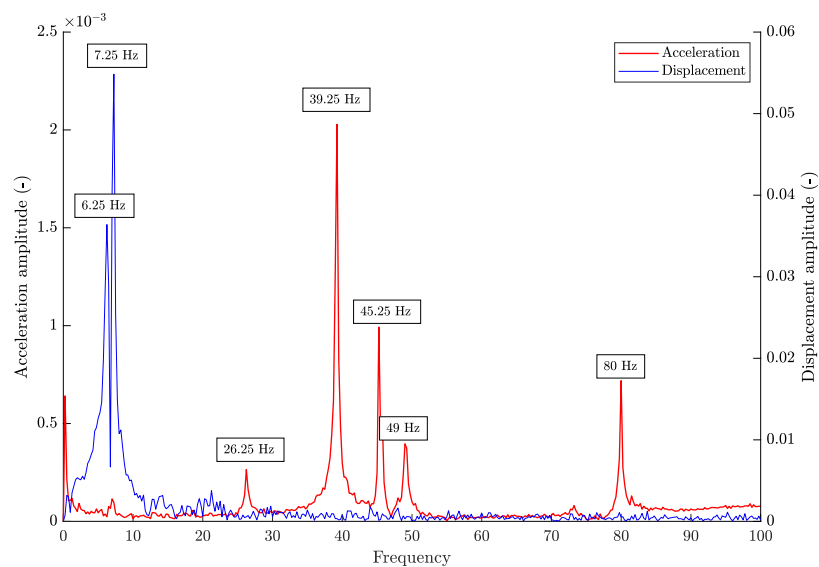


Figure 6.17 Frequency domain plot of the recorded data from the free response of the shaft. The red curve represents the acceleration signal monitored on the upper bearing housing external surface. The blue curve represents the displacement signal with two dominant peaks at 6.25 Hz and 7.25 Hz.

6.2.2 Ramping speed test

The ramping speed test on the shaft-disk configuration is done over a period of 164 s from 0 RPM to 390 RPM (6.5 Hz) and the results are presented on Fig.(6.18). As with the previous configuration, the same harmonics are clearly visible with the most dominant ones being at 50x, 100x and 200x. The fact that these harmonics are still present in such a different configuration than in section 6.1.2 sustains the argument that the bearings are the source of these vibrations. The first natural frequency of 7.5 Hz from the shaft-disk FEA computed in section 5.2 seems to be present but the lack of resolution makes it difficult to accurately identify. Remember that this computation was done considering that the bearings are properly loaded and exhibit very high stiffness akin to fixed-fixed boundary conditions which could be the case during the ramp experiment. Resonance around 45 Hz and 80 Hz are again visible in this configuration although the latter is very subtle. Looking back at the free responses of both Fig.(6.2) and Fig.(6.17) from the first and second configurations respectively, we see that peaks in amplitude are apparent from around 40 Hz to 50 Hz and around 80 Hz as well. This similarity in the response from different configurations of the rotating specimen and throughout multiple experiments strongly suggests that the structural frame of the VARM may be responsible for these resonant frequencies. We only need to go back to section 4.2.2 to confirm that these frequencies are in fact from the frame as they appear in the its free response in Fig.(4.8).

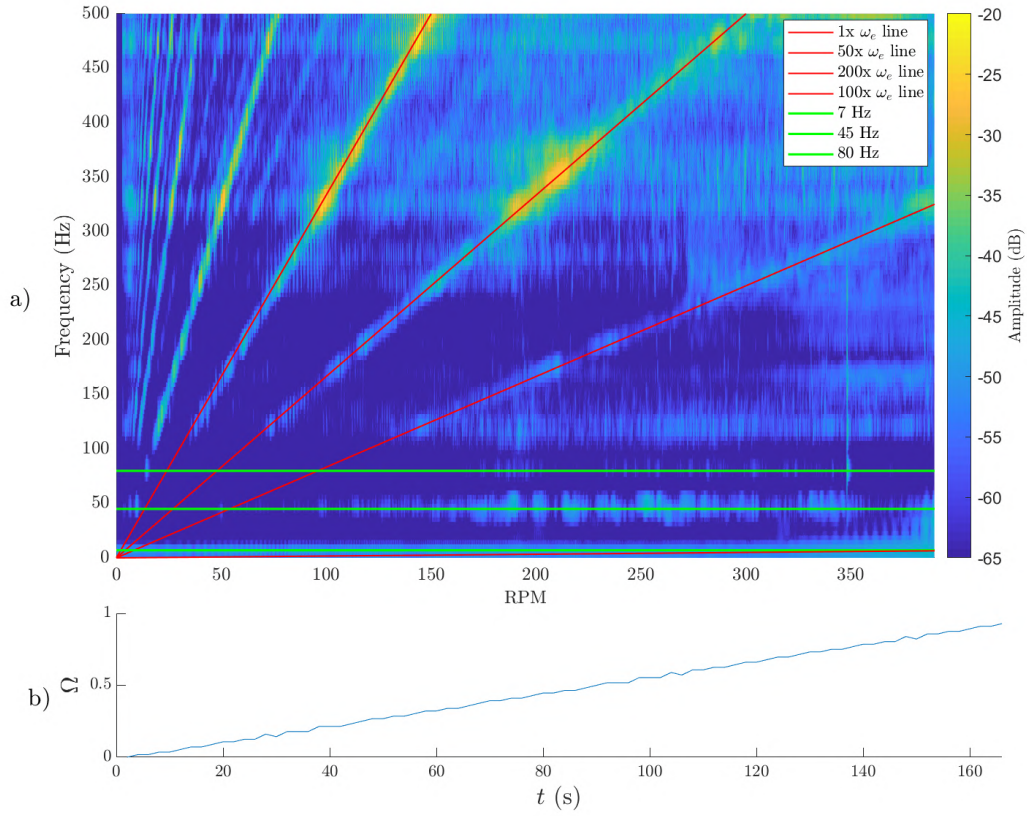


Figure 6.18 Ramping test frequency analysis. (a) Waterfall plot from accelerometer data. The red lines highlight a few excitation harmonics at 1x, 50x, 100x and 200x while the green ones highlight some resonant frequencies at 7 Hz, 45 Hz and 80 Hz. (b) Ramping speed profile over time.

6.2.3 Constant speed tests

In this section, all tests were realised over a duration of 120 s. No irregular orbits are observed and those recorded at low speeds are again circular and develop into ellipses as the speed increases which can be seen in Fig.(6.19) showing the orbits at 60 RPM, 270 RPM and 390 RPM. The absence of oddly shaped orbits when the unbalanced rotor is added could be resulting from the additional load caused by the unbalance which straighten the shaft irregularities and dominates the dynamics they cause. Additionally, the transient regime seems to last a significant amount of time due to the very low damping of the system. This is particularly visible in Fig.(6.20) showing the displacements amplitudes increasing over the first 60 seconds ($\tau \approx 2500$). If beats were present, their frequencies would be in the order of a few seconds considering the low excitation and natural frequency of the experiments but none are observable over a 120 s period.

Examining these responses in the frequency domain through an FFT algorithm, as shown in Fig.(6.21) at 390 RPM (6.5 Hz), reveals a similar behavior than with the shaft-only configuration. Harmonics and sidebands are present throughout the experiments pointing at amplitude modulation in the signals that could be caused by bearing imperfections, misalignment or a varying torque from the driving motor.

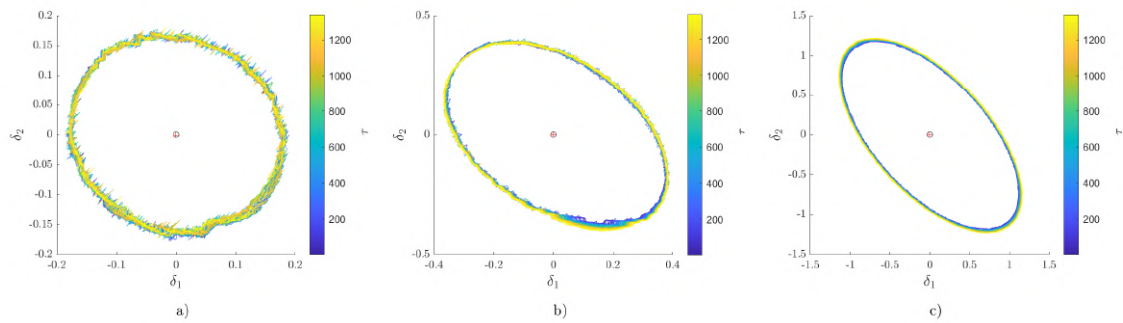


Figure 6.19 Displacement orbits at different speeds. (a) Orbit at 60 RPM (1 Hz). (b) Orbit at 270 RPM (4.5 Hz). (c) Orbit at 390 RPM (6.5 Hz).

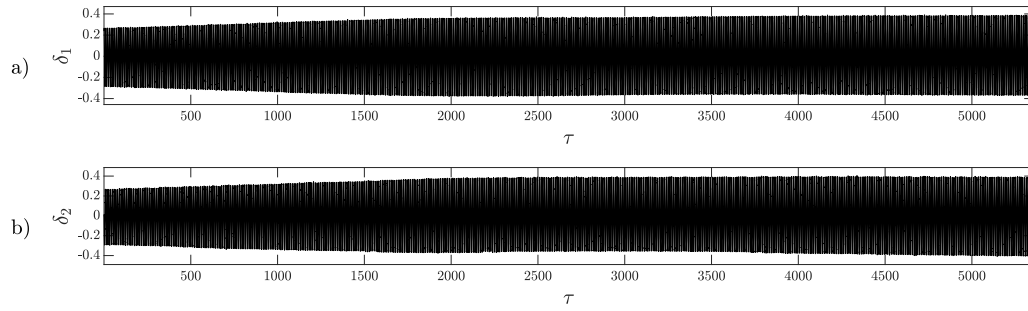


Figure 6.20 Displacement data recorded at 270 RPM (4.5 Hz) over 120 seconds showing a long transient regime. (a) Displacement at 160 mm above the lower bearing in the x direction. (b) Displacement at 60 mm above the lower bearing in the y direction.

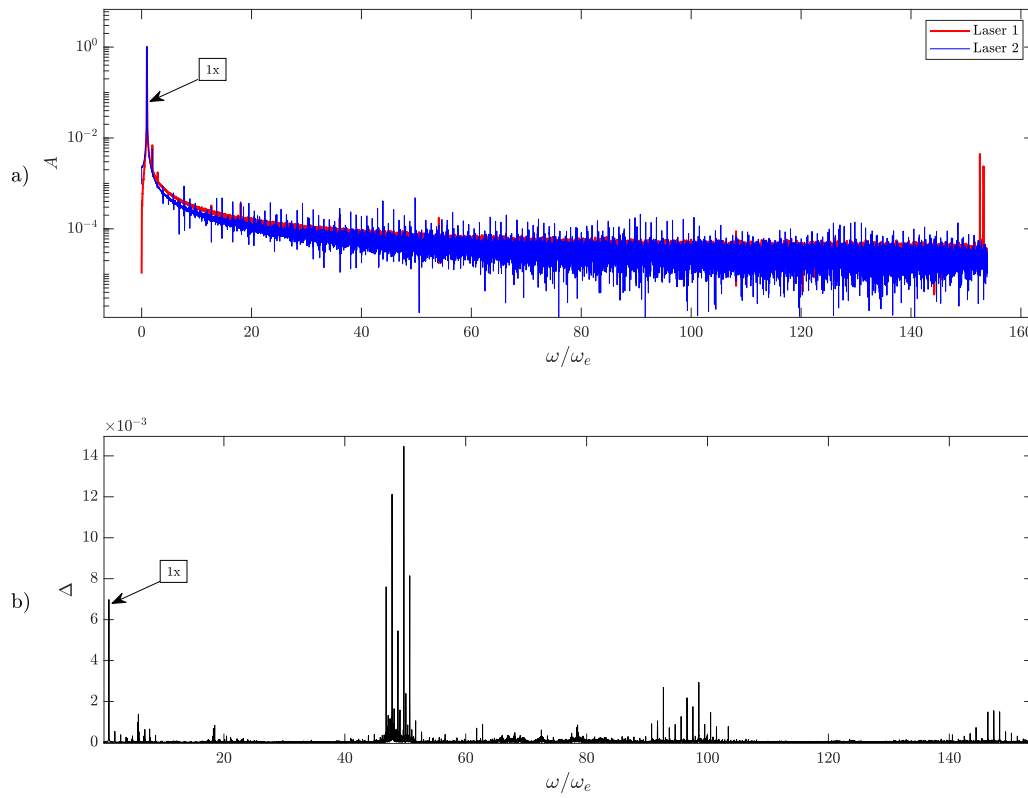


Figure 6.21 Frequency domain plots of the recorded data at 390 RPM (6.5 Hz) over 120 seconds. (a) Displacements amplitudes in both direction. (b) Acceleration amplitudes at the upper bearing.

The RMS amplitudes shown in Fig.(6.22) are similar in displacements to those previously observed in the shaft-only configuration with an exponential-like evolution as the speed increases and a visible sudden grow around half the natural frequency pointing again at non-linearities causing fractional resonance. In contrast, the acceleration recorded on the upper bearing housing does not follow the same trend. More research and investigation would be needed to uncover the reasons behind this behavior.

Finally, the estimated load on the bearings shown in Fig.(6.23) is much higher now with the unbalanced rotor than with the previous configuration. This could also explain the much more regular orbits observed here than in section 6.1 since the bearings are operating much closer and above their designed minimal load specification.

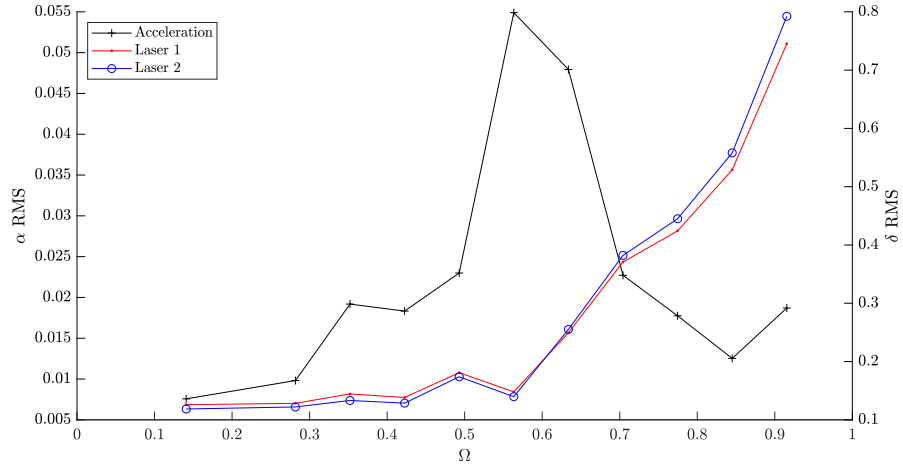


Figure 6.22 RMS amplitude of acceleration and displacement measurements over multiple experiments from 60 RPM to 390 RPM. The red and blue lines represent displacement RMS value in orthogonal directions. The black line represents acceleration RMS on the bearing housing.

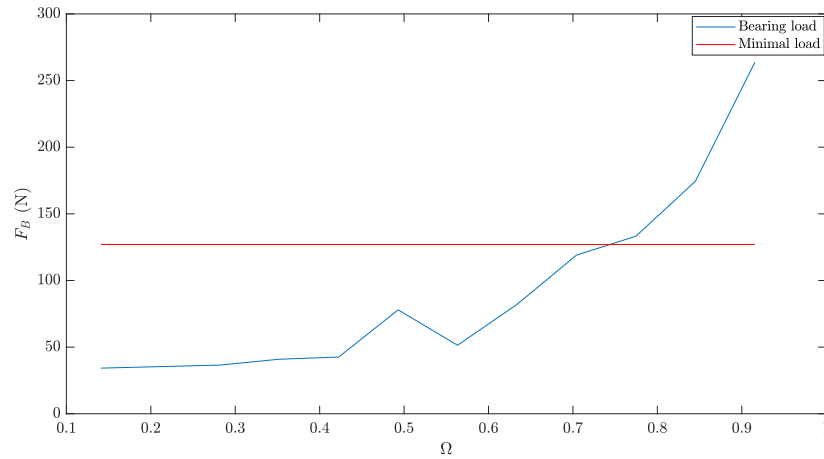


Figure 6.23 Bearings reaction forces over increasing excitation speeds in blue and the required minimal loading force in red.

6.2.4 Constant speed tests - Misalignment

The same settings were used to perform these tests where the upper bearing is shifted 18 mm to simulate misalignment. The general behavior of the VARM under this configuration is very similar to the previous case. The signals RMS shown in Fig.(6.24) have similar evolutions over the increasing excitation frequency. However, the displacements RMS values are noticeably lower while the acceleration RMS values are higher than with the aligned case. Finally, the bearings reaction forces displayed in Fig.(6.25) are visibly weaker here despite the additional static load caused by the misaligned bearing while still being substantially greater than in section 6.1. The effect of a potential secondary resonance at half the natural frequency is still slightly visible in the bearing reaction force.

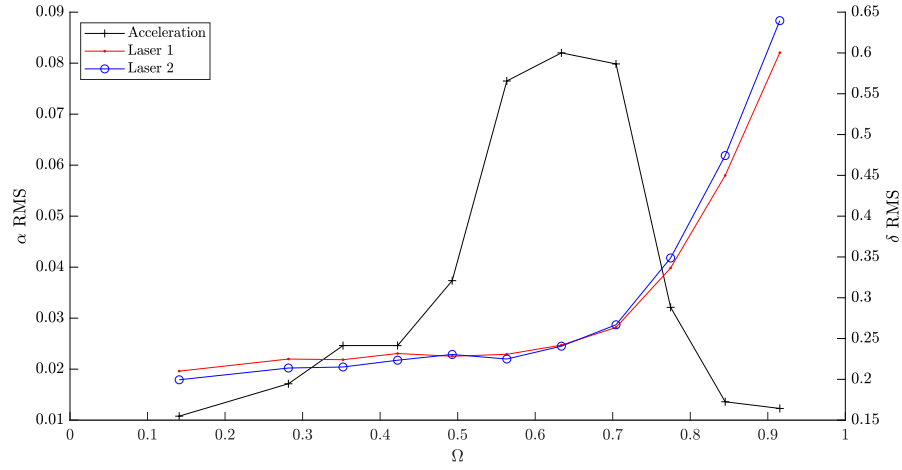


Figure 6.24 RMS amplitude of acceleration and displacement measurements over multiple experiments from 60 RPM to 390 RPM of the misaligned shaft-disk. The red and blue lines represent displacement RMS value in orthogonal directions. The black line represents acceleration RMS on the bearing housing.

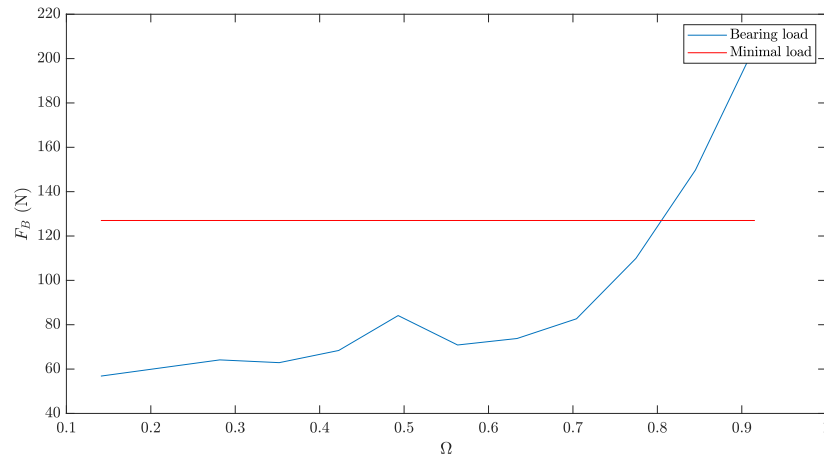


Figure 6.25 Bearings reaction forces over increasing excitation speeds in blue with misalignment and the required minimal loading force in red.

6.2.5 Tests summary

The objectives of these tests were to demonstrate the data acquisition capabilities of the VARM, but more importantly, its versatile arrangements and the rich dynamics they display. As expected from a vertically oriented rotating machine, linear models and assumptions have a hard time explaining the phenomenons that are observed due the high non-linearities that this orientation offers. The many imperfections in machine's components, notably in the shaft, unveil irregularities in the VARM responses that deviate from the more basic and already known analytical models such as the Jeffcott rotor. These imperfections are difficult to model and add an important element of unknown to the machine's dynamics, which could be statistically accounted for by a digital twin once enough data is obtained. These tests were selected based on the potential insights they might offer regarding the VARM's characteristics and behavior. The free-response and ramping-speed tests were designed to experimentally identify the first natural frequency (critical speed). While the ramping-speed test is standard for rotating machinery, a non-rotating free-response test was also conducted to compare the effects of stationary bearings with those of rotating bearings on the natural frequency. The constant-speed tests aimed to evaluate the VARM's response at different operating speeds and to identify as many parameters and phenomena as possible. This approach is similar to diagnostic tests on an actual hydroelectric turbine that displays irregular behavior due to malfunctions.

CHAPTER 7 FINITE ELEMENT RESULTS

This chapter presents the analysis of the VARM's FEM discussed in section 5. Due to the simplicity of the model in terms of bearing modeling and the straightforward analysis of its natural frequencies, only the results from a modal analysis and a harmonic response analysis are presented. Once more is known about the the VARM such as accurate bearing models trough research and experiments, the FEM can easily be upgraded by inserting these models in the form of tables or empirical equations into the software.

7.1 Modal analysis

The objective of the ramping speed experiment previously presented was to measure the natural frequencies of the rotor under rotation to provide contrast with the free response tests where the bearings were not loaded. Here, modal analysis will directly give the rotating natural frequencies. Moreover, the fact that the disk is located at the shaft's mid-span and the low rotation speeds applied mitigate the gyroscopic effect of the disk in the model. This makes the natural frequencies of the model constant over the studied speed range. Fig.(7.1) shows the results of the modal analysis on the model. The first natural frequency at 7.5 Hz closely matches the results from the previous chapter. However, the second natural frequency, corresponding to the disk's breathing mode shape at 12.1 Hz, was never observed in the experiments.

7.2 Harmonic analysis

This section presents the results of the harmonic results of the FEA in both aligned and misaligned configurations. The simulations were conducted over the same speed range as experimental shaft-disk configuration, 60 RPM (1 Hz) to 390 (6.5 Hz). A first series of simulations are ran with an unbalance of 0.08 kgm to achieve similar results as the experiments while the second series of simulations are loaded by a 18 mm lateral misalignment at the upper bearing. Because of the model isotropic characteristics, axis-symmetry and the lack of bearing defects and poor loading conditions, the orbitals are always circular and the frequency domain responses only show the excitation frequency, making these monitoring tools mundane in this case. Fig.(7.2) shows the displacement RMS for both the aligned and misaligned cases which is taken at 60mm above the lower bearing. As it was the case for the experimental results, the displacements of the aligned configuration is visibly superior than

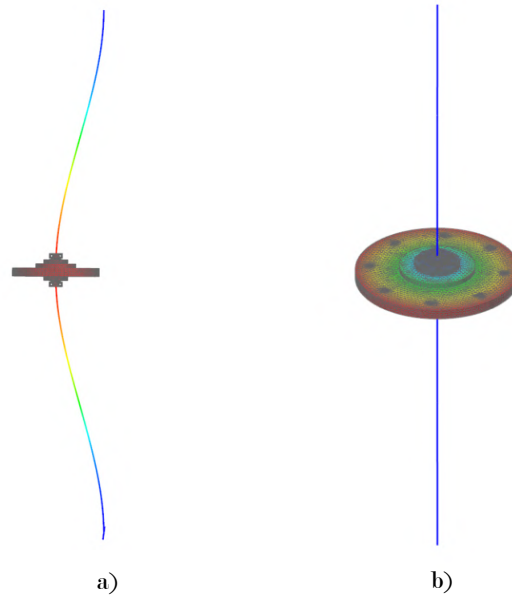


Figure 7.1 Results of the modal analysis. (a) First mode shape at 7.5 Hz. (b) Second mode shape at 12.1 Hz.

the misaligned case. However, this decline in displacement is shadowed by the lateral acceleration taken just below the upper bearing. Fig.(7.3) shows the difference in the acceleration resulting at the node preceding the upper bearing. This substantial difference in both cases is due to the fact that the misalignment causes a concentration of stress in the part of the shaft above the rotor as shown in Fig.(7.4). Finally, bearing loads presented in Fig.7.5 show an exponential evolution similar to the experiments with the reaction forces of the misaligned case being visibly inferior to the aligned one. This at least confirms that the responses from the experimental misaligned configuration could indeed help diagnose misalignment as the general behavior is reproduced in a primary FEM such as this one. It also illustrates that a relatively small misalignment can still be observed at the same time as shaft and bearing defects.

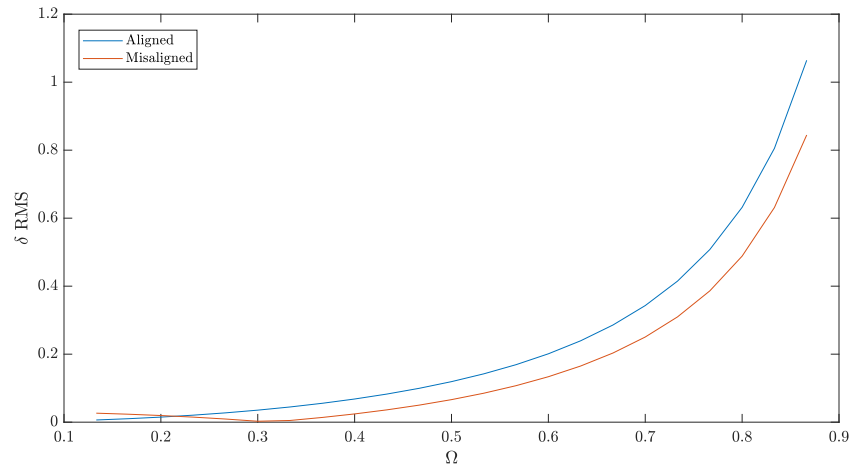


Figure 7.2 RMS amplitude of the shaft displacement at 60 mm above the lower bearing simulated over multiple experiments from 60 RPM to 390 RPM for both aligned and misaligned cases.

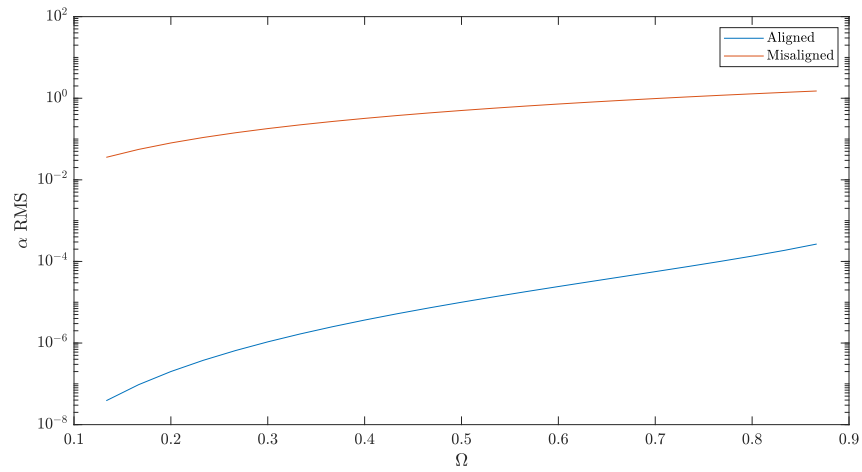


Figure 7.3 RMS amplitude of the shaft acceleration just below the upper bearing simulated over multiple experiments from 60 RPM to 390 RPM for both aligned and misaligned cases.

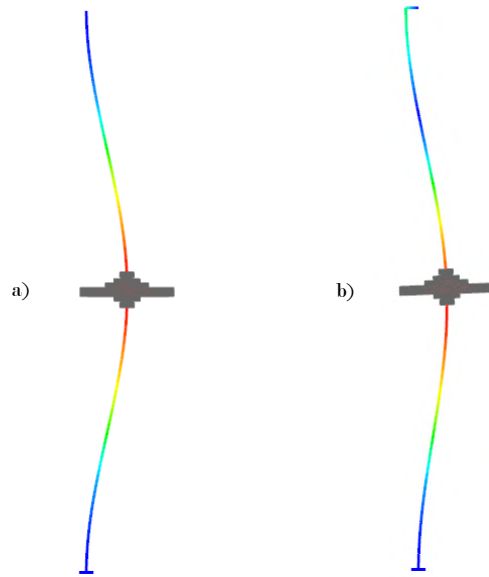


Figure 7.4 Normalized stress location from simulations at 330 RPM (5.5 Hz) showing concentrated stress close to the upper bearing in the misaligned case. (a) Aligned case. (b) Misaligned case.

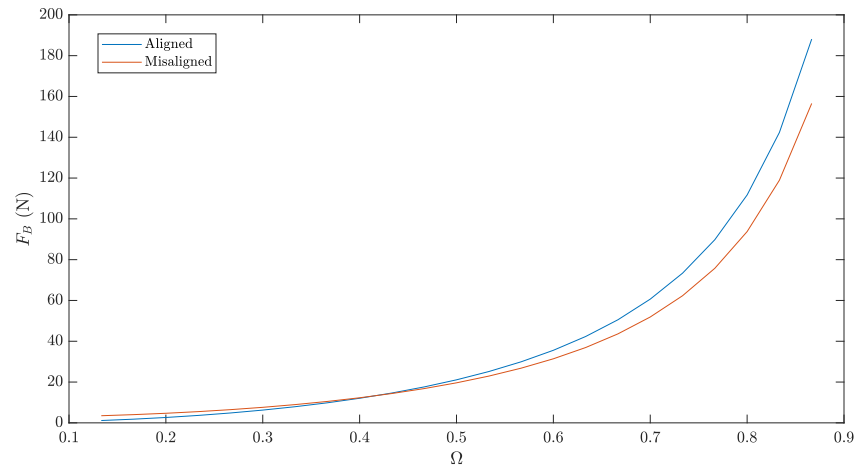


Figure 7.5 Bearings reaction forces simulated over multiple experiments from 60 RPM to 390 RPM for both aligned and misaligned cases.

CHAPTER 8 CONCLUSION

8.1 Summary of work

The primary focus of this thesis was to obtain a source of data for the future training of DT models. This was accomplished by the design of a Vertical Axis Rotating Machine (VARM) and the development of its plain finite element model. Particular attention and considerations were given to the VARM modularity and flexibility. By being able to simulate multiple cases of failures in rotor dynamics, it would provide a rich data base for future research.

The methodology employed involved several steps. First, a thorough literature review was conducted to understand the existing test-benches and models in rotor dynamics. The Jeffcott rotor model was identified as a practical model for understanding basic rotor dynamics concepts and providing initial estimations about the VARM characteristics like its expected first natural frequency. However, to produce a more accurate numerical data source, Finite Element Analysis (FEA) was recognized as an essential tool for simulating the behavior of rotating machinery under various conditions and is widely used to model rotating machinery.

In the design phase, material selection and final design considerations were made for the VARM. Particular attention was paid to the bearings, as they play a crucial role in the behavior of rotating machinery. Monitoring systems were also fitted to record data during operation. This included the use of laser micrometers for taking displacement measurements of the shaft line, an accelerometer set on the upper bearing housing and a high-speed camera to capture the shape deflection at any point to serve as a validation data source for future work on DT models. The frame was studied through modal analysis to make it stiffer and reduce the risks of resonance between it and the rotating components.

Misalignment and unbalance were identified as a significant and easy to reproduce failure modes. These failures can lead to increased vibrations and stresses in the rotor-bearing system and accelerated fatigue wear on equipment. We presented a few case studies of these failure modes and the data collected while trying to explain the responses recorded. The shaft deformity and less-than-optimal bearing performance were identified with confidence as an important source of vibrations and non-linearities. The addition of an unbalanced steel-disk regulates the responses orbitals and steady-state behavior of the rotor. Actual first natural frequencies for all configurations are ambiguous not too far from the FEM modal results, pointing to inexact modeling of the boundary-conditions. Many resonant frequencies associated with the frame were still successfully identified with a high level on confidence.

Results from simulations of the FEM confirm some of the behaviors noted in the experiments such as a disparities between aligned and misaligned configurations but much work is still needed to improve the accuracy of the model in reproducing real-world VARM performance.

In the end, the machine can be easily modified and upgraded, requiring only Allen-type keys to loosen or tighten components. The VARM can be operated by both students and technicians. The only training required involves safety checks before start-up and a basic understanding of the LabView programs needed to operate the motor and record data.

8.1.1 Limitations

Due to its design flexibility and low loads, the VARM displays unexpected dynamics stemming from part imperfections, such as a bent shaft and a suboptimal bearing locking mechanism. On one hand, this is crucial for the VARM since it will operate with various shafts and bearings. Thus, the disassembly and installation of these components should be straightforward and practical. On the other hand, this convenience introduces imperfections in materials, contacts, and boundary conditions, all of which influence the VARM's responses. Additionally, the system's lack of damping prevents the VARM from surpassing the first natural frequency, even at maximum acceleration. The frame's natural frequencies might be too low to disregard entirely, especially if the VARM operates at higher speeds.

The monitoring system could also benefit from upgrades as the lasers cannot be installed at a height superior of 370 mm above the lower bearing because of the frame structural parts. The high-speed camera image treatment procedure is not yet able to process images containing the rotor and the camera resolution is not high enough to detect small deflections close to the bearings. The data treatment process needs refining to improve the resolution of the waterfall plots from the ramping speed tests in sections 6.1.2 and 6.2.2.

Finally, the FEM is missing crucial parameters to produce results closer to the VARM. A more suitable bearing model is needed in the simulation and shaft defects are, for the time being, not reproduced in the FEM which causes important divergences with the experimental data. Without these, the simulations results are painfully elementary. A more representative model of the structure could also benefit the rotor model by simulating the effects of coupling between the two.

8.1.2 Future work

Better coupling and supports for the shaft should be investigated and attempted to have better control over its dynamics. For example, the use of a belt-pulley mechanism for torque

transfer between the motor and the shaft could significantly reduce the loads on the motor and improve its performance. A second-stage bearing between the jaw-coupling and the lower bearing would provide more stiffness to the shaft's lower-end thus reducing its lateral displacement and allowing the motor to achieve higher speeds. This could also be achieved by decreasing the distance between the jaw-coupling and the shaft protruding from the lower bearing.

Many rotor options were designed but not tested as we have yet to find a practical method to perform dynamic balancing of the VARM using the sensors currently on board. This would increase our control over the experiments as the residual unbalance is currently unknown as well as bearings parameters. The addition of damping to the system would prove profitable for conducting experiments closer to the first natural frequency. This could be done by adding rubber bushings in the bearings or by installing multiple rotors on the shaft as to create significant damping from gyroscopic effect. This data would allow for the identification of many of the system's parameters and would help determine the level of non-linearity being displayed. Pre-loaded bearings could also be tested to eliminate the effects of improperly loaded bearings.

As for the measurements, more methods and quantities could be monitored, such as noise and input motor current, through MCSA as discussed previously in section 2.2.2. These measurements could provide more insights into the machine's characteristics and help devise markers for different types of failure detection.

Finally, the many unknowns of the VARM and its FEM could serve for the development and implementation of practical system identification techniques which could detect and identify failures in real commercial assets. Methods like the Kalman-Filter are already being tried on similar systems and are always in need of experimental data to progress and mature. In order to facilitate the identification of the bearings coefficients, a compliant bearing housing could be made in order to decrease the supports stiffness of the shaft and thus make its influence on the system's responses more pronounced. More tests are still needed in order to identify adequate components that would maintain the VARM modularity while producing responses that do not exhibit such significant unexpected behaviors. We do not expect to completely get rid of such imperfections we want a digital-twin that will be able to account for such imperfections and still produce consequential results.

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APPENDIX A POTENTIAL SHAFT MATERIALS

The following page presents an excel sheet used to compute initial estimates of the VARM's dynamics and help decide for a suitable shaft material that would be able to handle high loads while still remaining in its elastic domain. Estimates of the output signal of a potential strain gages setup were also computed but this measurement option was later abandoned.

Diameter (mm)	6.35	Quadratic moment (mm4)	79.81	79.81			
Inner Diameter (mm)	0						
Length (mm)	1000	Strain gauge					
density kg/m3	8890	GF	2				
Unbalance mass (kg)	0.05	Resistance (Ohm)	350				
Distance (mm)	40	Vs (V)	5				
Unbalance (kg mm)	2						
RPM		rad/s					
Angular velocity	500	52.4					
Total radial force (N)	5.5						
			Tension (Mpa)	Rod	Tube		
Material	1045 Carbon Steel	Young Modulus (Mpa)	200000	109.06	109.06		
https://www.azom.com/article.aspx?ArticleID=9153		Strain (%)		0.05453	0.05453		
https://www.mcmaster.com/rods/material~1045-carbo		Yield Strength (Mpa)	310	0.3817	0.3817		
		Resistance vive (Re^2/2E)	0.24	1.363	1.363		
			Vo (mV)				
Material	1566 Carbon steel	Young Modulus (Mpa)	210000	0.05193	0.05193		
https://www.efunda.com/Materials/alloys/carbon_stee		Strain (%)		0.3635	0.3635		
https://www.mcmaster.com/shafts/material~1566-carb		Yield Strength (Mpa)	517	1.298	1.298		
		Resistance vive (Re^2/2E)	0.64				
			Vo (mV)				
Material	9310 Steel	Young Modulus (Mpa)	200000	0.05453	0.05453		
http://asm.matweb.com/search/SpecificMaterial.asp?b		Strain (%)		0.3817	0.3817		
https://www.mcmaster.com/9310-steel/material~9310		Yield Strength (Mpa)	1000	1.363	1.363		
		Resistance vive (Re^2/2E)	2.50				
			Vo (mV)				
Material	Phosphorous Bronze 510	Young Modulus (Mpa)	110000	0.09915	0.09915		
https://www.mcmaster.com/510-bronze/material~510-		Strain (%)		0.6940	0.6940		
https://morganbronze.com/wp-content/uploads/c510		Yield Strength (Mpa)	380	2.479	2.479		
		Resistance vive (Re^2/2E)	0.66				
			Vo (mV)				
Material	Phosphorous Bronze 544	Young Modulus (Mpa)	103000	0.1059	0.10589		
https://morganbronze.com/product/c54400-phosphor-		Strain (%)		0.7412	0.7412		
https://www.mcmaster.com/544-bronze/material~544-		Yield Strength (Mpa)	380	2.647	2.647		
		Resistance vive (Re^2/2E)	0.70				
			Vo (mV)				
Material	630 Nickel Aluminum Bronze	Young Modulus (Mpa)	121000	0.0901	0.09013		
https://morganbronze.com/wp-content/uploads/c630		Strain (%)		0.6309	0.6309		
https://www.mcmaster.com/630-bronze/material~630-		Yield Strength (Mpa)	450	2.253	2.253		
		Resistance vive (Re^2/2E)	0.84				
			Vo (mV)				

APPENDIX B BEARINGS SPECIFICATIONS

Mechanical specifications of the bearings installed on the VARM for the experimental data presented in this thesis.



YET 204



Insert bearing with an eccentric locking collar and narrow inner ring

Insert bearings are based on sealed deep groove ball bearings. This variant is intended for use in applications where the direction of rotation is constant. It has a narrow inner ring extended on one side, with an eccentric locking collar, enabling quick and easy mounting onto the shaft.

- Designed for rotation in one direction
- Quick and easy to mount onto the shaft
- Long service life
- Reduce noise and vibration levels

Overview

Dimensions

Bore diameter	20 mm
Outside diameter	47 mm
Width, total	30.5 mm
Width, inner ring	21 mm
Width, outer ring	14 mm

Performance

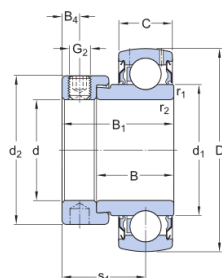
Basic dynamic load rating	12.7 kN
Basic static load rating	6.55 kN
Limiting speed	8 500 r/min
Note	Limiting speed with shaft tolerance h6

Properties

Retaining feature, inner ring	Eccentric collar
Bore type	Cylindrical
Rolling elements	Balls
Outer ring type	Spherical
Inner ring extension	On one side
Cage	Non-metallic
Rubber seating ring	Without
Material, bearing	Bearing steel
Coating	Without
Sealing	Seal on both sides
Sealing type	Contact, standard
Lubricant	Grease
Relubrication feature	With



Technical Specification



Dimensions

d	20 mm	Bore diameter
D	47 mm	Outside diameter
B ₁	30.5 mm	Overall bearing width
B	21 mm	Width of inner ring
C	14 mm	Width of outer ring
d ₁	≈ 28.2 mm	Outside diameter of inner ring
d ₂	32.4 mm	Outside diameter of locking collar
B ₄	4.75 mm	Distance from side face to thread centre
r _{1,2}	min. 0.6 mm	Chamfer dimension of inner ring
s ₁	23.5 mm	Distance from locking device side face to raceway centre

Calculation data

Basic dynamic load rating	C	12.7 kN
Basic static load rating	C ₀	6.55 kN
Fatigue load limit	P _u	0.28 kN
Limiting speed		8 500 r/min
		Limiting speed with shaft tolerance h6
Calculation factor	f ₀	13

Mass

Mass	0.16 kg
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Mounting information

Set screw	G ₂	M6x0.75
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Hexagonal key size for set screw	N	3 mm
Recommended tightening torque for set screw		4 N·m

Associated products

Rubber seating ring		RIS 204
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APPENDIX C FRAME RIGIDITY STUDY

In the process of stiffening the VARM's frame to mitigate coupling between its natural modes and the vibrations generated by the rotor we first considered replacing the original aluminum frame with a completely new design. This model would have been made of steel beams and a few aluminum extrusion rails to allow the movement of some important components such as bearings housing. It is taller and much heavier than the first frame at 1.6 m high and 110 kg of mass and could have allowed for a maximum effective shaft length of 990 mm. It is equipped with a security halo above the upper bearing to contain the deflections of the protruding shaft above the upper bearing when such a rotor configuration is studied. Fig.(C.1) shows in (a) the original aluminum design and in (b) the steel frame mentioned. Initial modal analysis showed great promise for the steel frame with the first 3 natural frequencies at 142 Hz, 184 Hz and 228 Hz versus 30 Hz, 42 Hz and 46 Hz for the original frame. Even though these values would be lower in real life due to approximations made in the model as discussed in section 4.2.2, we could still expect an increase of 3x to 5x in the real-life resonant frequencies. However, the important amount of time needed to take this design from CAD to reality was expected at 2 months at least. It was then decided to stiffen the already-existing frame as much as possible and keep experimenting on it. The FEM parameters used in the analysis are similar to those described in section 4.2.2 with 3 308 965 total CTETRA(4) elements.

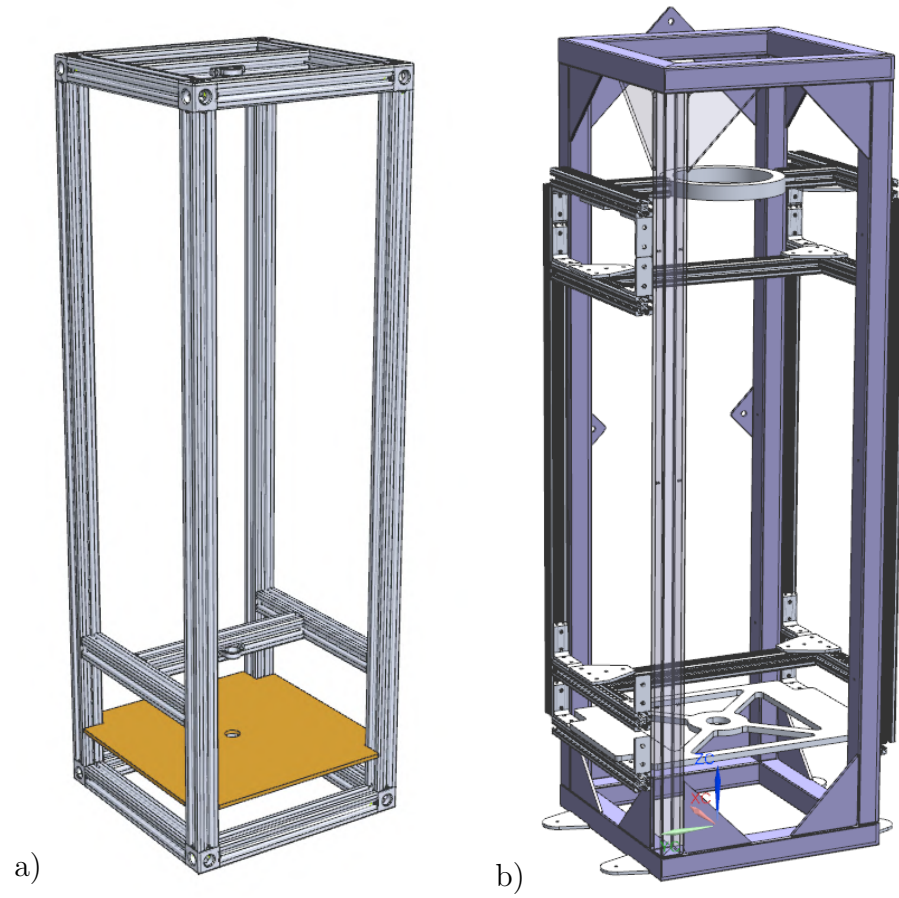


Figure C.1 CAD models of the VARM support frame: (a) original aluminum design before stiffening; (b) steel-aluminum design.