

Design of an Ultra-High Cycle Fatigue Setup for Thin Planar Specimens of Aluminium Alloys

by

Tomás Moutinho de Azevedo Brito

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Supervisor

Prof. Dr. Abílio Manuel Pinho de Jesus

Co- Supervisor

PhD. Felipe Fiorentin

ABSTRACT

This master's dissertation delves into the domain of ultra-high cycle fatigue (UHCF) behaviour in the AlSi10MnMg alloy, widely known as Silafont-36, with a particular emphasis on flat specimens. Understanding UHCF is critical for a variety of applications, ranging from wind turbines to automotive components, as materials undergo an increasing number of cycles over their operating lifetime.

The dissertation begins with a thorough literature review that covers the principles of fatigue, stress-based approaches, and the evolution to UHCF regimes. The study capitalises on ultrasonic testing, emphasising its benefits, such as shorter test times and costs and increased testing efficiency, and considerations for testing equipment and frequencies. A mechanical wave transmission model serves as the foundation for succeeding chapters.

The research focuses on flat specimen design and the challenges of gigacycle fatigue testing. A combination of analytical and finite element approaches is used to design specimens for testing on the Shimadzu USF-2000 machine, with particular emphasis on flat geometries. Finite element analysis emerges as an important method for gaining insights into resonance frequencies and stress concentrations. The chapter describes precise methods, including part sketches, measurements, and assembly design, that are required to achieve appropriate stress ranges inside flat specimens.

A crucial optimisation phase is the reduction of gripper stiffness, investigated through material selection and geometric changes. The simulation and assembly of the gigacycle fatigue machine are carried out with precision, assuring operational frequency alignment. Material properties, mesh parameters, and force analyses help to provide a full understanding of the experimental setup.

Moving on to the experimental programme, the dissertation describes the ultrasonic testing equipment as well as the fatigue experimental project.

The following chapter examines the outcomes from the experimental and numerical analyses of two distinct specimen designs. It presents an analysis of their performance under ultra-high cycle fatigue testing, highlighting the discrepancies between simulated predictions and experimental observations and the critical role of design refinements in achieving the desired stress distributions and resonance frequencies.

Keywords: Fatigue, Gigacycle Fatigue, UHCF, VHCF, Ultrasonic Fatigue Testing, Flat Specimens, Silafont-36, 20 kHz, Resonance.

RESUMO

Esta dissertação de mestrado explora o comportamento de fadiga em ciclos ultraelevados (UHCF) na liga AlSi10MnMg, amplamente conhecida como Silafont-36, com ênfase particular em provetes planos. A compreensão da UHCF é crucial para uma variedade de aplicações, desde turbinas eólicas até componentes automóveis, uma vez que os materiais são submetidos a um número crescente de ciclos ao longo da sua vida útil operacional.

A dissertação inicia-se com uma revisão abrangente da literatura sobre os princípios de fadiga, abordagens baseadas em tensão e a evolução para UHCF. O estudo potencializa testes ultrassônicos, enfatizando os seus benefícios, como tempos de teste e custos inferiores e eficiência de teste aumentada, e considerações para equipamentos e frequências de teste. Um modelo de transmissão de ondas mecânicas serve de base aos capítulos seguintes.

A investigação centra-se no design de provetes planos e nos desafios dos testes de fadiga em gigaciclos. Uma combinação de abordagens analíticas e de elementos finitos é utilizada para projetar provetes para testes na máquina Shimadzu USF-2000, com ênfase particular em geometrias planas. A análise de elementos finitos emerge como um método importante para obter conhecimentos subtis sobre as frequências de ressonância e concentrações de tensão. O capítulo descreve métodos precisos, incluindo esboços de peças, medições e design de montagem, necessários para atingir gamas de tensão apropriadas em provetes planos.

Uma fase crucial de otimização é a redução da rigidez da amarra, investigada por meio da seleção de materiais e alterações geométricas. A simulação e a montagem da máquina de fadiga em gigaciclos são realizadas com precisão, assegurando o alinhamento com as frequências operacionais. As propriedades dos materiais, os parâmetros de malha e as análises de força contribuem coletivamente para uma compreensão abrangente do cenário experimental.

O capítulo seguinte examina os resultados das análises experimentais e numéricas de dois designs distintos de provetes planos. Apresenta uma análise do seu desempenho em ensaios de fadiga de ciclo ultra-elevado, destacando as discrepâncias entre as previsões simuladas e as observações experimentais e o papel crítico dos aperfeiçoamentos dos designs para alcançar as distribuições de tensões e frequências de ressonância desejadas.

Palavras-chave: Fadiga, Fadiga em Gigaciclos, UHCF, VHCF, Teste de Fadiga Ultrassónica, Provetes Planos, Silafont-36, 20 kHz, Ressonância.

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LIST OF SYMBOLS

Symbol	Description	Unity
∇^2	Laplacian/Laplace operator	-
A_0	Displacement amplitude	m
A_{n-1}	Wake constant	-
A_r	Amplitude ratio	-
B_{n-1}	Wake constant	-
E_d	Dynamic elastic modulus	Pa
$F_{bending}$	Bending force	N
F_{bolt}	Preloaded bolt force	N
$F_{friction}$	Friction force	N
F_{spare}	Spare force	N
U_{sim}	Simulation's displacement	m
$\dot{\epsilon}$	Strain rate	1/s
σ_a	Stress amplitude	Pa
σ_m	Mean stress	Pa
σ_{max}	Maximum stress	Pa
σ_{min}	Minimum stress	Pa
σ_{sim}	Simulation's stress	Pa
σ'_w	Estimated fatigue limit	Pa
ω_n	Natural angular frequency	rad/s
Δa	Crack development rates	m
ΔK	Stress intensity factor range	Pa·m ^{1/2}
$\Delta\sigma$	Stress range	Pa
A	Area	m ²
c	Velocity of mechanical wave propagation	m/s
E	Elastic modulus (Young)	Pa
e	Volume dilatation	-
f	Frequency	Hz
F_N	Normal	N
K	Stiffness	N/m
k	Wave number	-
L	Specimen's length	m
M	Mass	kg
N	Number of cycles	-
N_f	Fatigue life	-
R	Stress ratio	-
$S-N$	Fatigue Strength curve	-
S_f	Fatigue limit	Pa
S_N	Fatigue strength at N cycles	Pa
T	Wave period	s
t	Time	s
u	Displacement along x-direction	m
U	Displacement	m
v	Displacement along y-direction	m
w	Displacement along z-direction	m
W	Specimen's width	m
x	Cartesian coordinate	m
y	Cartesian coordinate	m

z	Cartesian coordinate	m
ε	Strain	m/m
μ	Friction coefficient	-
ν	Poisson's ratio	-
ρ	Material density	kg/m ³
σ	Stress	Pa
ω	Angular frequency	rad/s

LIST OF ACRONYMS

Acronym	Description
3D	Three-dimensional
ASTM	American Society for Testing and Materials
DAQ	Data Acquisition System
DOF	Degree of Freedom
EDM	Electrical discharge machining
ERD	Electromagnetic Resonance Device
FEM	Finite Element Method
FEUP	Faculty of Engineering of the University of Porto
FFT	Fast Fourier Transform
HCF	High Cycle Fatigue
HPDC	high-pressure die casting
HV	Vickers hardness
INEGI	Institute of Science and Innovation in Mechanical and Industrial Engineering
LCF	Low Cycle Fatigue
ODA	Optical Dark Areas
UFT	Ultrasonic Fatigue Testing
UHCF	Ultra-High Cycle Fatigue
VHCF	Very High Cycle Fatigue

1. INTRODUCTION

1.1. Motivation

Understanding the changing nature of engineering materials and the increasing demand to understand their fatigue behaviour is crucial, especially when it comes to Very High Cycle Fatigue (VHCF). Components are exposed to extended operating cycles with comparatively small stress amplitudes in more complex and advanced engineering applications, which is testing the limits of conventional fatigue research.

In recent years, the study of VHCF has attracted significant attention due to the realisation that many critical components, especially in industries such as transportation and energy, experience stress cycles exceeding 10^9 cycles. This continuous cyclic loading exposure can cause fatigue damage, raising questions about structural integrity and even posing dangers to safety (Bathias, 2014).

Studies about fatigue usually focus on the High Cycle fatigue (HCF) regime, which typically involves up to 10^7 cycles. But as engineering applications advance and call for lighter, more effective materials, the importance of VHCF becomes more and more clear. In contrast to HCF, where surface-initiated cracks are more prevalent, VHCF presents a unique issue in which cracks may start inside the material's microstructure. The important role of material microstructure in controlling fatigue behaviour in the super-cyclic regime is highlighted by this phenomenon (Song, Zhu, Lu, & Lu, 2018).

Within this framework, the objective of the dissertation is to further explore the fatigue behaviour of metallic alloys, especially aluminium, under VHCF circumstances. Due to its lightweight nature, aluminium is preferred in many technical applications; however, more research is needed to determine how aluminium performs under extended cyclic loads in the VHCF regime (Mirzadeh & Razaghian, 2020).

It is also imperative to take into account, not only the critical functions of material selection, but also the production methods. In particular, high-pressure die casting (HPDC) techniques have grown becoming increasingly common in modern engineering processes (Altan & Saan, 2008).

HPDC stands out as an essential manufacturing process famous for its ability to produce intricate and thin-walled components with exceptional precision and efficiency. Using a steel cast cavity, molten metal is injected at high pressures (usually between 10,000 and 30,000 psi, or 69 and 207 MPa) (Makhlouf & Ali, 2017) to create complex structures with precise dimensional tolerances and rapid solidification.

HPDC is an excellent option for developing and producing delicate specimens because of its many benefits. First of all, it makes it possible to produce complex forms and thin-walled structures that would be difficult or impossible to produce using traditional manufacturing techniques (Makhlouf & Ali, 2017). Secondly, components with tight dimensional tolerances and little variation are produced by the high pressure used during the casting process, which ensures precise replication of the mould shape (Pawar & Choudhary, 2014). Additionally, HPDC is a very effective method that can quickly produce large quantities of components, cutting lead times and production costs (Makhlouf & Ali, 2017).

HPDC can be used to process an extensive range of metals and alloys, such as zinc, magnesium, and aluminium, providing flexibility in material selection to satisfy particular performance requirements (Makhlouf & Ali, 2017). Additionally, HPDC produces components with great surface finish, which minimises the need for further finishing processes and improves the overall quality of the final product (Pawar & Choudhary, 2014).

Given the slender nature of the components, it becomes imperative to fabricate test specimens with thin walls that are representative of the material properties while ensuring their dimensional accuracy. In this context, HPDC becomes a vital facilitator, providing the ability to create complex and accurate specimens with sophisticated characteristics. These specimens are crucial tools for material characterisation because they are extracted from the cast components. However, the need to subject them to VHCF testing, which demands prolonged testing durations, poses significant challenges. The development of innovative testing techniques created especially for these specimens is required to address this. In order to help with comprehensive research and testing across various domains, these procedures must be carefully developed to ensure the accurate representation of the material's behaviour under long-term cyclic loading conditions.

1.2. Objectives

1.2.1. General objectives

The main objective of this study is to thoroughly analyse the VHCF behaviour shown by flat specimens of the aluminium alloys. With a strong emphasis on flat geometries, the study seeks to uncover the deep complexity of VHCF in this specific arrangement. The research aims to provide a comprehensive understanding of how flat specimens of aluminium alloy respond to lengthy and recurrent stress cycles by

investigating their durability and performance under continuous cyclic loading. This targeted research is critical for applications where materials are subjected to severe cyclic loads over their operational lives, providing valuable insights into the realm of VHCF in flat geometries of the specified aluminium alloy.

1.2.2. Specific objectives

In turn, the specific objectives of this dissertation are summarized as follows:

a) perform a comprehensive review of the status of current fatigue research, paying special attention to new developments and innovative methods, by developing and implementing thin-plate specimens for fatigue testing at ultrasonic frequencies and with tensile uniaxial loading;

b) examine and evaluate the fundamental principles of fatigue, including stress-based techniques and recent advances in VHCF regimes, giving particular focus to how these concepts connect to the proposed novel fatigue design that involves the use of flat plate specimens;

c) review the concepts and characteristics of the testing apparatus, emphasising on the Shimadzu USF-2000 in specific, to assess its suitability and effectiveness for carrying out fatigue tests on proposed flat plate specimens under ultrasonic frequencies and tensile uniaxial loading conditions;

d) develop innovative procedures for testing that are specific to the proposed flat plate specimens, the objective of which is to provide accurate and reliable fatigue testing under tensile uniaxial loading conditions and ultrasonic frequencies, improving the understanding of the specimens' fatigue behaviour and facilitating their valuable application in engineering applications.

1.3. Document structure

This dissertation takes a structured approach to examine VHCF behaviour in metallic alloys. After this introductory chapter, the document progresses to thoroughly examine research literature in the area, specimen design, experimental protocols, outcomes, and definitive findings, as described below.

Chapter 2, *Literature Review*, provides a comprehensive exploration of fatigue fundamentals, stress-based methodologies, VHCF regimes, and the mechanical wave transmission model. It covers the theoretical underpinnings essential to fully understand how materials behave when subjected to ultrasonic fatigue.

Chapter 3, *VHCF testing details and specimen design*, presents a detailed analysis of the Shimadzu USF-2000 machine, which is the designated ultrasonic testing device, including sensors used to monitor testing parameters. It carries on to explain how the apparatus was adjusted to work with flat specimens, guaranteeing reliable and precise testing conditions. The rigorous design process of the specimen, grippers, bolts, and nuts is covered in detail in the following section, which describes every step from conception through final assembly. This includes the reasons behind the design decisions, factors to take into account when choosing materials, and the iterative design process used to optimise the components for VHCF testing. The third part of this chapter delves into the important aspects of the finite elements simulations used.

Chapter 4, *Results and Discussion*, is structured into two distinct sections, delineated by the examination of two sequential designs of thin-plate aluminium specimens: Design A and Design B. As a refinement on Design A, Design B was created to overcome limitations found when the original design was tested experimentally. Each section dedicated to a particular design is split into three additional sections. First, an extensive review of the intricate details of each component's design is provided, giving in-depth understanding of the assembly's components. Following this, the second section examines the numerical results of the simulations and the characteristics and performance of the corresponding designs. Last but not least, the third section carefully analyses the experimental data, exposing any difficulties or inconsistencies noticed during testing and providing a thorough evaluation of the experimental results.

Chapter 5, *Conclusions and Future Work*, summarises major findings and conclusions and identifies prospective areas for future investigation.

Each succeeding chapter contributes to the overall goal of improving the understanding of VHCF behaviour in specific alloys, with a focus on flat specimens.

2. LITERATURE REVIEW

The current chapter will provide the theoretical framework required for the development of this study. A review of the literature on fatigue principles will be conducted. VHCF will receive more attention, in particular flat specimens. There will also be a strong emphasis on aluminium alloys.

2.1. Fundamentals of fatigue

Fatigue is an inevitable phenomenon that occurs in materials subjected to cyclic loading, particularly in structural components exposed to various stresses. Its importance derives from the possibility of catastrophic failure, as well as the related economic consequences and threats to human life. According to studies, fatigue causes around 90% of metallic failures (Costa et al., 2020).

Fatigue is defined as the progressive structural alteration that occurs in a material under fluctuating stresses, potentially culminating in cracks or complete fracture after a sufficient number of load cycles, according to the American Society for Testing and Materials (ASTM) Standard E-1823 (ASTM International, 2013).

The ASTM defines fatigue strength (S_N) as the stress that causes failure after n cycles, and fatigue limit (S_f) as the median fatigue strength as the fatigue life (N_f) increases significantly (ASTM International, 2013). While not specifically stated, the ASTM suggests that the endurance limit corresponds to the fatigue limit (Bathias, 2014).

August Wöhler's pioneering work in 1870 gave rise to the S-N curve, sometimes referred to as the Wöhler curve. It displays the correlation between the stress range and the number of failure cycles. S-N curves, which will be discussed in the following section, can be established using modern technology for specific components or structures subjected to diverse loading conditions (Liu, 2019).

2.1.1. Stress-based approach

The stress-based approach is used in fatigue life modelling for components operating below the yield stress. This approach links a component's endurance to stress magnitudes, indicating that higher stress levels lead to shorter lifespans. In Figure 2-1, the terminology of alternating stress is illustrated (ASTM International, 2013).

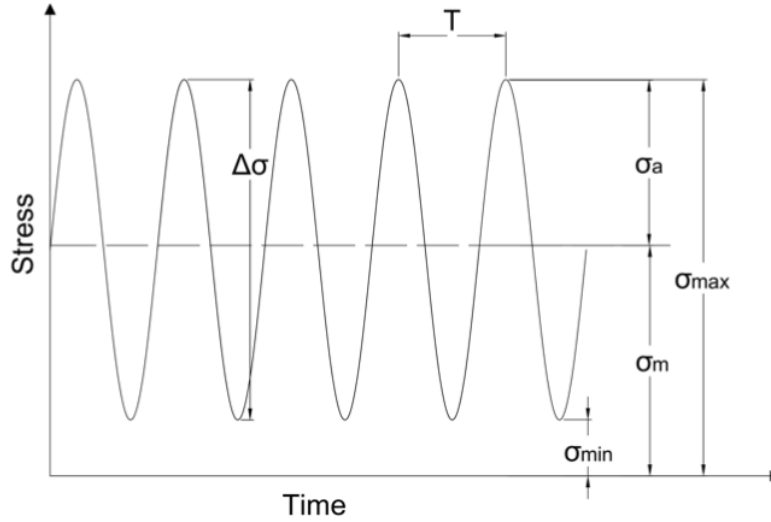


Figure 2-1 – Terminology for alternating stress (Gouveia, 2022).

Mean stress (σ_m) is calculated as the mean between maximum and minimum stress,

$$\sigma_m = \frac{\sigma_{max} + \sigma_{min}}{2}, \quad (2-1)$$

stress range ($\Delta\sigma$) is the difference between maximum and minimum stress,

$$\Delta\sigma = \sigma_{max} - \sigma_{min}, \quad (2-2)$$

stress amplitude (σ_a) is represented by half of the stress range,

$$\sigma_a = \frac{\Delta\sigma}{2} = \frac{\sigma_{max} - \sigma_{min}}{2}, \quad (2-3)$$

stress ratio (R) is defined by the ratio between the minimum and the maximum stress and its values are smaller than 1,

$$R = \frac{\sigma_{min}}{\sigma_{max}} \quad (2-4)$$

The stress ratio may assume infinite values, but there are two distinct values that are the most important: when it equals zero, the region under study is never subjected to compressive loads; this is known as pulsated loading, also called unidirectional stress. The minimal stress in this scenario is zero. The loading condition is entirely reversed when the R equals -1, meaning that the signals for the highest and minimum stresses have the same values in modulus but are opposite (Fiorentin, 2023; Gouveia, 2022)

Period (T) and frequency (f) are the time needed to complete one cycle or its inverse, respectively.

In a similar vein to the existence of R , it is pertinent to note the presence of amplitude ratio (A_r).

$$A_r = \frac{\sigma_a}{\sigma_m} = \frac{1 - R}{1 + R} \quad (2-5)$$

2.1.1.1. S-N curves

When compared to strain-based approaches, one of the key advantages of the stress-based approach is its simplicity and ease of implementation. The fundamental goal of stress-life approaches, as the name implies, is to establish a link between the stress in a region and its usable life, also known as the number of cycles. For the time being, the state of stress in a specific location will be simply represented as "stress". This relationship is frequently described in a Stress-Number of Cycles curve, also called S-N curve (R. Budynas & Nisbett, 2008).

When it comes to S-N curves, there are two types of materials. The first one has a fatigue limit, which indicates that the material will not fail due to fatigue when the stress ranges are below this fatigue limit. The suggestion of infinite life will be exposed to analysis in the following section 2.2. This is common for iron and titanium alloys. In most cases, the fatigue limit occurs between 10^6 and 10^7 cycles. The second sort of material is one that does not have a fatigue limit. Even minor stress ranges can cause fatigue failure in these materials, if a sufficient number of cycles is applied. This behaviour has been observed in aluminium, magnesium, and copper alloys. These differences can be observed in Figure 2-2 (Callister & Rethwisch, 2018).

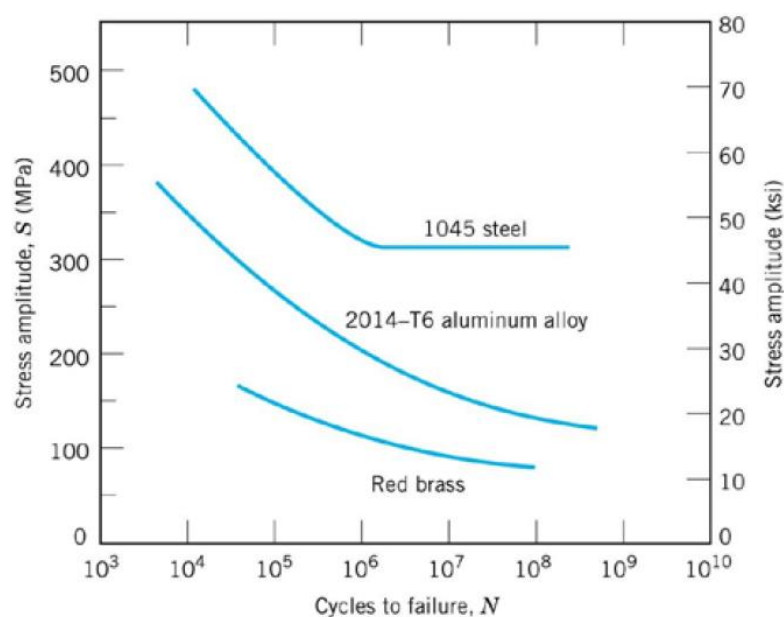


Figure 2-2 – S-N curve for materials with different behaviours (Chegg Inc., 2015).

The debate over fatigue limitations in materials, which was particularly significant in the 1980s and 1990s, was primarily driven by testing constraints that limited the number of cycles explored. Traditional testing procedures frequently reached a maximum of 10^6 or 10^7 cycles, raising concerns about the legitimacy of claimed fatigue limits. It is worth noting that traditional testing processes, often prolonged years, were not only time-consuming but also presented issues in thoroughly investigating the durability of materials. This temporal limitation required the use of novel methodologies, such as ultrasonic fatigue testing (UFT), which revealed failures that occurred much beyond these conventional cycle counts, prompting a reassessment of the established fatigue limits (Fiorentin, 2023).

2.1.2. Fatigue behaviour influencing factors

A complete assessment of a specific material's fatigue behaviour involves a systematic sequence of experiments. The fatigue life of a given specimen is dependent on several contributing factors, with paramount importance attributed to the important determinants described below (Fiorentin, 2023; Stephens, Fatemi, Stephens, & Fuchs, 2000; Vernon & Mackin, 2001).

Surface Finish: Poor surface finish, characterised by higher roughness, not only shortens the fatigue life of materials under equivalent loading conditions, but also acts as a powerful stress concentrator in these high-cycle fatigue regimes. It has a significant impact on High Cycle Fatigue (HCF) and, more specifically, Very High Cycle Fatigue (VHCF).

Specimen Size: In the domain of conventional materials, a broad classification that covers materials created through multiple manufacturing processes but excludes additive manufacturing, specimen dimensions appear as an important factor. Notably, greater specimen volumes in conventional materials improve the vulnerability to the development of flaws, particularly microstructural discontinuities at the surface. This increased susceptibility to flaws has a substantial impact on the material's fatigue performance, emphasising the importance of carefully selecting specimen size when evaluating material behaviour under cyclic loading conditions.

Loading Type: The complex nature of fatigue testing methodologies play an important part in generating the typical S-N curves, with different loading situations producing varied results. In the context of axial loading, specimens suffer significant volumes of material subjected to stresses near their maximum limits. This is in contrast to bending, in which sections close to the neutral axis suffer almost no stress while those farther away receive maximum stress. Notably, axial tests are subject to the effects of

misalignments and eccentricities, which can influence results. Bending specimens, on the other hand, show a stress gradient throughout their structure, with highest stress at the surface and zero stress at the neutral axis. This sensitive interplay of loading types emphasises the importance of carefully examining testing settings to assure the accuracy and reliability of fatigue data produced.

Residual Stress and Stress Ratio: Lower stress ratios and compressive residual stresses tend to improve fatigue performance.

Microstructure: Finer microstructures have better mechanical characteristics, which influences fatigue behaviour positively, but inclusions and pores have a negative impact.

Notch Sensitivity and Anisotropy: These characteristics determine stress concentration effects and material response based on load direction and structural orientation.

Environmental and Temperature Effects: Environmental factors such as temperature and humidity have a substantial impact on fatigue life. Corrosive conditions hasten crack nucleation and propagation, resulting in a shorter fatigue life.

Excitation Frequency: The impact of excitation frequency on fatigue testing is a significant factor to consider. Higher frequencies introduce complexity such as greater strain rates and the possibility of local plastic deformation, which may result in temperature increases during tests. This dynamic is amplified when combined with temperature effects and exposure to harsh conditions such as corrosion. The frequency of testing in corrosive settings also influences the material's exposure to atmospheric impacts. Understanding the relationship between excitation frequency, temperature, and ambient parameters is critical for a thorough fatigue assessment.

2.2. Very High Cycle Fatigue

Mechanical parts found in many different industries, such as auto and truck parts, railroad infrastructure, bridges, medical equipment, and high-pressure power plant components, are subjected to cyclic loads that frequently exceed 10^7 cycles. High frequencies or extended product life spans may be the cause of this intense cycling. The average number of cycles that various components experience over their lifetimes is shown in Figure 2-3 (Bathias, 2014).

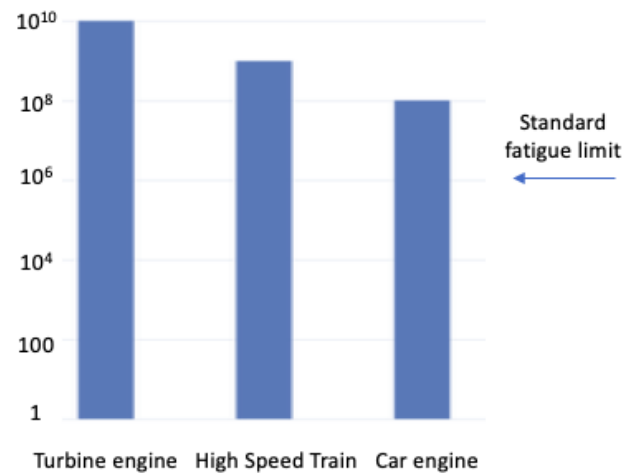


Figure 2-3 – Fatigue life of machines and components (Adapted from Bathias (2014)).

In Figure 2-4, the schematic S-N curve for a high-strength steel shows failures occurring beyond the range when materials experienced lower stress amplitudes.

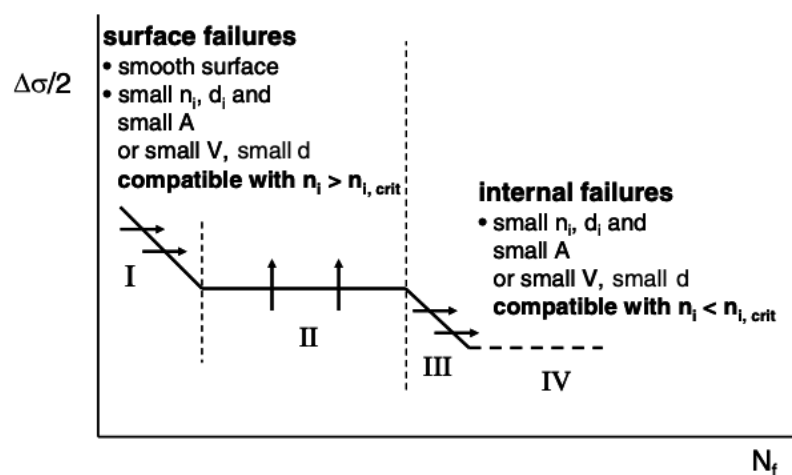


Figure 2-4 – Schematic S-N curve for a high-strength steel (Mughrabi, 2006).

Testing limits, notably the relatively lower testing frequencies of the time, could have influenced the observed fatigue limit. These constraints hampered fatigue tests that went beyond very high cycle counts, such as 10^8 or 10^9 cycles.

The number of cycles to failure is used to categorise fatigue failures (Figure 2-5):

- Low Cycle Fatigue (LCF): Failure happens in under 10^4 cycles.
- High Cycle Fatigue (HCF): Failure happens in cycles ranging from 10^4 to 10^7 .
- Very High Cycle Fatigue (VHCF): Failures happen in more than 10^7 cycles (Fatemi, 2016; Pyttel, Schwerdt, & Berger, 2011).

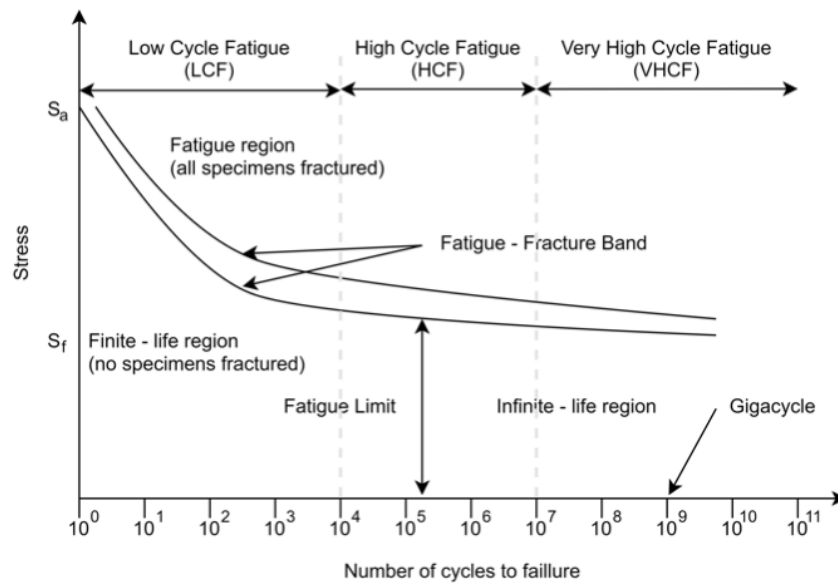


Figure 2-5 – S–N curves that typify fatigue test results when testing medium-strength steels, categorizing the cycles to failure (Gouveia, 2022).

By the close of the 20th century, materials that fatigued after a significant number of cycles were recognised, especially in Japan. Bathias (1999) emphasised that metallic materials do not have an unlimited fatigue life. Determining safe fatigue strength became dependent on an understanding of fatigue life beyond 10^7 cycles, up to 10^{10} cycles (VHCF regime). Traditional fatigue testing poses significant difficulties for cycles up to 10^{10} . The difference of duration of fatigue testing approaches for continuous loading can be seen in Table 2-1 (Bathias, 2014; Fiorentin, 2020).

Table 2-1 – Comparison between the duration of different fatigue testing methods for continuous loading (approximate values, rounded to the nearest 0.5) (Costa et al., 2020).

Number of Cycles	Ultrasonic (20 kHz)	Resonance (150 Hz)	Rotating Bending (30 Hz)
10^7	8.5 minutes	1 day	4 days
10^8	1.5 hours	1 week	1.5 months
10^9	0.5 days	2.5 months	1 year
10^{10}	1 week	2 years	1 decade

The development of ultrasonic fatigue testing methods addressed the shortcomings of traditional testing. Testing specimens to fatigue failure within the VHCF regime is made faster and more affordable with ultrasonic testing, which operates at frequencies between 15 and 30 kHz, with a typical frequency of 20 kHz (Bathias, 2014; Pacheco, 2016).

Historical innovations in the development of ultrasonic fatigue testing included: early in the 20th century, Hopkinson's Electromagnetic Resonance Device (ERD), which broke through previously restricted testing frequencies to enable ultrasonic fatigue testing; modern machinery, based on Mason's Piezoelectric Transducers in 1950, which made it possible to conduct ultrasonic fatigue testing at 20 kHz; and later developments, which brought about higher frequencies, such as 199 kHz, by Girard in 1959, and 92 kHz, by Kikukawa in 1965 (Bathias, 2014).

The most commonly used frequency is still 20 kHz, despite higher achievable frequencies, because of the possibility of high strain rates leading to sample overheating and changing the outcome (Bathias, 2014; Fiorentin, 2020).

Neppiras in 1959 suggested that ultrasonic fatigue testing be used to calculate S-N curves. Fatigue fracture propagation testing was first demonstrated by Mitsche in 1973, who provided ultrasonic data on crack development rates (Δa) compared to stress intensity range (ΔK) (Bathias, 2012, 2014; G. R. Budynas & Nisbett, 2011).

In order to conduct experiments with precise load sequences and identify the fatigue characteristics of materials under varying amplitude loadings, Bathias, Wu, and Ni in 1994 designed computer-controlled devices for ultrasonic fatigue testing.

Rapid generation of VHCF data using ultrasonic fatigue technologies finds use in a variety of industries, including aircraft, cars, trains, offshore, and construction, making it a practical technique to get VHCF data (Bathias, 2012, 2014).

VHCF research has been transformed by the development of ultrasonic fatigue testing, which has made it possible to understand materials' fatigue behaviour in greater detail than only 10^7 cycles. It is a useful instrument to guarantee the longevity and dependability of engineering components functioning in the VHCF regime, because of how quickly and easily it can gather VHCF data.

UFT has changed fatigue analysis in mechanical engineering, by providing an accelerated and precise method for assessing VHCF behaviours, with a particular emphasis on stress amplitudes greater than 10^7 cycles. This method has gained popularity due to its utility in inspecting essential components in a variety of industries, including motor parts, automotive components, and power plant elements (Bathias & Paris, 2004).

2.2.1. Reduced test time and costs

The key benefit of UFT is its capacity to significantly reduce testing duration and costs. UFT greatly reduces testing times when compared to conventional methods, operating at frequencies ranging from 15 kHz to 30 kHz, with a typical frequency of 20

kHz. There is a unique time and money-saving element that permits testing to take up to 400 times less time than with traditional methods: from days to minutes, months to hours, years to days, decades to weeks. For example, producing 10^7 cycles under UFT takes about 9 minutes as opposed to days or even years in typical settings. This can also be confirmed in Table 2-1 (Bathias & Paris, 2004; Stanzl-Tschegg, Mayer, & Stich, 2002).

2.2.2. Improved testing efficiency

The use of high-frequency techniques in UFT not only speeds up the determination of fatigue limits, but also allows for more thorough examinations of material behaviours under rigorous cyclic loading conditions. Increased testing frequency allows for finer resolution of material reactions to strong cyclic stress, resulting in a more nuanced understanding of fatigue performance. This increased sensitivity enables a more rigorous and precise investigation of the material's endurance and behaviour, resulting in a thorough characterisation of its fatigue properties (Stanzl-Tschegg et al., 2002).

2.2.3. Cooling and energy considerations

Efficient cooling systems are necessary to counteract heat generated during UFT, creating challenges in terms of potential corrosion but giving significant energy savings. Controlling the subsequent damping heat is especially important when loading with high frequencies. Loading with frequent interruptions (pulsed loading) is one method to avoid specimen heating. As a result, pulse lengths are typically between 25 and 100 ms (500-2000 cycles), while pauses can range between 25 and 1000 ms, depending on factors such as stress levels and material properties (including thermal conductivity, damping coefficient and specific heat). Liquids and compressed air can also be utilised for cooling; however, in the case of liquids, potential corrosion effects must be considered (Stanzl-Tschegg, 2014).

2.2.4. Frequency Effects in Ultrasonic Fatigue Testing

UFT variations in frequency have a significant influence on material behaviour, affecting crack initiation, propagation, and fatigue response. Such variations cause considerable alterations in fatigue behaviour, influencing crack initiation mechanisms in studied materials (Bathias, 2012).

Investigations across numerous materials highlight the nuanced changes in fatigue strength and life at different frequencies during UFT, providing useful insights into

material durability under varying frequency settings. Understanding these oscillations improves our perception of material reactions, which is critical to appropriately interpret fatigue events (Tsutsumi, Murakami, & Doquet, 2009).

Comparative analyses of UFT and conventional testing procedures reveal intriguing differences in fracture development mechanisms caused by frequency effects. These differences highlight the unique effects of frequency on crack formation and propagation, emphasising the importance of a thorough understanding of frequency-induced effects on material fatigue (Furuya, Matsuoka, Abe, & Yamaguchi, 2002).

This thorough examination of frequency-induced changes in fatigue behaviour, including crack initiation, mode-specific influences, and differences in crack growth between UFT and conventional methods, provides a comprehensive view of the intricate interplay of frequencies in determining material response and endurance under fatigue conditions.

Furuya et al. (Furuya et al., 2002) investigated how frequency affects fatigue properties in JIS SNCM439, a brittle, high-strength low-alloy steel prone to inner crack nucleation and fish-eye formation. This phenomenon in fatigue refers to the formation of interior cracks, which is particularly common in brittle, high-strength materials such as JIS SNCM439. This unusual crack initiation originates within the material, frequently from non-metallic inclusions, resulting in the creation of remarkable circular features.

Testing at frequencies of 100, 600, and 20,000 Hz indicated brittleness and intrinsic strength, which are typical of such materials. The fatigue limit was estimated using Murakami's technique (Murakami, 2019) to estimate the fatigue limit of a material (σ'_w), for a given inclusion area, which is written as

$$\sigma'_w = \frac{1.56(HV + 120)}{(\sqrt{area})^{\frac{1}{6}}}, \quad (2-6)$$

where HV is the Vickers hardness. It took into account both nonmetallic inclusions and optically dark areas (ODAs) that are caused by hydrogen embrittlement.

The modified S-N diagram, illustrated in Figure 2-6, reveals an interesting pattern where S-N curves for varying inclusion sizes move vertically to align fatigue limits at about the same level, represented by $\sigma_w/\sigma'_w = 1$, where σ_w is the stress of the fatigue limit for each S–N diagram. The modified S-N diagram seems to represent curves specific to notched specimens, as this correlation shows a unique curve for notched specimens (Furuya et al., 2002).

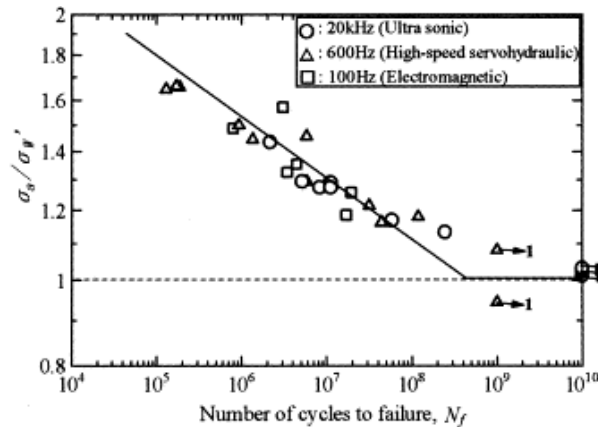


Figure 2-6 – Modified S–N diagram in which stress amplitudes are normalized with fatigue limits estimated by Murakami's equation (Furuya et al., 2002).

In Figure 2-6, the modified S-N diagram indicates that fatigue lives and fatigue limits are independent of frequency, with scattering attributed to variations in inclusion sizes of fracture origins. When taking into consideration ODA+inclusion sizes, Murakami et al. (Murakami, Nomoto, & Ueda, 1999) discovered that σ_a/σ'_w converged to about 1.0, in contrast to convergence below 1.0 for inclusion sizes alone. With σ_a/σ'_w converging to roughly 1.2 with ODA+inclusion sizes and approximately 1.0 with inclusion sizes alone, this study confirms the observed pattern. Moreover, ODAs with sizes independent of frequency were detected even in 20 kHz fatigue tests, indicating a complex relationship between fatigue damage processes and hydrogen (Furuya et al., 2002).

These results might be especially relevant to low-temperature-tempered high-strength steel that has a lot of inclusions because the frequency effects are reduced by decreased plasticity and dislocation mobility. To confirm these findings, more research utilising different steels tempered at standard temperatures is required (Furuya et al., 2002).

2.2.5. VHCF ultrasonic testing technology

The desire to comprehend material behaviour under prolonged fatigue conditions has resulted in the development of specialised testing procedures. Among these, VHCF testing has emerged as a critical method for measuring material endurance beyond conventional fatigue limits. Current ultrasonic testing machines are equipped with sensors and sophisticated measurement tools, allowing researchers to investigate material responses under high cyclic loads. A better understanding of material fatigue behaviour in the ultra-high cycle domain is explored by utilising these sophisticated testing approaches, improving the understanding of material performance under

prolonged cyclic loading conditions. Figure 2-7 displays a representative example of an ultrasonic testing machine, including the layout as well as the displacements and stresses throughout its actuator and specimen.

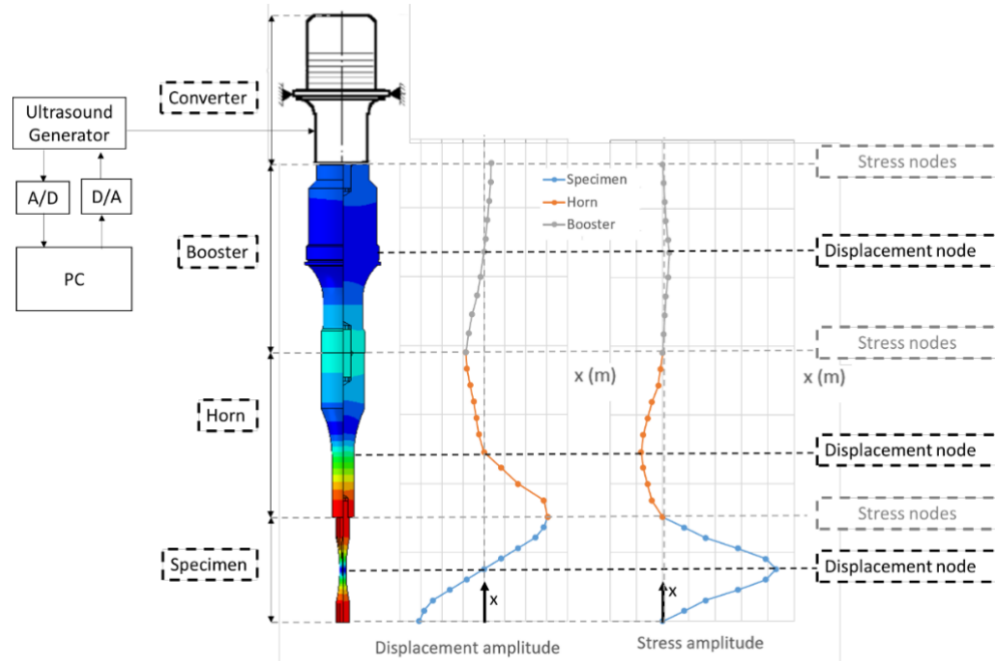


Figure 2-7 – Configuration of an ultrasonic testing machine, as well as the stress and displacement amplitudes (Fiorentin, 2023).

The piezo-electrical actuator is at the core of the system, operating with a small tolerance band at the set excitation frequency (usually around approximately 20 kHz), typically within a range of a few hundred Hz. The machine's pulsing heart is formed by this actuator, which is methodically driven by a wave generator, supplying the required signal. The amplifying horn and booster considerably improve these movements by increasing the actuator's displacements. The most important part, however, is the sample itself, which must be precisely manufactured to fulfil rigorous design criteria and tolerances. Its resonance at the required excitation frequency is essential, demanding rigorous manufacturing precision. Additionally, a thorough understanding of this resonance needs knowledge not only of the physical phenomenon but also of its operational mode (vibrations), which is heavily dependent on material characteristics such as elastic modulus and density. Surface finishing is crucial, having a significant impact on high and ultra-high cycle fatigue tests (Bathias & Paris, 2004; Fiorentin, 2023).

The machine is composed of several interconnected components that work together to provide precise fatigue assessments. During testing, auxiliary components such as displacement or velocity sensors are critical in most machines. These sensors

not only detect movement but also help to set the system's resonance, which is very useful when implementing specimens, horns, and actuators. Creating accurate S-N curves requires a good understanding of stress and the correct number of cycles. In such machinery, estimating the number of cycles for fracture start is typically dependent on detecting changes in the sample's inherent frequency. The effective load-bearing area decreases as the fracture expands, leading in a decrease in the specimen's stiffness, denoted by K .

The natural frequency (ω_n) of a system with one degree of freedom (DOF) is defined as

$$\omega_n = \sqrt{\frac{K}{M}}, \quad (2-7)$$

where M is the mass and K is the stiffness. This equation emphasises the linear relationship between natural frequency and the square root of the stiffness; as stiffness reduces due to crack formation, the specimen's natural frequency lowers as well. Although a considerable change in natural frequency may indicate severe crack formation, it is crucial to remember that evaluating natural frequency is still an indirect method.

Displacement measurements are yet another method for monitoring fracture length variations and its impact on natural frequencies. A growing crack reduces the cross-section of the specimen, lowering its stiffness. As a result, specimens with cracks display larger displacements under the same load than defect-free ones. However, recognising these stiffness-related displacement changes can be challenging, especially when dealing with modest stiffness changes caused by small cracks. The next step would be to estimate stress after determining the number of cycles for crack beginning (Bathias & Paris, 2004; Fiorentin, 2023).

2.2.6. VHCF of flat specimens

VHCF testing is critical in evaluating materials provided in thin-sheet or thin-walled forms. This section focuses on VHCF of flat specimens, including alternate methods for testing thin materials such as bending.

Bathias clearly demonstrates the distinct issues involved with flat specimens versus hourglass-shaped specimens. Because of their low transverse bending stiffness, they are prone to lateral vibrations. To alleviate this weakness, one proposed option is to use a secondary horn attached to the lower portion of the specimen, even in fatigue

tests at $R = -1$, in order to reduce lateral oscillations. Figure 2-8 illustrates a representation of this configuration (Bathias & Paris, 2004; Ebara & Yamada, 1987).

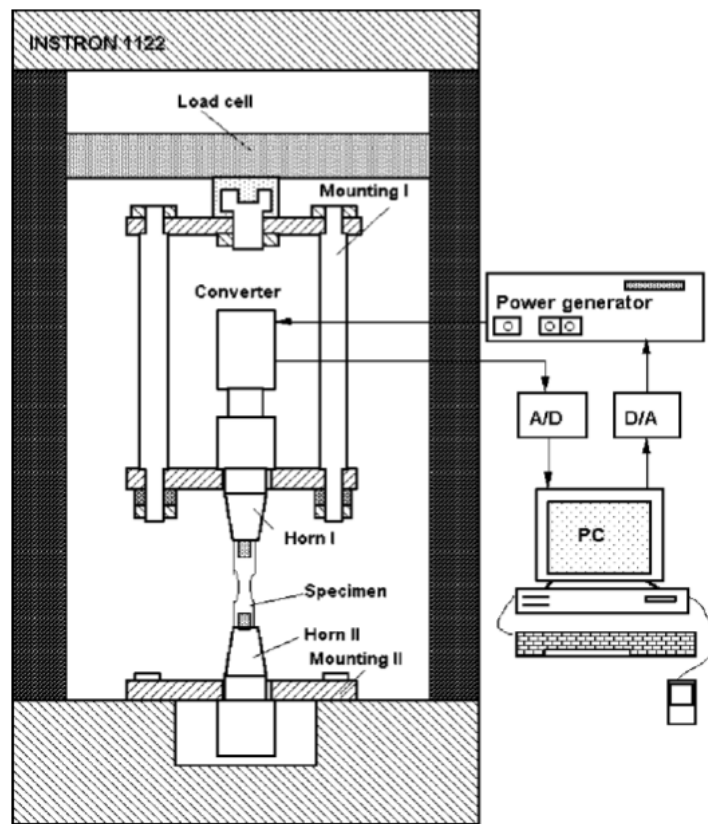


Figure 2-8 – VHCF test system for thin sheet (Bathias & Paris, 2004).

Another key challenge is fixing flat specimens for VHCF assessments. In contrast to hourglass-shaped specimens, which can easily contain a machined thread for horn fixation, connecting a thread to thin flat specimens requires inventive techniques such as riveting, welding, or gluing. For example, tests were successfully conducted on a 2-mm thick copper flat specimen, by soldering each specimen into a screw, as shown in Figure 2-9. Nonetheless, the stresses encountered during testing, while reaching 115 MPa for copper, may exceed the joint strength, particularly for materials requiring significantly greater stress levels (Bathias & Paris, 2004; Phung, Favier, Ranc, Valès, & Mughrabi, 2014).

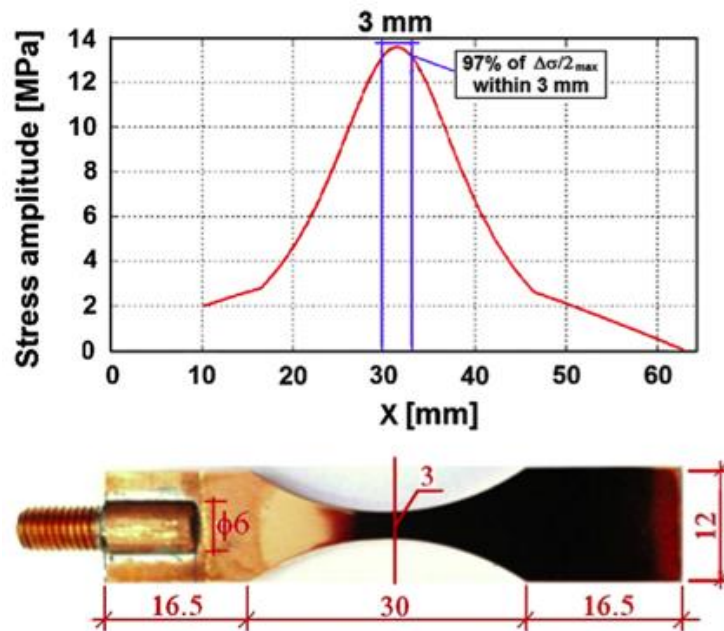


Figure 2-9 – Flat VHCF hour-glass shaped specimen with a soldered screw and stress distributions along the specimens axis x (Phung et al., 2014).

Another novel approach for VHCF evaluations was provided by Tofique et al., which involved the use of 1.0 mm thick duplex stainless steel. Figure 2-10 depicts their design and dimensions, as well as their gripping system, which was precisely engineered to provide symmetric loading and boost stress levels at the testing section for a given displacement amplitude. Testing at stress amplitudes of up to 500 MPa was possible thanks to the adoption of a titanium (Ti6Al4V) gripping system. Surprisingly, the slit cut in their grasping device was slightly shorter (0.9 mm) than the specimen thickness, ensuring a snug fit for improved attachment (Tofique, Bergström, Hallbäck, & Burman, 2018).

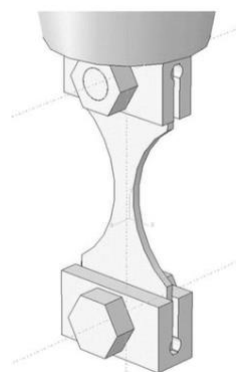


Figure 2-10 – Specimen with horseshoe fixtures (Tofique et al., 2018).

However, the assembling procedure required some flexibility in the gripping system to accommodate the specimen, resulting in the insertion of grooves in the

gripping system's internal region. These design characteristics were carefully used to lower the stiffness of the gripping system and facilitate specimen assembly. The objective of designing the parts with such characteristics was to increase the compression force between the gripper and the specimen to enhance friction. This novel approach not only demonstrated the validity of VHCF testing for flat specimens, but also highlighted the importance of precision gripping, particularly for materials requiring higher stress amplitudes (Tofique et al., 2018).

Addressing the issues that arise during the assembly process is critical. Designing the assembly (specimen and grippers) for the desired frequency, a range of minimum and maximum stresses, specimen attachment, alignment, and other considerations offer significant challenges.

Brugger et al. developed a disc geometry VHCF testing, that showed a turning point in fatigue evaluation approaches, deriving from thorough specimen design considerations. Computational models, assisted by finite element analysis, repeatedly optimised the disc's design to resonate ideally under ultrasonic frequencies, assuring the induction of equi-biaxial proportional stress states, notably at the lower face's stress-maximum centre. The disc geometry, made of cast AlSi7Cu05Mg03 T7 alloy, was precisely aligned with the material's usual yield stress of 250 MPa, reflecting real-world conditions. This geometry is shown in Figure 2-11 (Brugger, Palin-Luc, Osmond, & Blanc, 2016).

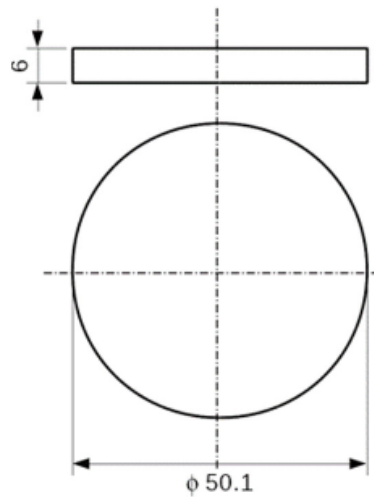


Figure 2-11 – Disc geometry, dimensions in mm (Brugger et al., 2016).

The experimental verification included various stress assessments, employing strain gauge rosettes to measure strains across the material under varied displacement amplitudes. These analytical insights were vital in linking displacement levels to stress amplitudes, providing important insights into fracture initiation mechanisms and fatigue

life predictions. To prevent self-heating effects during ultrasonic loading, the thermal management technique included specimen cooling. Analytical results revealed a median fatigue strength of 63 MPa after 10^9 cycles. In Figure 2-12, the basic principle of this ultrasonic fatigue testing machine is shown, as well as its device. Fracture analyses revealed multiple crack initiation sites, which corresponded to observations from other thin-flat specimen studies, supporting the comprehensive and nature of disc geometry VHCF testing and thus enriching our understanding of intricate fatigue behaviours under heightened stress regimes (Brugger et al., 2016).

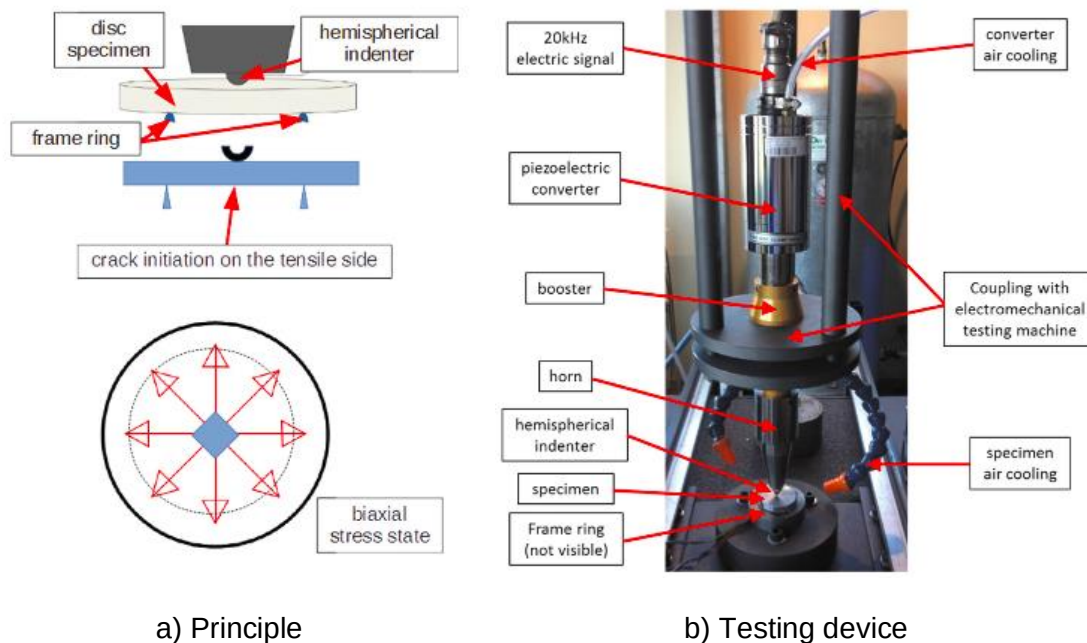


Figure 2-12 – Ultrasonic biaxial fatigue testing (Brugger et al., 2016).

This comprehensive study highlights the unique problems and innovative approaches used for UHCF testing of flat specimens, emphasising the vital relevance of precision gripping in these specialised evaluations.

David Peliteiro's dissertation, "Giga-Cycle Fatigue Behaviour of High Pressure Die-Casting Aluminium Alloys," developed at FEUP and INEGI in 2023, delves into the fatigue life study on light metal alloy structures and components. In response to societal expectations that are pushing component lifespans over traditional fatigue limitations, the concept of ultra-high cycle fatigue was explored. In this dissertation, a potential method for evaluating fatigue behaviour in very high cycle domains with the development of ultrasonic fatigue testing devices that operate at 20 kHz. However, problems remained, including the impact of frequency on damage, the influence of heat

during testing, and the need to define appropriate specimen geometries for thin components (Peliteiro, 2023).

Developing a new testing procedure designed especially for thin, flat specimens resonating at frequencies of 20 kHz was one of the main goals of this work. The idea was to design a special screw that had an opening to fix the specimen between the screw and the horn in order to solve the problem of securely fastening and transmitting mechanical waves from the machine to the specimen. An image of the assembly of the screw and the specimen is displayed in Figure 2-13 (Peliteiro, 2023).

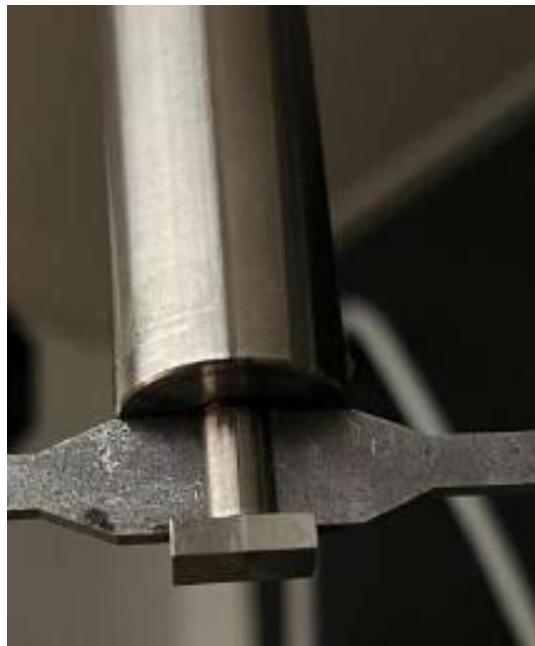
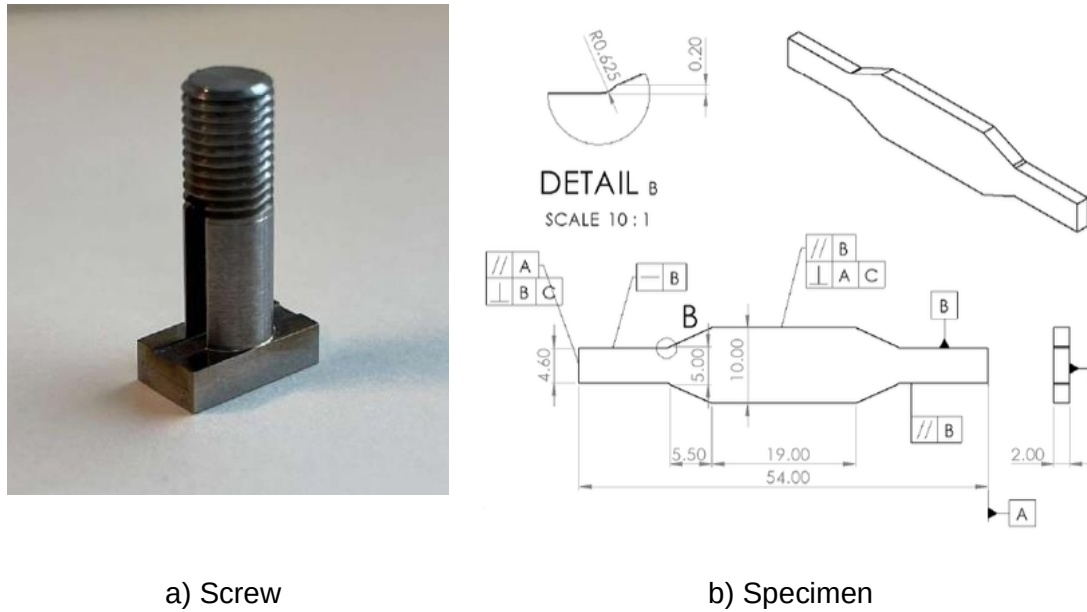


Figure 2-13 – Assembly of the screw and the specimen (Peliteiro, 2023).

Achieving a system frequency near 20.0 kHz while making sure that the lowest stress applied in the concentration area was within acceptable limits required several iterations during the specimen design process. To optimise testing conditions, the titanium screw and specimen's final dimensions and design were carefully chosen, shown in Figure 2-14 (Peliteiro, 2023).



a) Screw

b) Specimen

Figure 2-14 – Screw with rectangular slot and thin-plate specimen (Peliteiro, 2023).

AlSi10MnMg (Silafont-36) alloy thin sheets from casting components were used to create flat specimens. Specimens with the required dimensions were obtained using wire EDM (Electrical discharge machining) and machining procedures. The thin sheet specimens were polished. The casting component specimens had a complex production process because of the shape of the original part. Interesting insights into the behaviour of flat specimens under bending loading were obtained through fatigue testing. The fatigue lives of smooth and flat specimens were compared, and it was shown that the flat specimens had a shorter fatigue life at some stress levels. Additionally, the fatigue performance of cast components was affected by the thermal treatments that were performed, highlighting the importance of material processing for determining fatigue resistance (Peliteiro, 2023).

2.2.7. Behaviour of AlSi10MnMg alloy and its composition

The AlSi10MnMg alloy, commercially also known as Silafont-36, originates from Rheinfelden Alloys' advances in pressure die casting materials and is widely employed in the automotive industry due to its versatile mechanical properties, making it a suitable candidate for various industrial applications. This alloy was specifically chosen for the GCYCLEFAT project, highlighting its importance in the development of setups for high-cycle fatigue testing. The project intends to address the need for comprehensive testing methodologies to evaluate the fatigue behaviour of the alloy by designing a testing setup capable of subjecting it to axial compressive loads. Because of its remarkable qualities,

such as its high tensile strength, yield strength, and elongation properties, Silafont-36 is a suitable material for application in structural automotive components. Its ability to withstand corrosion and fatigue further increases its suitability for use in a variety of environments. While the development of testing setups for aluminium alloys is the primary focus of this work, knowledge of the behaviour and composition of Silafont-36 is still important in order to comprehend its mechanical properties and possible uses in a variety of industries.

This alloy, known for its diversified mechanical properties, has an excellent capability for enhancement through various thermal treatments. Notably, its tensile properties can be increased to reach 15% elongation, but it also can reach a yield strength of approximately 260 MPa, making it a versatile material for a wide range of industrial applications. Figure 2-15 depicts the alloy's mechanical strength response in different heat treatment states, particularly yield strength and elongation for different magnesium levels, providing vital insights into its mechanical behaviour under varying situations (Rheinfelden).

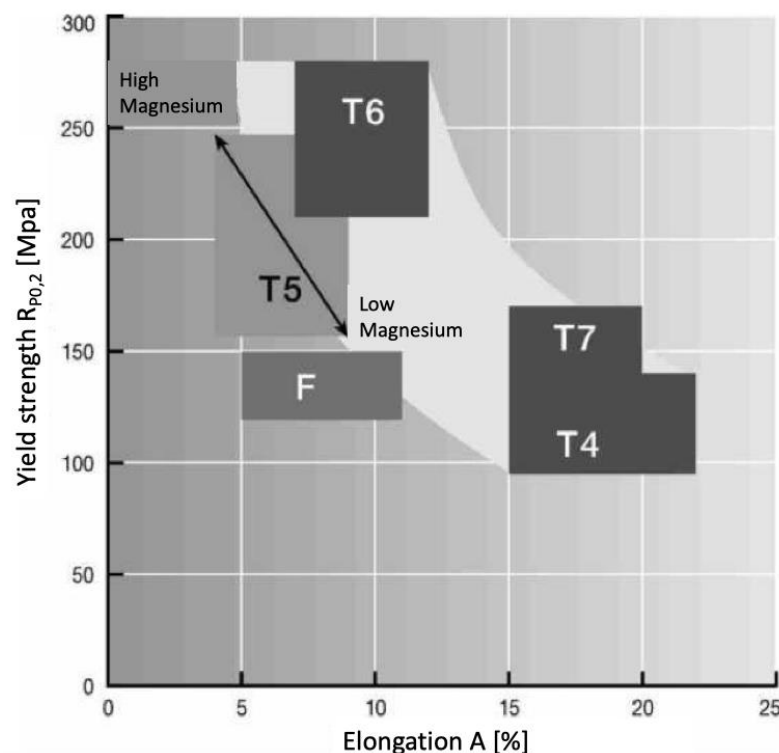


Figure 2-15 – Yield strength and elongation in various heat treatment states of Silafont-36 (Adapted from Rheinfelden (2013)).

Silafont-36 has a number of inherent features that highlight its extraordinary utility in pressure die casting operations. Among its many advantages is its excellent die cast castability, which is characterised by the absence of sticking to the die, resulting in

a smooth and quick casting process. Its great machinability adds to its popularity in manufacturing operations, allowing for easy fabrication and moulding.

The alloy's prowess extends particularly to the automobile industry, where its set of properties becomes especially valuable. Its exceptional corrosion resistance acts as a barrier against environmental degradation, increasing the lifetime and reliability of automotive components exposed to varied environments. Its high fatigue strength is also impressive, indicating resilience and endurance, which are critical in vehicle structural elements.

Silafont-36's adaptability extends beyond mechanical toughness, making it suitable for a wide range of manufacturing procedures. Its weldability, particularly in aluminium profile-cast designs, demonstrates its versatility in joining techniques. Notably, it is an excellent option for self-piercing riveting and comparable joining processes, demonstrating its compatibility with modern production techniques. Furthermore, its applicability for adhesive bonding in vehicle design demonstrates its versatile and all-encompassing utility across assembly processes (Rheinfelden, 2013). Table 2-2 displays the chemical composition of the Silafont-36 alloy.

Table 2-2 – Silafont-36's chemical composition (AlSi10MnMg) (Adapted from Rheinfelden (2013)).

Wt%	Si	Fe	Cu	Mn	Mg	Zn	Ti	Sr	Other
min	9,5			0,5	0,1		0,04	0,010	
max	11,5	0,15	0,03	0,8	0,5	0,07	0,15	0,025	0,10

2.3. Mechanical wave transmission model

In classic fatigue testing, the test machine's external load system enforces a frequency that differs from the specimen's inherent frequencies, causing forced vibration. Ultrasonic fatigue tests, on the other hand, require that the test machine's external frequency exactly match one of the specimen's natural frequencies. Therefore, the mechanical wave classical transmission model serves as the foundation for calculating strain, stress, and displacement on a specimen, as well as for analytical determination in this area (Bathias & Paris, 2004).

2.3.1. Longitudinal elastic waves and resonance frequency

Using the elastic wave theory, the differential equations for a generic three-dimensional isotropic elastic body in a Cartesian coordinate system are (Bathias & Paris, 2004):

$$\rho \frac{\partial^2 u}{\partial t^2} = \frac{E}{(1+\nu)} \left(\frac{1}{1-2\nu} \frac{\partial e}{\partial x} + \nabla^2 u \right), \quad (2-8)$$

$$\rho \frac{\partial^2 v}{\partial t^2} = \frac{E}{(1+\nu)} \left(\frac{1}{1-2\nu} \frac{\partial e}{\partial y} + \nabla^2 v \right), \quad (2-9)$$

$$\rho \frac{\partial^2 w}{\partial t^2} = \frac{E}{(1+\nu)} \left(\frac{1}{1-2\nu} \frac{\partial e}{\partial z} + \nabla^2 w \right), \quad (2-10)$$

where u , v , and w are displacements along x , y , and z respectively, ρ is the material density, t the time, E the elastic modulus, ν the Poisson's ratio, ∇^2 the Laplacian, and:

$$e = \frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} + \frac{\partial w}{\partial z}, \quad (2-11)$$

where e is the volume dilatation.

According to elastic wave theory, an infinite isotropic elastic body can contain the two types of waves shown below. For a longitudinal wave, the curl of the displacement field is zero and the velocity of wave propagation (c) is (Bathias & Paris, 2004):

$$\rho \frac{\partial^2 u}{\partial t^2} = \frac{E}{(1+\nu)} \left(\frac{1}{2(1-2\nu)} + \frac{1}{2} \right) \frac{1}{c^2} \frac{\partial^2 u}{\partial t^2}, \quad (2-12)$$

$$c = \sqrt{\frac{E(1-\nu)}{(1+\nu)(1-2\nu)\rho}}. \quad (2-13)$$

For a transverse wave, the volume dilatation e disappears and the velocity of wave propagation is:

$$\rho \frac{\partial^2 u}{\partial t^2} = \frac{E}{(1+\nu)} \left(\frac{1}{(1-2\nu)} * 0 + \frac{1}{2} \right) \frac{1}{c^2} \frac{\partial^2 u}{\partial t^2}, \quad (2-14)$$

$$c = \sqrt{\frac{E}{2(1+\nu)\rho}}. \quad (2-15)$$

If there is a boundary, a surface wave may be generated. This wave is comparable to the gravity surface wave in a fluid. The amplitude of vibration falls with the distance from the surface, and the wave velocity is lower than the velocity within the body. The explanation is simplified by using a one-dimensional example of a straight cylinder. As seen in Figure 2-16, a longitudinal elastic wave originates at one end of the bar and travels the length ($l = x$) before being reflected at the other end and returning to the original point of entrance. U is the displacement, ε the strain, k the wave number, and A_0 the displacement amplitude (Bathias & Paris, 2004).

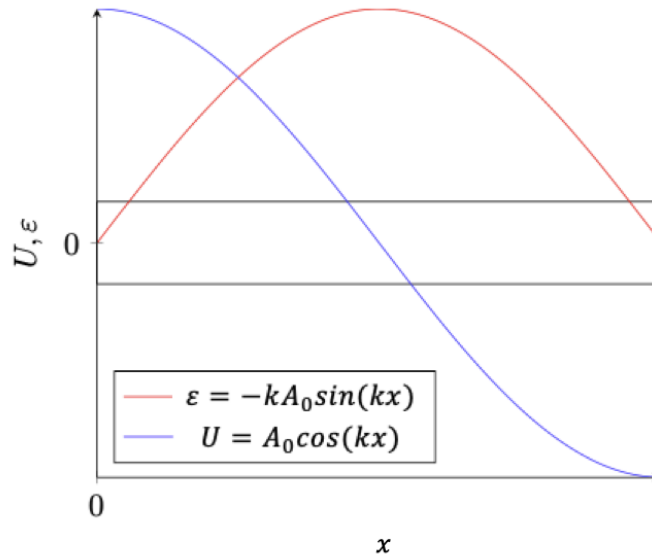


Figure 2-16 – Displacement and strain variation along an elastic bar (Adapted from Bathias and Paris (2004); Gouveia (2022)).

For this one-dimensional case, the wave velocity c will be derived simply from Equation (2-12), with $\nu=0$ (Bathias, 1999; Kolsky, 1964):

$$c = \sqrt{\frac{E}{\rho}}. \quad (2-16)$$

The subsequent deductions will most likely be influenced by this reduction of a three-dimensional problem to a one-dimensional method. The differential Equations (2-8), (2-9), and (2-10) are reduced to a single equation

$$\frac{\partial^2 u}{\partial t^2} = \frac{E}{\rho} \frac{\partial^2 u}{\partial x^2}, \quad (2-17)$$

the solution to Equation (2-17) can be obtained by

$$u = \sum_{n=1}^{\infty} u_n(x, t), \quad (2-18)$$

with

$$u_n(x, t) = \left(A_{n-1} \cos \frac{n\pi ct}{l} + B_{n-1} \sin \frac{n\pi ct}{l} \right) \cos \frac{n\pi x}{l}. \quad (2-19)$$

The boundary conditions of ultrasonic fatigue testing demand maximal displacement at both ends (A_0), whereas strain disappears at the same locations. That means,

$$\left(\frac{\partial u}{\partial x} \right)_{x=0,l} = 0, \quad (2-20)$$

therefore, Equations (2-18) and (2-19) for the first mode of vibration ($n = 1$) are

$$u_n(x, t) = A_0 \cos(kx) \sin(\omega t), \quad (2-21)$$

where

$$k = \frac{\pi}{l}, \omega = \frac{\pi c}{l}. \quad (2-22)$$

The amplitude of vibration at each point along the bar is

$$U(x) = A_0 \cos(kx), \quad (2-23)$$

where A_0 is the displacement amplitude at the end of the bar. The strain ε of each point is supplied by the derivative of Equation (2-23)

$$\varepsilon(x, t) = -kA_0 \sin(kx) \sin(\omega t), \quad (2-24)$$

with its maximum

$$\varepsilon(x, t) = -kA_0 \sin(kx). \quad (2-25)$$

The strain rate is given by the derivate of Equation (2-25)

$$\dot{\varepsilon}(x, t) = -k\omega A_0 \sin(kx) \cos(\omega t), \quad (2-26)$$

with its maximum

$$\dot{\varepsilon}(x, t) = -k\omega A_0 \sin(kx). \quad (2-27)$$

Based on Equations (2-16) and (2-22), the length of the bar is

$$l = \frac{1}{2f} \sqrt{\frac{E_d}{\rho}}, \quad (2-28)$$

to generate the first mode of vibration, where $f = \omega/(2\pi)$ is the frequency and E_d the dynamic elastic modulus to account for dynamic effects.

Finally, Equation (2-28) provides the length of resonance of the one-dimensional specimen for the first vibration mode. The displacement node (where the displacement is zero) at the middle of the specimen corresponds to the maximum strain, stress, and strain rate. The greatest displacement occurs at both ends, which represents the nodes of strain, stress, and strain rate. That means (Bathias & Paris, 2004):

$$u = 0, \varepsilon = -kA_0, \sigma = -E_d kA_0, \dot{\varepsilon} = -k\omega A_0 \quad \text{for } x = \frac{l}{2} \quad (2-29)$$

$$U = A_0, \varepsilon = 0, \sigma = 0, \dot{\varepsilon} = 0 \quad \text{for } x = 0, l \quad (2-30)$$

Equation (2-28) shows that the resonance length is inversely proportional to frequency. This explains why some extremely high frequencies are not practical. These frequencies result in smaller specimens that are more difficult to machine, monitor displacement or strain, and dissipate energy. Table 2-3 shows an example of a typical steel specimen of uniform section with $E = 200\,000$ MPa and $\rho = 7800$ kg/m³. Different specimen lengths are available based on the resonance frequency (Bathias & Paris, 2004).

Table 2-3 – Comparison of the specimen's length at various test frequencies
(approximate values) (Bathias & Paris, 2004).

$f(kHz)$	$l(mm)$
20	127
92	27.5
199	13.2

The final two values of l clearly cause problems with machining, displacement, and strain measurements, as well as energy dissipation.

3. VHCF TESTING DETAILS AND SPECIMEN DESIGN

3.1. Experimental apparatus for ultrasonic tests

The Shimadzu USF-2000 machine, which was primarily intended to test smooth cylindrical specimens, served as the central equipment of the ultrasonic testing apparatus used in this investigation. However, for the purposes of this research, thin and flat specimens needed to be tested, which meant adapting the machine setup. This chapter offers an extensive overview of the adjustments set in place to meet the special needs of using the Shimadzu USF-2000 machine for testing flat specimens.

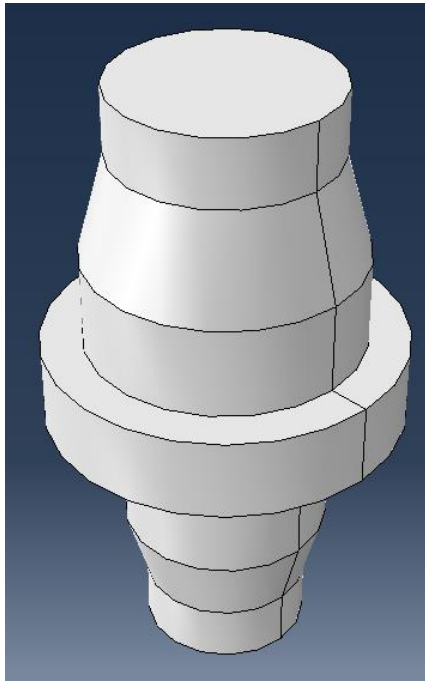
3.1.1. Machine information

In setting up the VHCF testing apparatus, several crucial machine parameters have been carefully defined for the Shimadzu USF-2000. The machine's operating frequency, which is a critical factor affecting fatigue behaviour, is limited to a tight range of 19.5 kHz to 20.5 kHz. According to the testing methodology's requirements, this narrow frequency range is required to keep the Shimadzu USF-2000 functional and reliable. It is imperative to ensure that there are no other vibrational modes present within a similar frequency range to the desired one, as this could compromise the accuracy of the tests. Some of its components, such as the horn and the booster, were already displayed in Figure 2-7.

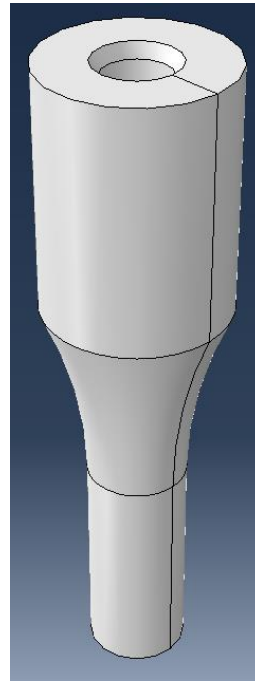
Carefully controlled displacement amplitude is a major factor affecting stress levels and directly affecting fatigue performance. The upper limit for cyclic loading was determined to a maximum displacement amplitude of 55 μm . In order to create a controlled loading spectrum, the minimum displacement amplitude is intentionally maintained at 20% of the maximum. This proportionate relationship between stress levels and displacement amplitude underscores the significance of these metrics in accurately describing material fatigue.

The horn and the booster are crucial parts of the Shimadzu USF-2000 apparatus that are necessary for accurate and controlled VHCF testing. The horn with a tapered geometry is essential to send ultrasonic vibrations to the specimen. For effective energy transfer to occur and for testing to minimise energy loss, its mechanical design is essential. The booster is positioned next to the horn and acts as an amplifier to increase the displacements that the piezo-electric actuator produces. The mechanical properties of the booster contribute to the system's overall effectiveness and performance.

The booster and horn show their geometries in Figure 3-1. Generally speaking, the horn has an extended, thin shape, while the booster has an amplifying shape that maximises energy transfer. A smooth energy transfer path is ensured by the well-designed connection between these parts. It is important to notice that the machine's operating frequency is measured at about 19.870 kHz, without any specimen attached to it.



a) Booster



b) Horn

Figure 3-1 – Illustration of the booster (a) and horn (b) of the Shimadzu USF-2000 testing machine.

Table 3-1 provides a complete summary of the mechanical characteristics of the horn, booster, and also of the parts of the specimen's assembly, which are going to be presented in the following sections.

Table 3-1 – Mechanical properties of the parts (Peliteiro, 2023).

Part	Material	$E(\text{Pa})$	ν	$\rho(\text{kg/m}^3)$
Booster	Titanium Alloy	1.14e^{11}	0.34	4430
Horn	Titanium Alloy	1.14e^{11}	0.34	4430
Specimen	Silafont-36	7.80e^{10}	0.33	2640
Gripper	Titanium Alloy	1.13e^{11}	0.34	4430
Bolt and nuts	Steel	2.00e^{11}	0.3	7800

A unique characteristic of the Shimadzu USF-2000 that had a big impact on the specimen design is the horn's threaded M6x0.75 hole. This threaded system is essential for firmly fastening the specimen to the horn. In order to ensure a strong and sturdy connection during testing, the specimen design needs to include a threaded screw for a seamless connection and fastening to the machine, displayed in Figure 3-2. The system with the horn, booster and specimen is illustrated in Figure 3-3.

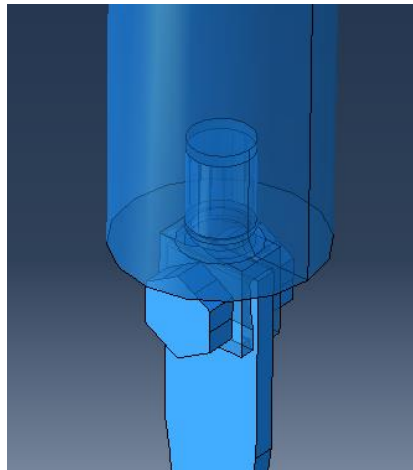


Figure 3-2 – Threaded screw of the gripping component placed inside the machine's hole.

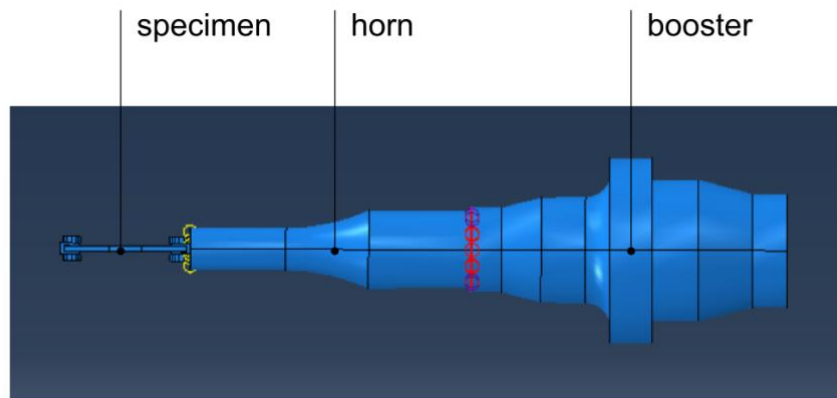


Figure 3-3 – Representation of the assembly of a specimen with horn and booster.

3.1.2. Experimental apparatus components

The USF-2000 machine made by Shimadzu, shown in Figure 3-4, was used for the ultrasonic testing, which was carried out, as stated before, at a frequency of 20 kHz.



Figure 3-4 – Shimadzu USF-2000 machine (Peliteiro, 2023).

An overview of the several parts and components used in connection with the USF-2000 machine is displayed in Figure 3-5 and Table 3-2. Each part and component is essential to the testing procedure.

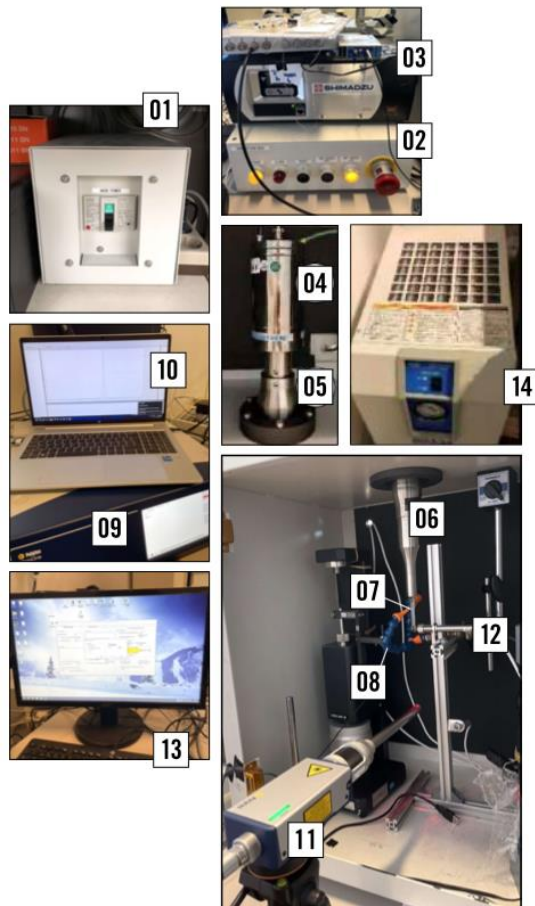


Figure 3-5 – Machine components labels.

Table 3-2 – Components from Figure 3-5.

Label	Component
1	Current stabilizer
2	Operation box
3	Signal generator from Branson
4	Piezoelectric Transducer
5	Booster
6	Horn
7	Specimen
8	Compressed air nozzle
9	VibroOne controller from Polytec
10	Laptop with Data acquisition software - <i>VibSoft</i>
11	Vibrometer Sensor head from Polytec
12	Pyrometers
13	PC to control the test
14	Air dryer

An electric current stabiliser provides power to all system components and accessories, guaranteeing steady and reliable functioning during testing procedures. The Operation box, one of the system's most useful components, is notable for its versatility since it enables emergency shutdown, error reset, and machine power control. Furthermore, as stated in Section 2.2.5, parts 5 through 7 play a major role in the testing apparatus's overall functionality and control.

In addition, the integration of the Polytec VibroOne laser Doppler vibrometer, particularly the VIO-BW-500kHz model with a bandwidth up to 500 kHz and the VIO-DispResH option offering high resolution down to 1 μm peak displacement, was pivotal in accurately capturing the minute displacements of the specimen under high-frequency loading conditions (Polytec, 2019). It successfully reduced any interference with the specimen's dynamics by employing non-contact measurement principles, guaranteeing accurate capture of data. The high-frequency response and advanced signal processing capabilities of the VibroOne allowed for the extraction of crucial modal data, with the data acquisition system (DAQ) effectively processing this information to provide a detailed temporal displacement dataset, essential for synchronizing displacement data with other test parameters and for real-time monitoring and analysis of the specimen's dynamic response under cyclic loading.

Compressed air is another essential part of the experimental setup since it controls temperature. First of all, it helps keeping the transducer cold while the machine is running, which guarantees longevity and peak performance. In addition, while testing, pressurised air is directed towards the specimen under the guidance of a nozzle, which efficiently regulates its temperature and upholds constant testing conditions.

Pyrometers make it easier to monitor temperature by providing accurate measurements of component temperatures at various points of interest. Pyrometers are useful tools for evaluating thermal conditions, whether they are used to monitor the specimen, horn, or screw head temperature during flat specimen testing. Applying black paint to the region of interest will improve compatibility with pyrometer measurements by reducing issues with metal surface emissivity and reflection and ensure accurate temperature measurement.

When traditional temperature measurement techniques are difficult to use, like with flat specimens, a thermal camera is a good alternative. By allowing non-contact monitoring of temperature, the thermal camera offers comprehensive insight into the distribution of temperature and the transmission of stress all over the surface of the specimen. Researchers can obtain significant insights into the thermal dynamics of the testing environment by utilising modern thermal imaging technologies. This can lead to improved experimental outcomes and informed decision-making.

3.1.3. Flat specimen test in the Shimadzu

Careful planning and attention to detail are essential in order to execute tests on the Shimadzu USF-2000 equipment with reliability and precision. An important first step is assembling the specimen, which includes aligning the grippers, bolts, and nuts. All these parts of the assembly are going to be described in Section 3.2. Ensuring correct alignment along the safe axis reduces the risk of symmetry issues which may affect machine performance or negatively impact the results of the tests.

The very first thing to do in alignment is to line up the grippers with the specimen. Then, use a torque wrench to tighten the bolts and nuts. It is essential to apply torque properly to every fastener during testing to avoid loosening, which could lead to machine issues. Once tightened, the specimen set is manually screwed onto the horn's top part to prevent torsion on the test piece. Once the stability of the specimen is secured, the vibrometer sensor head is carefully set under the specimen in order to avoid interference with surrounding components. In addition, extreme caution is exercised to avoid obstructing the laser's path, ensuring uninterrupted transmission for interaction with the specimen and reception of the subsequent signal. To improve signal reflection from the

specimen's bottom gripper, reflective paper is implemented, as seen in Figure 3-6. This adhesive application ensures that the sensor head captures signals as efficiently as possible, improving data acquisition fidelity.



Figure 3-6 – Picture of gripper with reflective paper.

After carefully placing compressed air nozzles to cool the specimen, the valve is opened, and a frequency check is performed to ensure that the specimen is securely fixed.

The vibrometer is then used to calibrate the Shimadzu machine's displacement system. A calibration curve is uploaded into the Supersonic Software to establish a predetermined power and achieve the desired nominal stress on the specimen, allowing for fatigue testing. The DAQ system is used to generate a data file, and measurements are recorded at specific power levels (e.g., 20% and 40%), while the remaining values are iterated for the entire range. The gain value is calculated using an Excel spreadsheet, allowing for accurate conversion of vibrometer readings into experimental displacement values.

The Pyrotec data acquisition system performs subsequent data analysis. The system uses the data file to generate a file containing the resonance frequency, conversion factor, and measured displacement values. In addition, time-displacement curves and Fast Fourier Transform (FFT) plots are created to investigate the specimen's dynamic response. Testing is limited to predetermined stress limits to avoid causing excessive damage to the specimen.

System settings in the Supersonic software are carefully configured, including material properties, specimen geometry, and stress conversion factors. Intermittent driving tests reduce the risk of overheating by allowing specimens to cool down during stops. Throughout the testing process, precise monitoring and control are maintained by specifying test execution parameters such as cycle count and test interval.

Data is saved after testing is completed, ensuring comprehensive documentation and traceability of experimental results for future analysis and interpretation.

3.2. Specimen Design

This section's main focus is on the careful design of specimens used in gigacycle fatigue testing. It makes its way through a workflow of specimen design, emphasising how important it is to match the specimen's size exactly to the testing machine's set excitation frequency. This method's key component is reaching a zero-displacement node near to the midpoint of the specimen's gauge section, which is strategically placed to maximise stress concentration. This section begins with an examination of the project's idea, covering topics like material selection and testing machine conditions. This sets the stage for a detailed exploration of the design and function of each component in the specimen assembly.

3.2.1. Project inspiration

The methodology is inspired by Tofique et al. (2018)'s unique gripping mechanism and assembly procedures, as depicted in **Error! Reference source not found.** The following sections of this chapter will go into the technical materials, geometry, and procedures used, providing an extensive review of the specific adaptations and contributions to the VHCF testing framework.

3.2.2. Project characteristics

This section explores the fundamental qualities that characterise and direct the VHCF testing project. The specimen's material is the first area of attention, with the AlSi10MnMg alloy, also known as Silafont-36, taking centre stage. The mechanical properties of this alloy have to be carefully evaluated, and the calculated stress parameters are critical to keeping cyclic loads within elastic bounds and guaranteeing the dependability and safety of the specimens.

We also go into detail about the Shimadzu USF-2000 equipment, including important machine characteristics like displacement amplitude and operation frequency. Upon closer examination, parts such as the booster and horn exhibit intricate mechanical details that are necessary for precise VHCF testing. The groundwork for a solid understanding of the project's parameters and how they affect the research is established by these components.

3.2.2.1. Specimen's material

The AlSi10MnMg alloy, frequently referred to as Silafont-36, is the main focus in this investigation due to its exceptional mechanical properties. As previously described in the state of the art, Silafont-36 exhibits a wide range of mechanical characteristics.

The mechanical properties of Silafont-36 were carefully considered when designing the specimen and conducting the tests. Calculating the greatest and lowest stress levels that the specimen could experience was a critical step in this process. The maximum stress desired values were carefully adjusted below the previously indicated maximum yield strength (about 260 MPa), as seen in Figure 2-15. This is an essential parameter for ensuring that cyclic loads stay within the material's elastic limits. This strategic decision emphasises the importance of maintaining a controlled loading range, prioritising safety margins, and preserving the Silafont-36 specimens.

Additionally, the minimum stress was precisely calculated to be 20% of the maximum stress, which helped to create a loading scenario that was balanced and appropriate for the material's fatigue behaviour. In order to prevent specimen failure during testing, these stress parameters were carefully selected, highlighting the importance of a methodical approach to stress determination.

The specimen's mass considerations and the alloy's elastic response to loading are established, with a Young's modulus of 78 GPa and a density of 2640 kg/m³ (Peliteiro, 2023). By combining these material characteristics into the testing settings, this work seeks to expand our understanding of Silafont-36's mechanical performance in high and ultra-high cycle fatigue regimes, in addition to elucidating the material's fatigue behaviour.

3.2.3. Part design

3.2.3.1. Bolt and nuts

The selection of bolts and nuts was crucial in the part design for VHCF testing, especially when it comes to gripping the specimen. The specimen's stability, which depended on the friction force produced between the gripper and the specimen (shown in Figure 3-7), was the main concern.

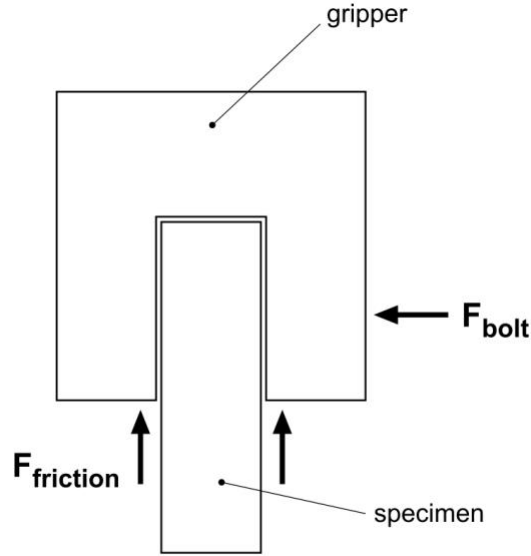


Figure 3-7 – Illustration showing the friction force between the gripper and the specimen as well as the preloaded bolt force applied to the gripper.

Knowing that the friction force can be calculated from:

$$F_{friction} = \mu F_N, \quad (3-1)$$

where μ is the friction coefficient and F_N the normal force, the goal was to increase the gripping strength by maximising the normal force. The normal force is obtained with:

$$F_N = F_{bolt} - F_{bending}, \quad (3-2)$$

where F_{bolt} is the preloaded force applied by the bolt on the gripper and $F_{bending}$ is the force required for the gripper's bending deformation. The idea was to maximise the normal force by minimising this reaction force, which is going to be discussed in the next section. Increasing the friction force that hold firmly the specimen is the ultimate goal. Notably, the preloaded force applied by the bolts played a significant role during the selecting process.

The M4, class 12.9 steel bolt was the optimal choice. This choice was based on the bolt's highest resistance, which provided the highest preloaded force, measured at a notable 6584.4 N, a value retrieved from (TSF). The design aims to maximise gripping efficiency by carefully combining these components, resulting in a safe and dependable groundwork for VHCF testing.

3.2.3.2. Gripper

The gripper, an accurately designed part that holds the specimen in position, is essential to the success of the testing. Achieving a balance between force transmission efficiency and flexibility is the main goal of the gripper design. This means that the stiffness (k) (see equation (3-3)) of the gripper needs to be reduced as much as possible. An indicator of the gripper's flexibility would be of much that each flap could bend without failing (x):

$$k = \frac{F_{bolt}}{x} . \quad (3-3)$$

The gripping deformation is displayed in Figure 3-8.

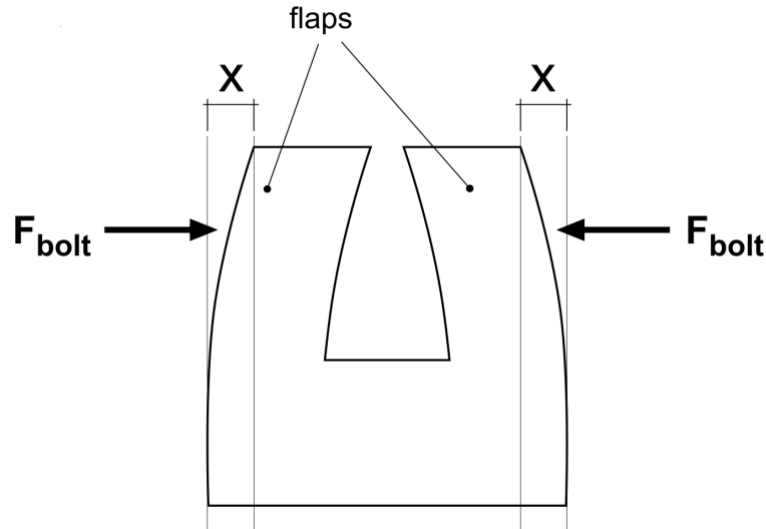


Figure 3-8 – 2D illustration of the gripper's flaps bending.

The gripper's stiffness (k) is directly related to moment of inertia (I),

$$I = \frac{Wt^3}{12} , \quad (3-4)$$

where W is the width and the t is the thickness. In Figure 3-9, a 3D illustration of the gripper's idea of design can be observed. Thus, the best course of action was to reduce the thickness of the flaps nearer the bottom of the gripper. This careful modification maximises the gripper's flap displacement while minimising the force required during gripper bending. This particularity can be observed later on the final design of the gripper.

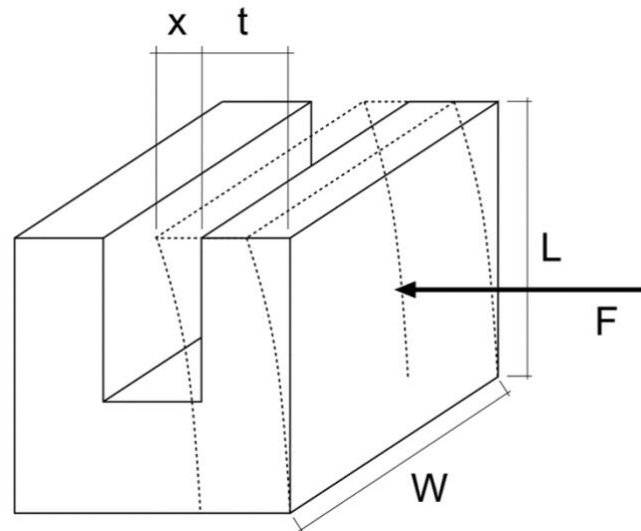


Figure 3-9 – 3D illustration of the gripper's idea of design

The selection of titanium over steel and aluminium for the gripper is a result of a careful evaluation of material properties and their compatibility with the intended application. First, because of its less advantageous characteristics for threaded applications, aluminium was excluded. Because of aluminium low strength, it is not as well suited to the unique requirements of the gripper design, where tight preloaded fastening is crucial. Secondly, titanium was thought to be a better material for this application than steel, despite steel could have higher strength. The decisive element was the notable difference in Young's Modulus between steel and Titanium. Stiffness is proportional to Young's Modulus, and Titanium's Young's Modulus is significantly lower than of steel, almost two times lower. Choosing steel would have reduced the flexibility that is required, producing a gripper with double the stiffness for the same geometric configuration. Titanium is the perfect material for the threaded and flexible gripper design since it guarantees the gripper's flexibility without compromising structural integrity.

The geometric changes made were a result of an iterative process rather than a unilateral judgement. This iterative refinement occurred in accordance with the development of the specimen, emphasizing the interdependence of specimen and gripper optimisation. Decreasing W reduces stiffness because the cross-section area decreases with W , and increasing the bolt load (F in Figure 3-9) distance to the gripping base also helps to reduce stiffness. Using Abaqus simulations, these modifications resulted in a final gripper stiffness of 1.95 kN/mm. This value was found by measuring the deflection of the gripper's flaps when subjected to a known bolt preload force, already mentioned before.

3.2.3.3. Flat specimen

This section dives into the core of the experimental setup, thoroughly investigating all of the aspects of geometry optimisation, modal analysis, and specimen design. The first part studies the rectangular specimen, which was the initial option for fatigue testing. The second part looks into the "dog bone" specimen, which is an improved design that aims to improve mechanical properties and test accuracy. This section leads the specimen refining process by iterative geometric adjustments and comprehensive modal analysis, emphasising the pursuit of improved fatigue performance and greater understanding of material behaviour under cyclic loading circumstances.

3.2.3.3.1. Rectangular specimen

The rectangular specimen design serves as the basis element in the exploration of fatigue behaviour under cyclic loading conditions. The specimen's simplified representation, shown in Figure 3-10, illustrates its geometric configuration and includes important dimensions and features that are necessary for accurate analysis and testing.

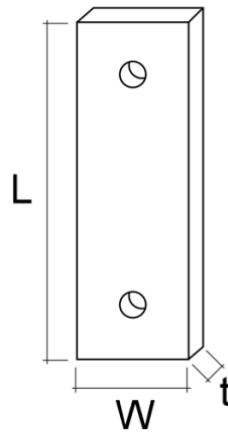


Figure 3-10 – 3D illustration of the rectangular specimen's idea of design.

The basis of the specimen's behaviour is the principle of linear elasticity, wherein stress and displacement display a linear relationship within the material's elastic limits. This crucial property makes it possible to use the concepts of linear elastic fracture mechanics to project the start and spread of cracks, which is essential for evaluating fatigue life.

Modal analysis turns into an essential tool in the attempt to optimise the specimen's mechanical properties and resonance characteristics. By applying

established formulas from the field of VHCF, as equation (2-7), modal frequencies can be calculated according to stiffness (k) and mass (m). Adjusting the specimen's modal frequency becomes possible by manipulating parameters like mass (m) and length (L), because stiffness is dependant from the L according to,

$$k = \frac{E \cdot A}{L}. \quad (3-5)$$

Considering the equations (2-7) and (3-5), reducing mass raises the frequency, and increasing the length decreases stiffness, consequently decreasing the modal frequency. Most notably, all assembly components' influence is taken into account in these computations and subsequent Abaqus simulations, guaranteeing complete and precise analysis. These were the pivotal criteria in the design of the specimen.

According to initial simulations, the specimen's stress levels were below the desired, requiring additional modifications to achieve the ideal stress conditions. The following section details the iterative procedure used to raise stress levels while preserving the integrity of the specimen and test validity.

3.2.3.3.2. "Dog bone" specimen

The "dog bone" specimen represents an evolved version of the rectangular design, incorporating strategic modifications to enhance stress levels and frequency characteristics. As depicted in Figure 3-11, a notable divergence lies in the variation of width (W) within the central section of the specimen, which differs from the uniform width of the rectangular alternative.

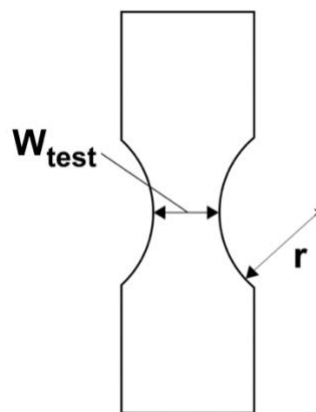


Figure 3-11 – 2D illustration of the "dog bone" specimen.

After careful analysis of stress distribution and frequency modulation, the decision was made to add variation in the specimen's central section width (W_{test}). The specimen's cross-sectional area is essentially reduced by decreasing W_{test} . Since stress is inversely proportional to the cross-sectional area (A), this area reduction increases the concentration of stress. Furthermore, because mass (m) is inversely proportional to area (A) and directly affects the modal frequency through equation (2-7), this decrease in cross-sectional area naturally modifies the specimen's modal frequency. Then, another iterative process has to be done until assuring that the frequency and stress levels are as desired.

Once all iterative changes were made to achieve proper frequency and stress levels, final simulations took place to confirm the specimen's performance in all aspects. However, these needed to be tested with symmetries-free simulations, guaranteeing an accurate representation of resonance characteristics and vibrational modes. This all-encompassing method reduces the occurrence of additional vibrational modes (e.g. bending) that have frequencies similar to the desired frequency, which could compromise the validity and integrity of later testing.

3.3. Numerical Simulation

This section explores the modal analysis simulations performed using Abaqus/CAE 2020, which are critical for understanding the dynamic behaviour of the specimen and machine components under consideration.

Initially, simulations were only focused on the specimen, improving its response to the desired frequency. All simulations performed during the iteration process to achieve the required frequency employed symmetry across the three axes to simplify the computational model. However, the final specimen simulations were symmetry-free in order to rigorously confirm the absence of competing vibrational modes near the target frequency of 20 kHz. After achieving this goal, the machine components were introduced into the model, ensuring that the global frequency would remain unchanged from the specimen alone, and also to make sure no unintended vibrational modes near the 20-kHz limit were present in the simulation. This comprehensive approach highlights the delicate relationship of simulation fidelity and experimental realism.

Identifying contact interactions between these parts was a critical step in the simulation setup. Unlike the real setup, which keeps the specimen and gripper together due to friction and bolt preload, the simulation used 'Tie' constraints to ensure a fixed bond between the contacting surfaces, eliminating the possibility of separation or sliding that could occur in the actual experiment.

The simulation did not include any external loads, allowing the specimen to vibrate naturally and freely. Mesh generation was given special attention, particularly for the specimen, which required a fine mesh with an approximate element size of 0.00035 m to capture detailed dynamic responses. The mesh employed for the simulations was not regular, it consisted of tetrahedral elements, using a free meshing technique. This choice was pivotal in accurately capturing the complex geometries and dynamic responses of the specimen. The simulations were conducted using solid (3D) modelling techniques, with the finite element type selected as C3D10. The C3D10 tetrahedral element, which is recognised for its precision in 3D solid modelling, was important in the simulation. Its ten-node configuration enabled a detailed representation of the specimen's geometry, resulting in accurate stress and strain calculations that are critical for understanding the material's behaviour under cyclic loading conditions.

Subsequent iterations of the simulation process, involving repeated refinements of the part geometries, were essential for tuning the system to achieve the desired resonant frequencies and stress distributions.

It is important to understand that numerical models, particularly those dealing with high-frequency interactions and complex contact dynamics, inherently contain some degree of uncertainty. The simplifications used, such as the interactions mentioned before, may not fully capture the multifaceted realities of these systems. Consequently, experimental testing of the derived solutions is critical for validating the experimental setups and ensuring the reliability of the simulation results. This step is critical for closing the gap between theoretical predictions and practical experimental observations, increasing the credibility of the research findings.

4. RESULTS AND DISCUSSION

This chapter discusses the testing results for two sequential designs, called Design A and Design B. The second design iteration was required due to difficulties with the initial design that emerged throughout the experimental process. Because of these obstacles, the experimental design had to be modified to boost its effectiveness and dependability. This chapter carefully analyses numerical simulations and experimental data to provide insights into the capabilities and limitations of each design. By validating the results and highlighting potential improvements, this comparison study will contribute to the advancement of material fatigue testing methods.

4.1. Design A

This section presents the completed geometry of the components constituting Design A, which represents the outcome of iterative processes detailed in the preceding chapter. After several iterations and refinements, the geometry was finally optimised to allow for effective ultrasonic fatigue testing. The pictures and illustrations of each component are given below in Figure 4-1, Figure 4-2, Figure 4-3, and Figure 4-4, displaying their detailed geometry. These parts include the specimen, the bolt and nuts, the upper and lower grippers, and each one is produced to ensure precise mechanical wave transmission and reliable testing conditions.

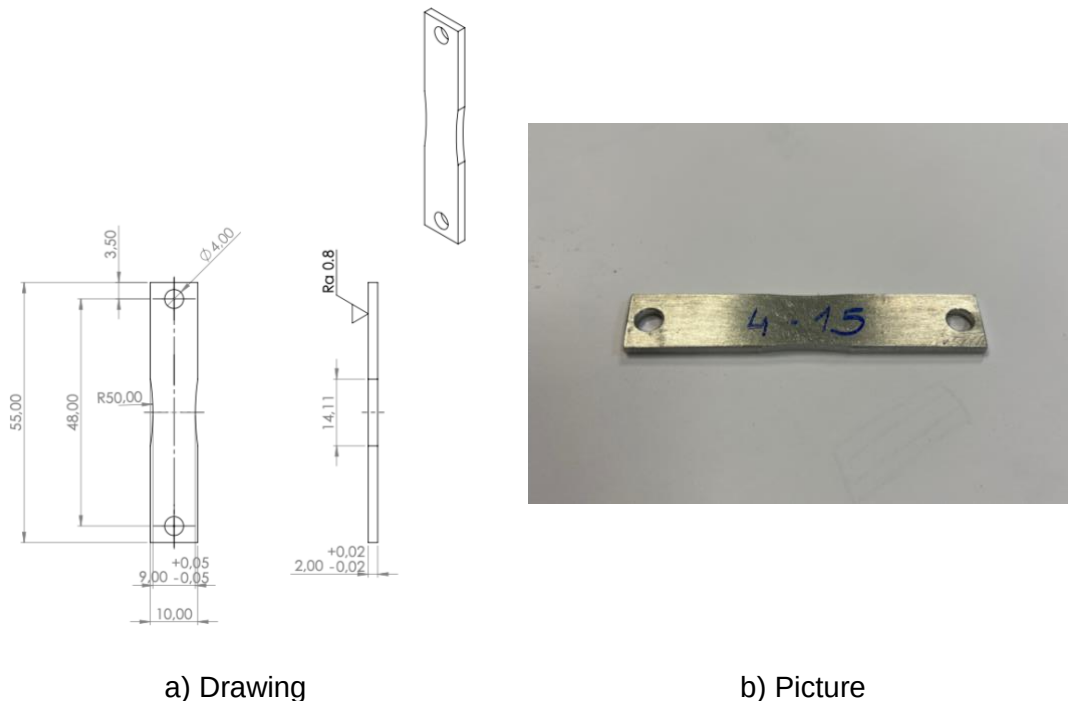
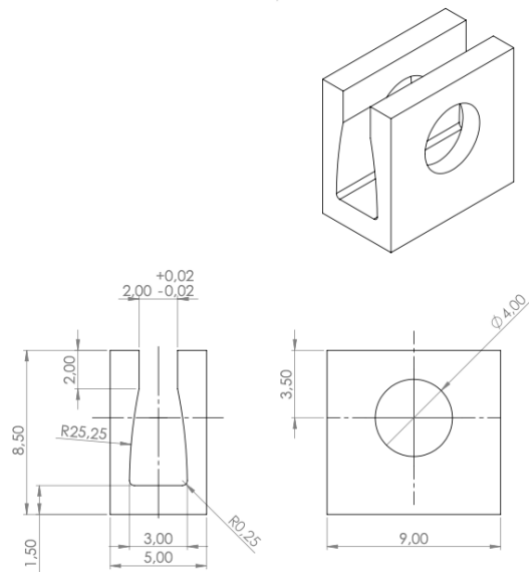


Figure 4-1 – Specimen.

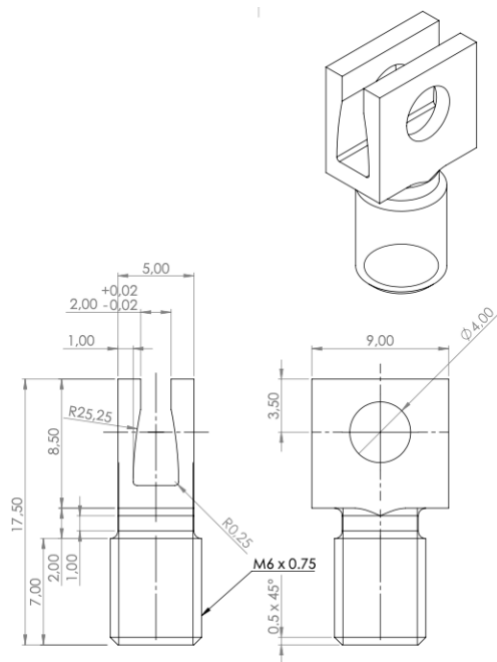


a) Drawing



b) Picture

Figure 4-2 – Lower gripper.

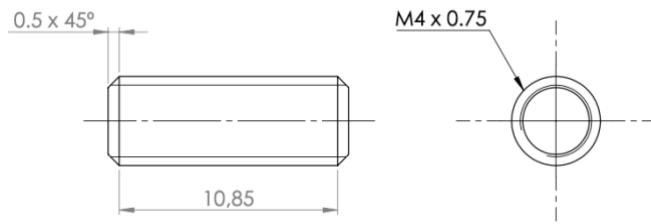


a) Drawing



b) Picture

Figure 4-3 – Upper gripper.



a) Drawing (bolt)



b) Picture

Figure 4-4 – Bolt and nuts.

Lastly, Figure 4-5 provides a detailed illustration of how every component was assembled and then properly attached to the testing apparatus.

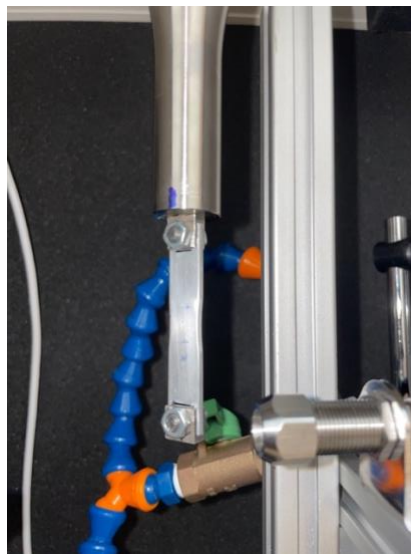


Figure 4-5 – Assembly of each part of Design A, attached to the machine.

4.1.1. Numerical results

This section undertakes a comprehensive modal analysis of the specimen from Design A using Abaqus software. The main objective is to gain useful insights into the specimen's dynamic response under various loading conditions. To lay the groundwork for the subsequent discussion of displacement and stress results, this section begins with an examination of the achieved frequency and stress levels using the modal analysis.

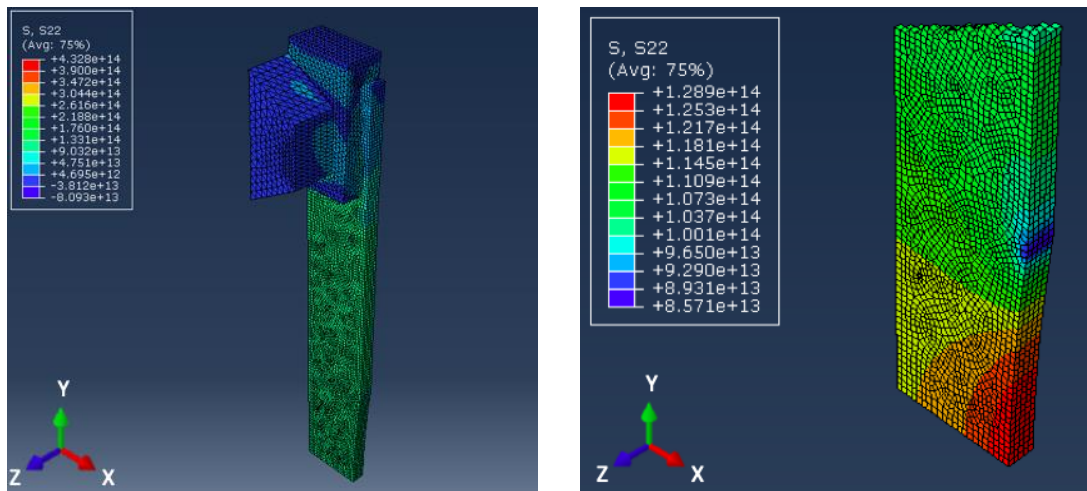
Once the geometry was defined, the numerical simulations achieved maximum stress levels suitable for fatigue testing and reached a frequency of 19.771 kHz. With a maximum displacement amplitude (A_0) of 55 μm , the calculated maximum stress (σ_{max}) came out to be 219.93 MPa. This value was calculated by

$$\sigma_{max} = \frac{A_0 \cdot \sigma_{sim}}{U_{sim}}, \quad (4-1)$$

with σ_{sim} and U_{sim} being the maximum values of stress and displacement of the simulation, respectively, retrieved from Abaqus.

Because Design A is symmetrical in all three axes, computational efficiency allows the Abaqus simulation to be run on only 1/8th of the specimen. However, a final, symmetry-free simulation was run to confirm that no other vibrational modes with frequencies similar to the required longitudinal deformation mode existed. These results were focused just on the specimen and removing the booster and horn from visualisation, as well as the restrictions between them. This was done to assure the accuracy of the results.

A clear pattern can be seen in the specimen's stress distribution, with the highest stress localised near the specimen's centre, as seen in Figure 4-6. This phenomenon results from the loading conditions, where positive stress is induced in the centre region of the specimen by tensile loading. Stress levels gradually decrease as the specimen's distance from the centre gets bigger and closer to its edges. The mechanics of the loading configuration, in which the applied force dissipates along the specimen's length and results in a declining stress gradient, are responsible for this difference in stress distribution.



a) Along all the specimen

b) Focused on the centre of the specimen

Figure 4-6 – Stress distribution along the specimen in axial direction (Pa).

In addition to stress distribution, the displacement results, displayed in Figure 4-7, provide further insights into the dynamic behaviour of the specimen under loading. Interestingly, near the mid-section of the specimen, where the stress is at its highest, the displacement is seen to be zero. This zero displacement represents the equilibrium point, or stationary condition, reached when the specimen's resistance equalises the applied stress. In addition, the displacement behaves symmetrically along the vertical axis, reflecting the pattern of stress distribution and emphasising the symmetrical character of the loading conditions.

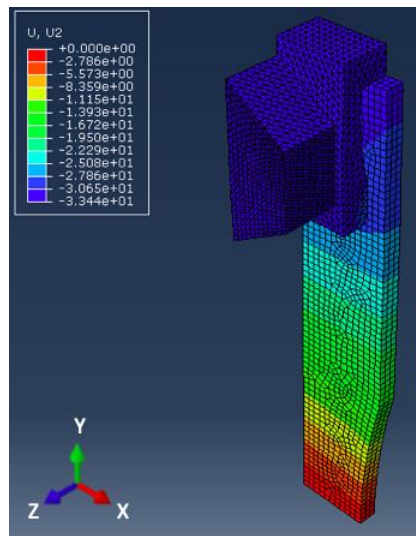


Figure 4-7 – Displacement distribution along the specimen in axial direction.

The machine models, when included in the simulation with the specimen, always bring the simulation results closer to the intended frequency. This is because the machine components account for the majority of the vibrating system's mass and are already dimensioned to achieve the desired value.

Evaluating the resulting vibrational mode's reliability to other alternate modes is crucial. The primary focus remains on verifying the existence of the specified longitudinal deformation mode and making sure that there are no highly conflicting modes. The Abaqus simulation confirms the required vibrational mode's existence within the specimen, as seen in Figure 4-8. The two vibrational modes closest to the target frequency are then displayed in Figure 4-9. Indeed, comprehensive numerical analysis confirms that these modes exhibit frequencies both considerably lower and higher, respectively, than the intended mode. This disparity supports Design A's ability to produce and maintain the intended longitudinal deformation mode while reducing the impact of possible interfering modes.

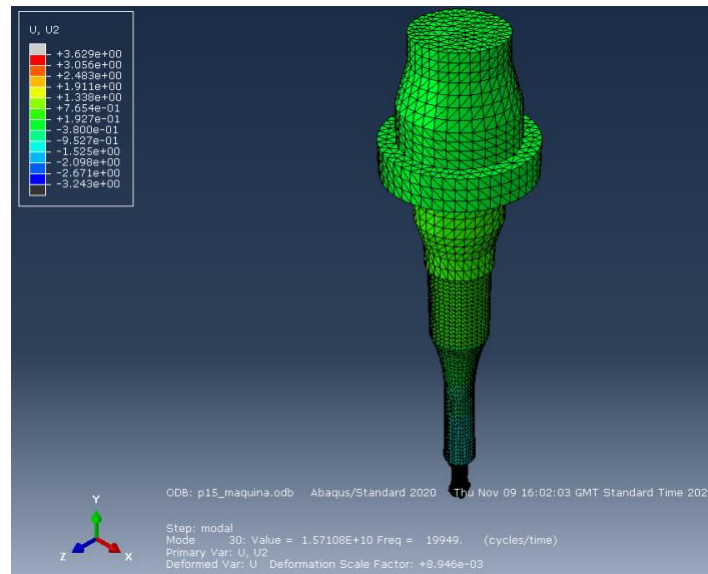
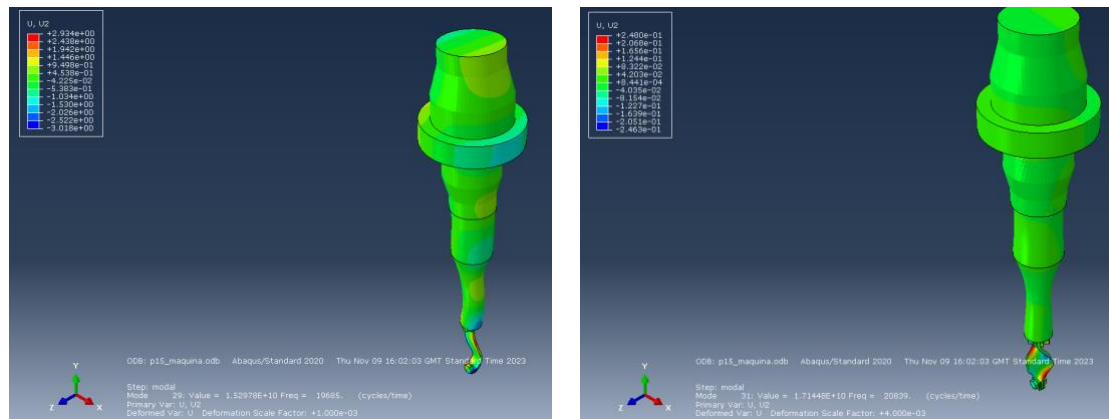


Figure 4-8 – Required vibrational mode with the necessary frequency (19.949 kHz).



a) Previous mode - bending (19.685 kHz) b) Posterior mode - torsion (20.839 kHz)

Figure 4-9 – Two vibrational modes closest to the target frequency.

4.1.2. Experimental results

During the experimental evaluation of Design A, significant challenges were encountered. The first tests were carried at a displacement amplitude of 20% of the maximum displacement, which is approximately 12 μm . The time-displacement graphic is presented in Figure 4-10. A vibration mode associated with 20 kHz displayed a significant magnitude, as shown in Figure 4-11. This frequency was measured through the graphic and its value was 19.818 kHz.

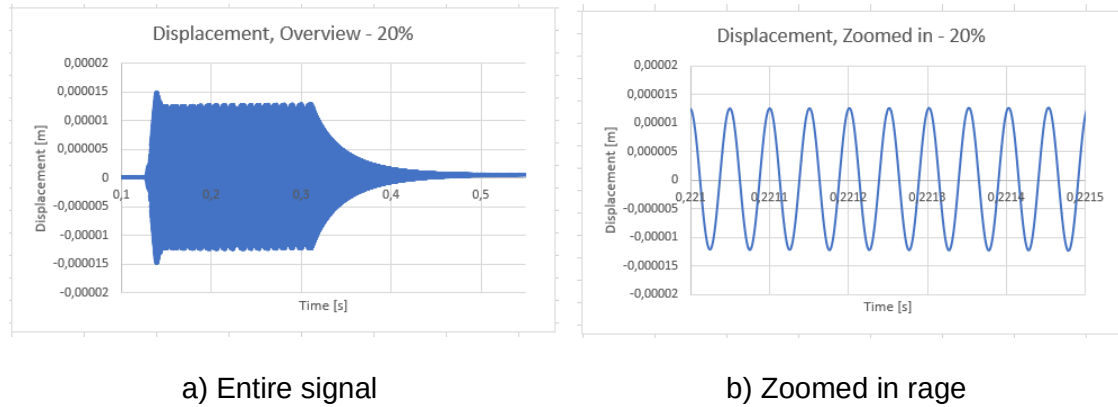


Figure 4-10 – Time-displacement chart for 20% of machine's maximum amplitude displacement (Design A).

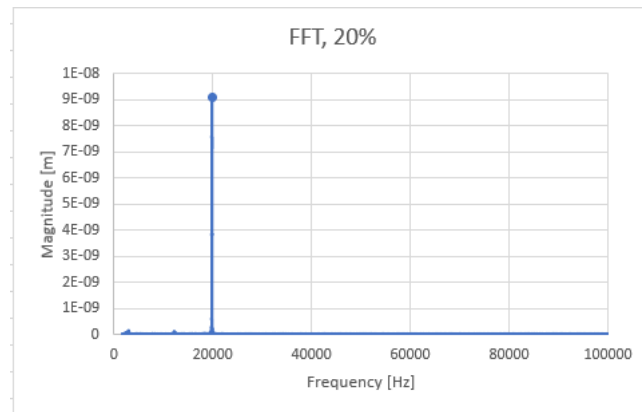


Figure 4-11 – FFT graphic for 20% of machine's maximum amplitude displacement (Design A).

The analysis of the time-displacement chart in Figure 4-10 shows a detailed visual representation of the specimen's response throughout the testing process. From initial activation to the establishment of a steady state, and finally to the cessation of vibrations, each phase of the graph provides information about the specimen's dynamic behaviour under test conditions. The initial segment of the graph represents the machine's activation phase, which rapidly progresses to the first pronounced peak, known as the overshoot. This overshoot is caused by the machine's control mechanism attempting to rapidly reach the set amplitude within a ramp time of only 10 ms, inadvertently exceeding the target amplitude momentarily. Following this transient phase, the graph stabilises into a constant section, which represents the permanent regime in which the signal stabilises and reflects the specimen's steady-state response under the specified testing conditions. The final section of the graph illustrates a gradual decline, representing the free vibration state after machine deactivation, in which the specimen continues to exhibit diminishing oscillations until it comes to rest.

Even though there were no problems with the first tests at 20%, an unexpected observation was made regarding the time-displacement graph generated during tests at a 40% displacement amplitude. When analysing a small range of time, two distinct sine wave patterns were observed, as can be seen in Figure 4-12, in contrast to the uniform sine wave observed at a 20% displacement amplitude. Although additional research is necessary to confirm the exact cause of this phenomenon, a few plausible causes could have contributed to the difference observed. The FFT analysis (Figure 4-13) suggests that there may be a second vibrational mode present, with a frequency of 9.902 kHz, measured from the graphic. It is possible that this additional mode's excitation affected the specimen's dynamic response, leading to the observed variation in sine wave patterns. Another plausible explanation could be related to the fastening system of the specimen and gripper. If the bolt fastening became loosened during testing, it could have introduced subtle shifts or misalignments in the specimen-gripper assembly, consequently impacting the dynamical behaviour of the system and contributing to the observed discrepancies in the sine wave patterns. To guarantee the consistency and reliability of experimental results in subsequent tests, more research and analysis are required to determine the precise factors causing the observed variation.

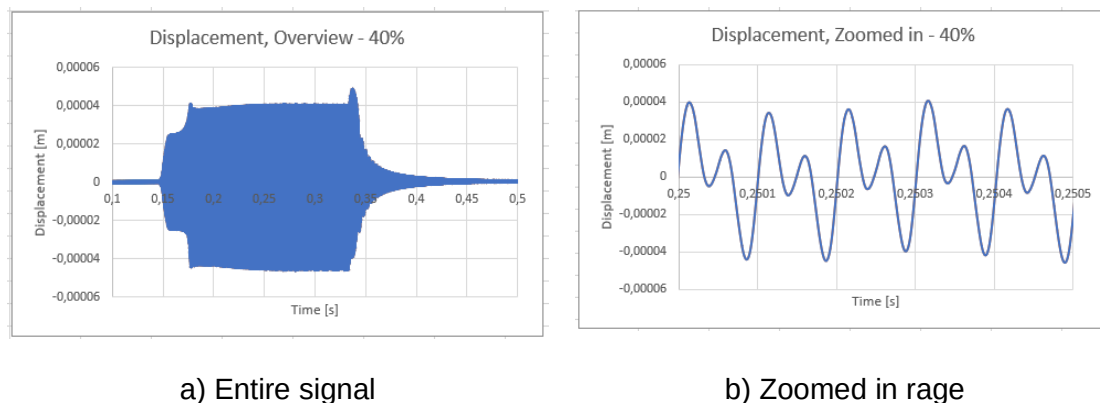


Figure 4-12 – Time-displacement chart for 40% of machine's maximum amplitude displacement (Design A).

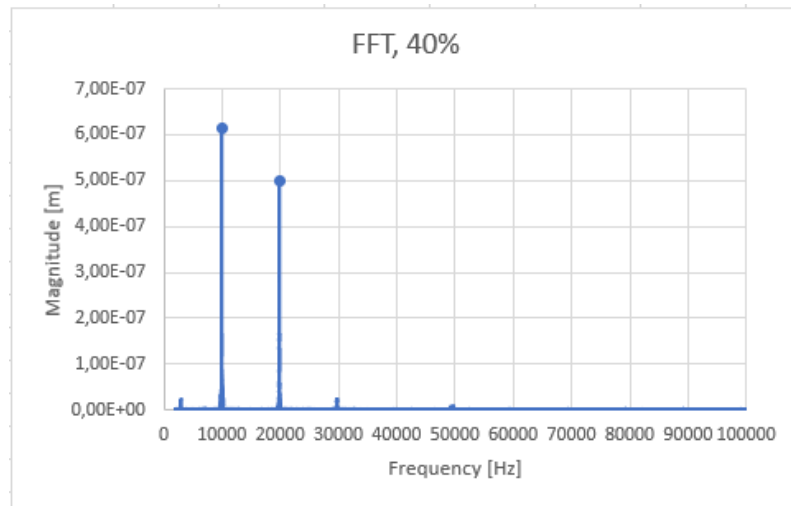
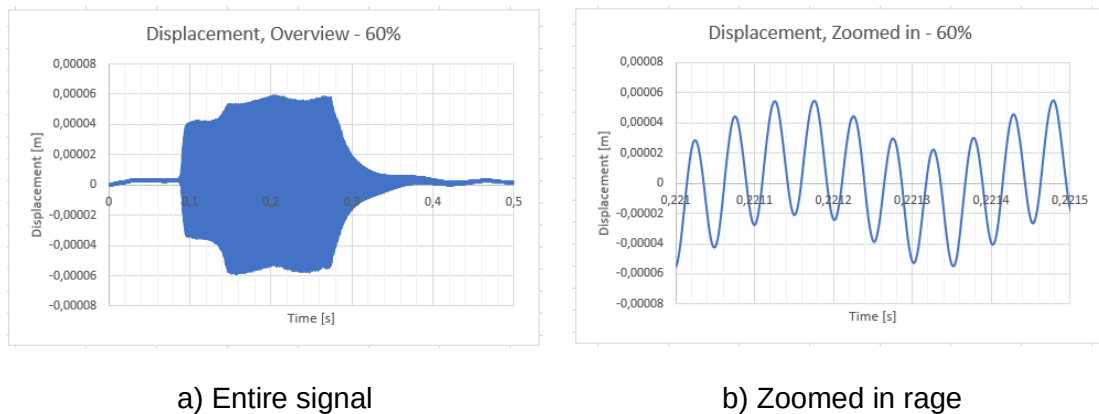


Figure 4-13 – FFT graphic for 40% of machine's maximum amplitude displacement (Design A).

An attempt was made to push testing to a 60% displacement amplitude before reevaluating at 40%. Unfortunately, the outcomes also did not meet expectations either. It is likely that another mode at approximately 2.5 kHz was excited, as can be seen in Figure 4-15, and interfered with the intended testing conditions. The displacement-time chart for the 60% testing, displayed in Figure 4-14, showed instability, which could have been caused by the bolts' looseness/gripper-specimen sliding and the referred excited mode as well.



a) Entire signal

b) Zoomed in rage

Figure 4-14 – Time-displacement chart for 60% of machine's maximum amplitude displacement (Design A).

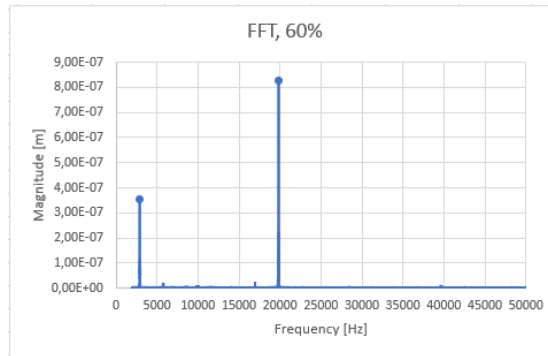
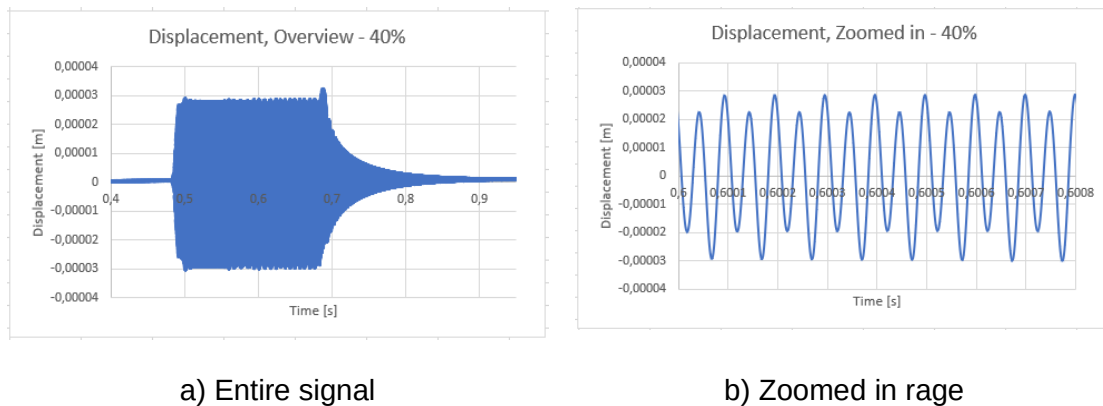


Figure 4-15 – FFT graphic for 60% of machine's maximum amplitude displacement (Design A).

After an attempt to push testing to a 60% displacement amplitude was not successful, another evaluation was performed at the previous 40% amplitude setting. The proper fastening of bolts with the intended torque specifications was again ensured before retesting. The results weren't entirely satisfactory even after these changes. Two distinct sine waves kept showing up in the time-displacement graph, Figure 4-16, even though the displacement values slightly improved from the first 40% test. In addition, there were no alterations found in the FFT analysis, Figure 4-13, suggesting that the extra vibrational mode at about 10 kHz was still present.



a) Entire signal

b) Zoomed in rage

Figure 4-16 – Time-displacement chart for 40% of machine's maximum amplitude displacement (Design A).

Despite the less-than-ideal indications from the previous tests, the decision was made to proceed with the VHCF test at 40% displacement amplitude. However, at approximately 10^6 cycles into the test, the gripper component experienced a critical failure, resulting in its fracture at the midpoint of the hole section, as seen in Figure 4-17. It is possible that insufficient torque resulted in a progressive loosening, as mentioned before, that aggravated stress concentrations and ultimately resulted in structural failure. The repetitive assembly and disassembly required in order to realign the grippers with

the specimen after each test probably took an impact in the mechanical fatigue and structural integrity degradation.

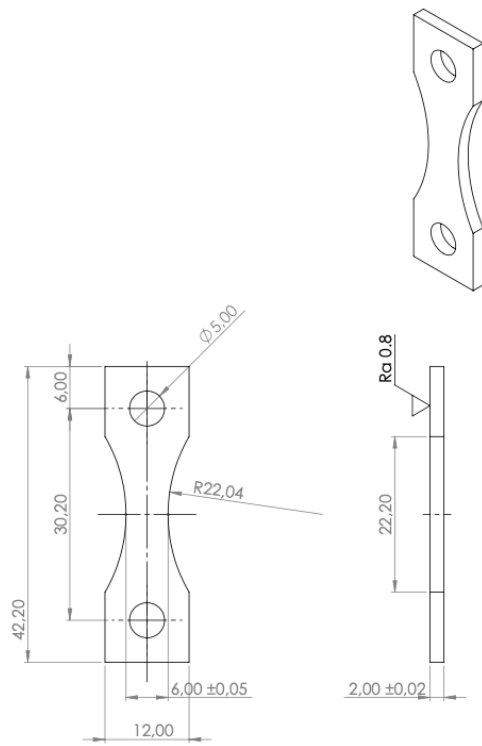


Figure 4-17 – Gripper after critical failure.

To address the shortcomings of Design A, a new geometry needed to be created for each component in response to these challenges. With the new design, the aim was to help ensure that the specimen could reach its maximum stress levels at a displacement amplitude of about 40% of the machine's capacity. By targeting this specific amplitude, it was intended to minimize the need for testing beyond 40%, reducing the risk of encountering the same issues of loosening bolts and interference from additional vibrational modes. This strategic approach aimed to simplify testing procedures, improve reliability, and improve the overall efficiency of the experimental setup. The details of this refined design will be covered in detail in the upcoming chapter, along with a review of how well it worked to get past the challenges that arose during the testing of Design A.

4.2. Design B

As mentioned above, the need to solve the issues that emerged during the Design A experiment led to the development of Design B. Several changes were implemented to improve the geometry and functionality of the components, since it was crucial to refine the experimental setup in order to guarantee reliable testing conditions. These changes will be discussed in this section and can be observed in Figure 4-18, Figure 4-19, Figure 4-20, and Figure 4-21.

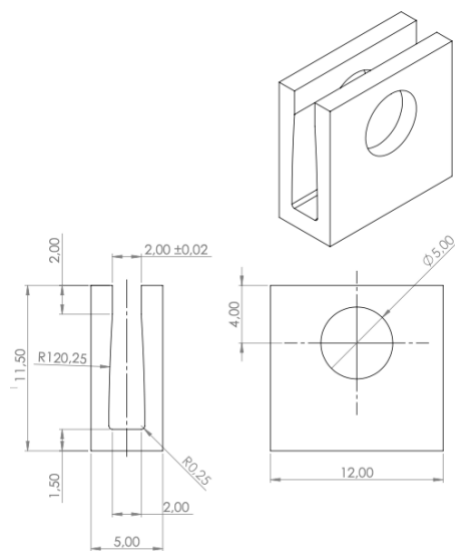


a) Drawing

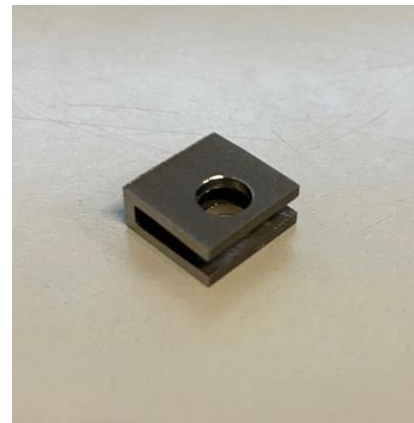


b) Picture

Figure 4-18 – Specimen.

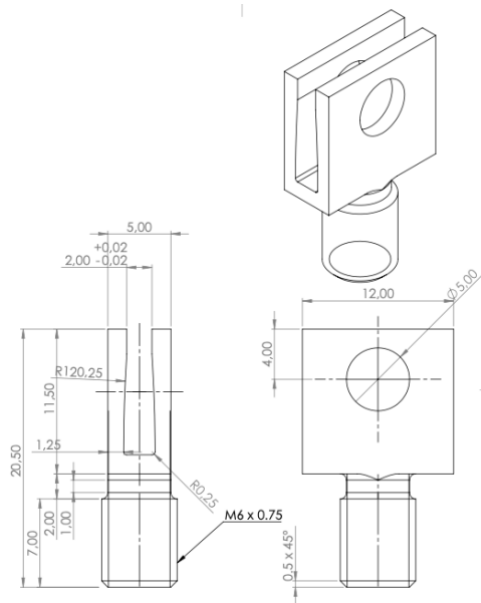


a) Drawing

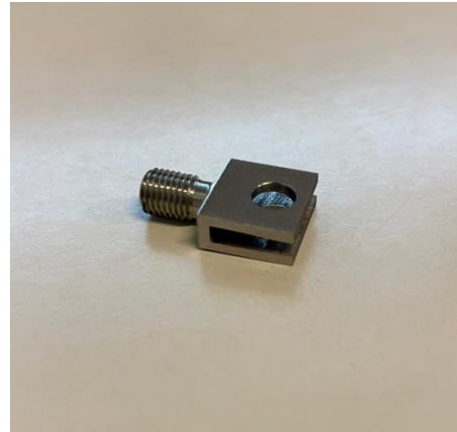


b) Picture

Figure 4-19 – Lower gripper.

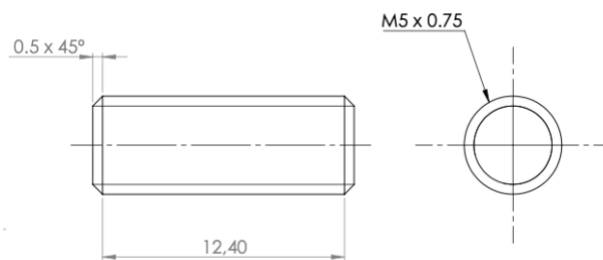


a) Drawing



b) Picture

Figure 4-20 – Upper gripper.



a) Drawing



b) Picture

Figure 4-21 – Bolt and nuts.

The transition from an M4 to an M5 bolt and nuts configuration was one of the main changes implemented in Design B. This modification was made in order to minimise the possibility of specimen loosening during testing, by allowing higher torque levels to be applied during bolt fastening. Additionally, the switch to M5 hardware helped to maintain testing conditions constant by minimising misalignment between the specimen and the grippers.

The revision of the gripper component in Design B was possibly the biggest improvement. Recognising the flaws of the previous gripper design, in particular its fragility, Design B included major improvements to increase durability and reliability.

Improved and strengthened, the gripper has more thickness to endure the demanding ultrasonic fatigue testing.

Alongside the gripper improvement, the specimen was significantly redesigned to maximise its functionality in the updated experimental setup. To fit the dimensions of the modified grippers, the specimen's width was increased and its length was decreased. To overcome the limitations of the previous design iteration, the width variation was also greatly increased to enable greater stress levels during testing, as the next section will attest.

Specific attention was paid to preserving the intended operating frequency of 20 kHz, while concurrently increasing stress levels, to solve the previous problems during the iterative refining process that resulted in Design B. The application of an iterative technique ensured that the final design achieved a balance between stress-induced boosts in performance and frequency requirements.

Finally, Figure 4-22 provides a representation of the assembly system, showing the integration of each component of Design B onto the testing apparatus.



Figure 4-22 – Assembly of each part of Design B, attached to the machine.

4.2.1. Numerical results

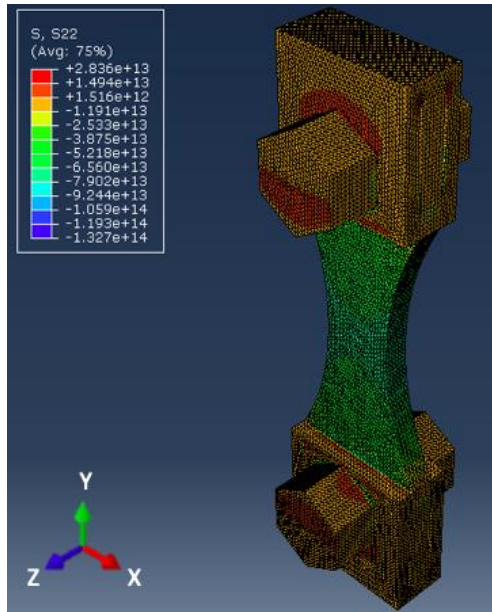
The numerical simulations performed for Design B provide an extensive analysis of the specimen's dynamic response under varying loading conditions, representing a significant refinement over the insights gathered from Design A. Design B's main goal was to optimise stress levels while maintaining operational integrity and

staying within the limits of the material, shown in previous sections. This required a careful balance between vibrational characteristics and mechanical stability.

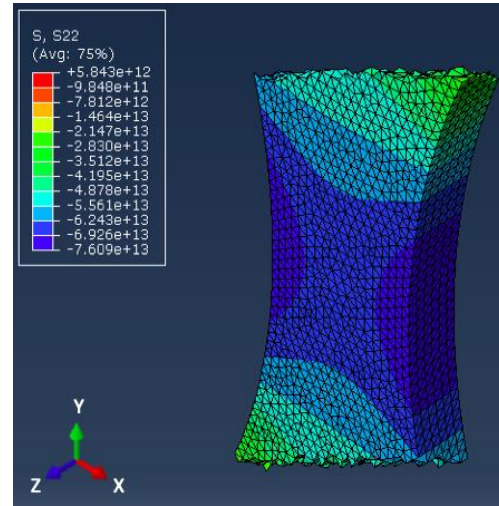
Design B achieved a frequency of 19,900 Hz. Considering a maximum displacement amplitude of 40% of the machine's capacity, or 22 μm , the calculated maximum stress reached 175.80 MPa, quite close to the goals that were previously discussed. Equation (4-1) was also used to calculate this value. The difficulties encountered during earlier experimental testing led to the decision to design for a maximum stress at a displacement amplitude of 40% of the machine's capacity. Testing amplitudes greater than 40% of the machine's capacity were found to cause problems to the original design, which could have resulted in risks and issues with the experimental setup. Therefore, in order to reduce the need to subject the specimen to higher displacement amplitudes and the likelihood of running into similar issues during testing, Design B was developed with the intention of achieving high stress levels at this amplitude threshold. Developing the specimen to reach these stress levels for 40% of the machine's capacity was essential to deal with the issues that emerged in the previous experimental testing.

As was the case with Design A, specimen's symmetry allowed simulations to be run on just 1/8th of the specimen. A complete simulation of the whole geometry was then run in order to confirm the initial results and guarantee there was no vibrational mode with a frequency close to the one desired.

The specimen's stress distribution matched up to that of Design A, but with noticeably higher stress levels, indicating that the refinement efforts to increase stress magnitudes were successful. At the same time, the displacement distribution showed patterns similar to those seen in Design A, yet with slightly lower displacement values. A visual representation of the stress distribution and displacement characteristics will be shown in Figure 4-23 and Figure 4-24, respectively, that provide a visual interpretation of the numerical results.

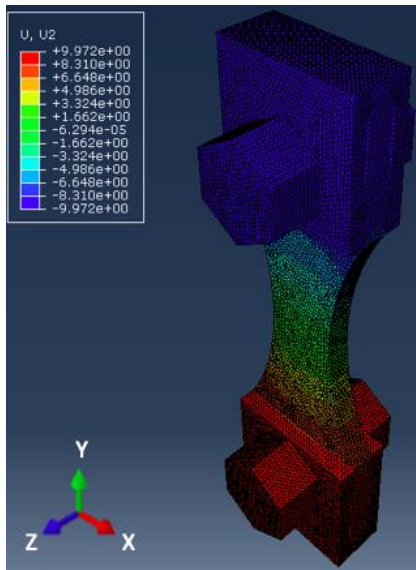


a) Along all the specimen

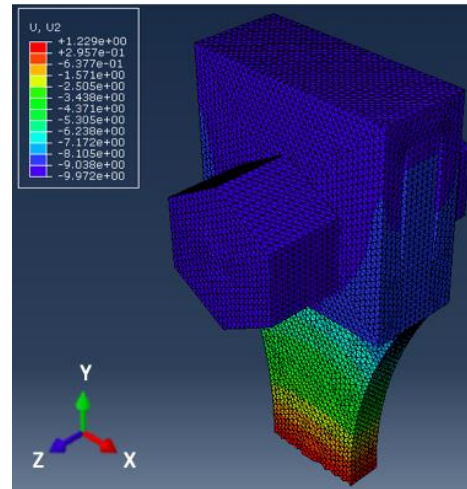


b) Focused on the centre of the specimen

Figure 4-23 – Stress distribution along the specimen.



a) Along all the specimen



b) Focused on the centre of the specimen

Figure 4-24 – Displacement distribution along the specimen.

Additionally, an assessment of the potential conflicting vibrational modes confirms that Design B is effective in producing and maintaining the desired longitudinal deformation mode, as no conflicting modes have been identified near the target frequency. These results show Design B's improvements by illustrating how it can maximise stress levels while preserving operational robustness and fidelity to its intended vibrational characteristics. A comprehensive evaluation of Design B's

vibrational modes is made possible by the provided figures, labelled as Figure 4-25 and Figure 4-26, which offer a visual representation of the simulated vibrational modes, including the intended longitudinal deformation mode and the nearest vibrational modes.

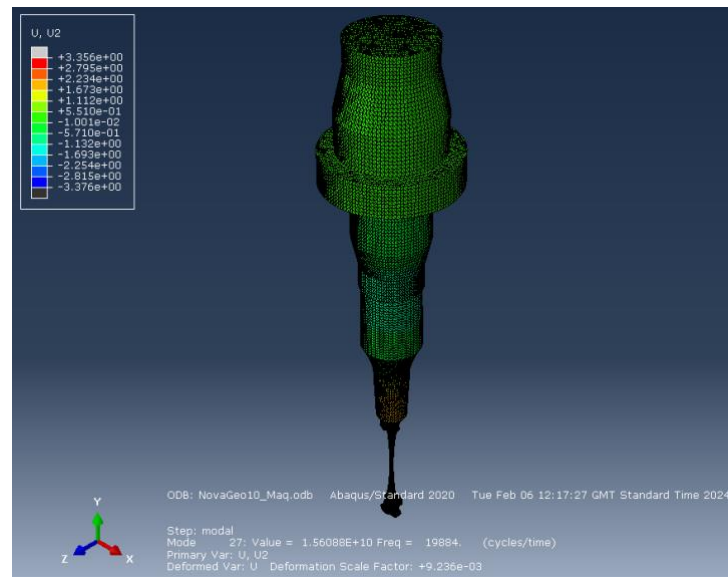
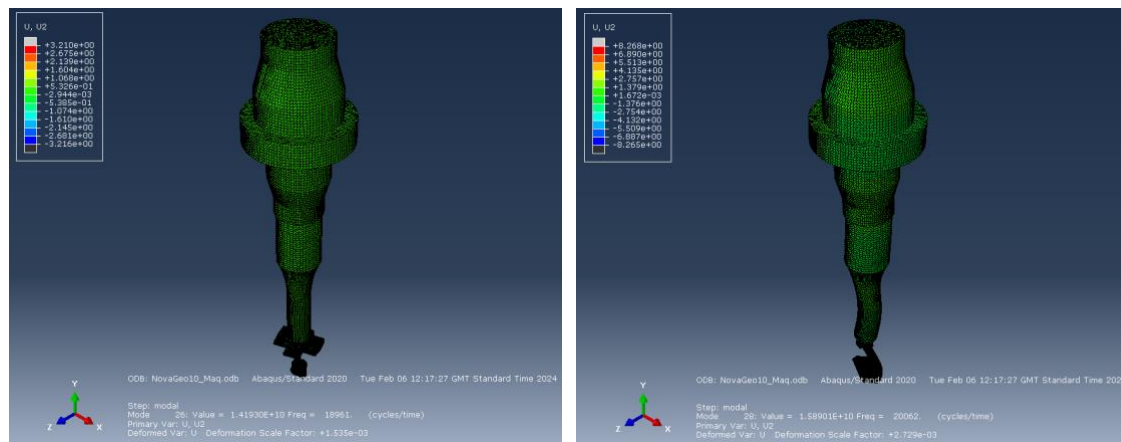


Figure 4-25 – Required vibrational mode with the necessary frequency (19.884 kHz).



a) Previous mode - torsion (18.961 kHz)

b) Posterior mode - bending (20.062 kHz)

Figure 4-26 – Two vibrational modes closest to the target frequency.

It is worth noting that the machine's oscillation frequency, in the absence of any specimen, is 19.870 kHz, as mentioned in Section 3.1.1. This value is remarkably close to the frequencies associated with the modes discussed in this chapter, highlighting the experimental setup's complexities. The proximity of this oscillation frequency to the frequencies of the various vibrational modes, such as bending and torsion, reveals the delicate balance required to ensure that the desired longitudinal mode is predominantly

excited during testing. This closeness in frequency values emphasises the difficulty of maintaining precise control over the experimental conditions, as even minor deviations in specimen geometry or assembly integrity (e.g., due to bolt placement or gripper loosening) might inadvertently excite these alternative modes.

4.2.2. Experimental results

Design B's experimental investigation consisted of a systematic series of tests performed on three similar specimens to assess their behaviour under varying stress amplitudes and frequencies. Unlike Design A, Design B sought to improve on previous challenges by using a more refined approach to testing and design modification. This section describes the procedures, observations, and results of the experimental evaluation of Design B, divided into three sections for each specimen tested.

4.2.2.1. Specimen 1

The initial examination of Specimen 1 started with an automatic test at a 15% displacement amplitude upon machine activation, which is a standard procedure for ensuring system readiness. Following this preliminary check, a calibration test was performed at a 20% displacement amplitude, producing results within expected parameters and displaying a clear signal and the anticipated amplitude in the time-displacement graphics, as shown in Figure 4-27. The calibration for the 20% test was characterised by a frequency of 19.540 kHz.

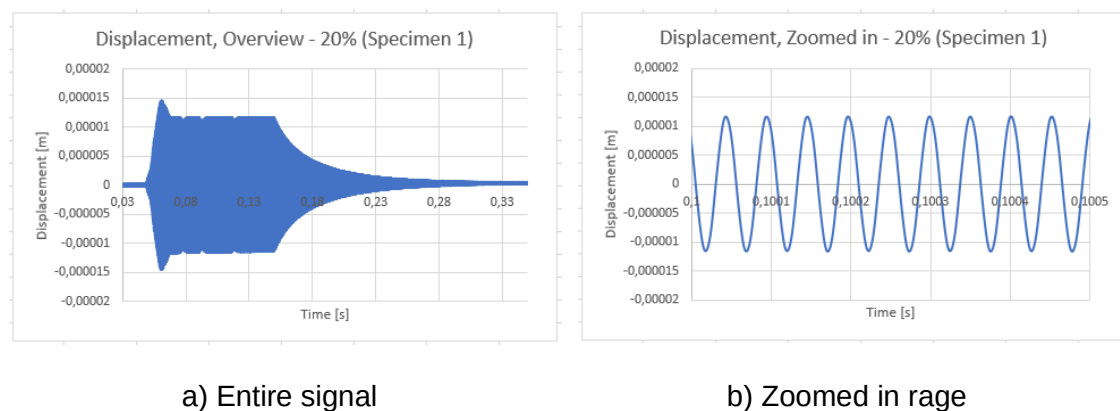


Figure 4-27 – Time-displacement chart for 20% of machine's maximum amplitude displacement for Specimen 1 (Design B).

As the displacement amplitude was increased to 40%, Specimen 1 began to exhibit undesirable behaviours such as twisting along the machine's axis and the

emergence of an additional bending vibrational mode. This mode, which may be a recurrence of previous observations in Design A, had an impact on the specimen's performance, as shown in the time-displacement graphics for the 40% displacement test in Figure 4-28, where its signal is not the desirable one.

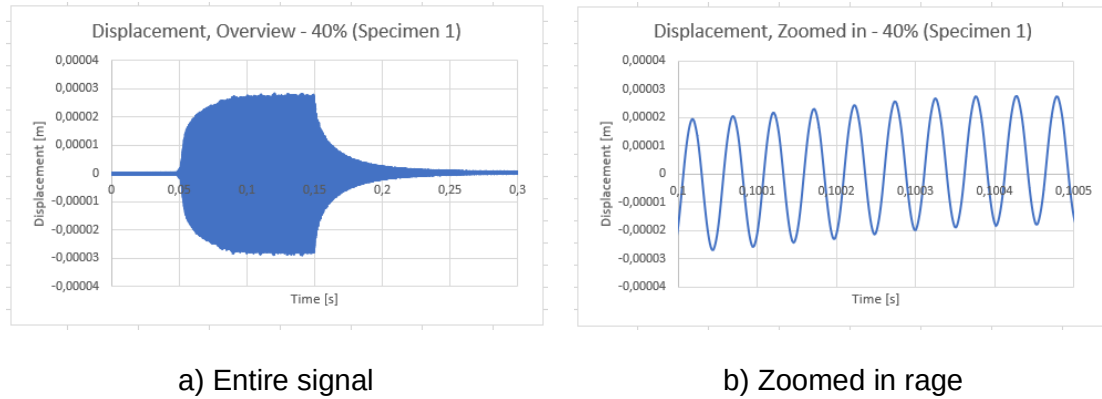


Figure 4-28 – Time-displacement chart for 40% of machine's maximum amplitude displacement for Specimen 1 (Design B).

Adjusting the test parameters to find a balance between stress amplitude and vibrational stability, a nominal stress amplitude of 140 MPa was selected after several trials. This level was determined to be optimal for avoiding the excitation of adverse vibrational modes while still imposing significant stress on the specimen. A VHCF test at this amplitude was then carried out. Calibration data for this test showed an output amplitude of 19.99% corresponding to a horn side amplitude of 11.52 μm and, at 39.63% output amplitude, the horn side amplitude was 24.18 μm . These values were retained by the system during the calibration tests done at 20 and 40%, mentioned before. The machine's oscillation time was set to 190 ms and the stop time to 650 ms to prevent specimen overheating, a precautionary measure necessary to maintain the specimen's structural integrity during the extended testing period. After approximately 200,000 cycles, the fracture of the specimen at its centre occurred, as displayed in Figure 4-29. This cracking is a promising indicator, suggesting that the specimen collapsed precisely at the zone of greatest stress concentration, which is consistent with theoretical stress distribution predictions under the loading conditions applied. This result not only validates the specimen's design and material behaviour under high-cycle fatigue, but it also demonstrates the experimental setup's accuracy in simulating the intended stress environment.



Figure 4-29 – Specimen 1 after the cracking.

4.2.2.2. Specimen 2

Specimen 2 underwent an initial calibration test at a displacement amplitude of 20%, mirroring the results obtained with Specimen 1, with a frequency of 19.600 kHz. The calibration data from this test revealed an output amplitude of 19.95%, resulting in a horn side amplitude of $10.36\ \mu\text{m}$. As there was no intention to submit this specimen to the high stresses the first one was subjected to, maybe resulting in the premature failure of the specimen, the values of the horn side amplitude for 40% were iterated at $20.77\ \mu\text{m}$.

Following the results from Specimen 1, the testing amplitude for Specimen 2 was cautiously set to 130 MPa, to prevent the problems found when testing the specimen at higher stress amplitudes. At this amplitude, the previously mentioned undesirable bending mode became significantly excited, resulting in noticeable bending deformation in the specimen's thickness direction. This behaviour, illustrated in Figure 4-30, demonstrated the specimens' sensitivity to specific vibrational modes under stress.

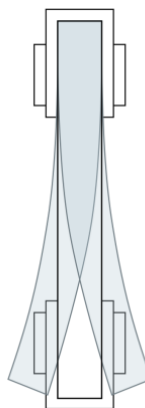


Figure 4-30 – Illustration of bending of specimen.

4.2.2.3. Specimen 3

Following the experimental results from Specimens 1 and 2, Specimen 3 was strategically assigned for immediate VHCF testing at a predetermined stress amplitude of 130 MPa. This approach was carefully chosen to ensure that Specimen 3 was not subjected to any wear or pre-test fatigue that could affect its performance, preserving its original condition for a more accurate assessment of its inherent material properties under VHCF conditions. A brief calibration test was performed at a 20% displacement amplitude to determine the horn side amplitude, which is critical for correlating input energy to the specimen's mechanical response. The calibration resulted in a horn side amplitude of 10.96 μm at 20.14% output amplitude. After iterating at 40% output amplitude, the horn side amplitude was calculated as 21.77 μm . The test frequency was held at 19.560 kHz. To reduce the risk of thermal accumulation and potential overheating, the machine's oscillation time was again set to 190 ms and the stop time to 650 ms. The results of Specimen 3's VHCF testing were notably consistent and promising, as shown in the time-displacement chart in Figure 4-31. This specimen failed after 4.28 million cycles. Surprisingly, the fracture occurred in the area near the bottom gripper, rather than the anticipated high-stress zone, as observed in the Figure 4-32.

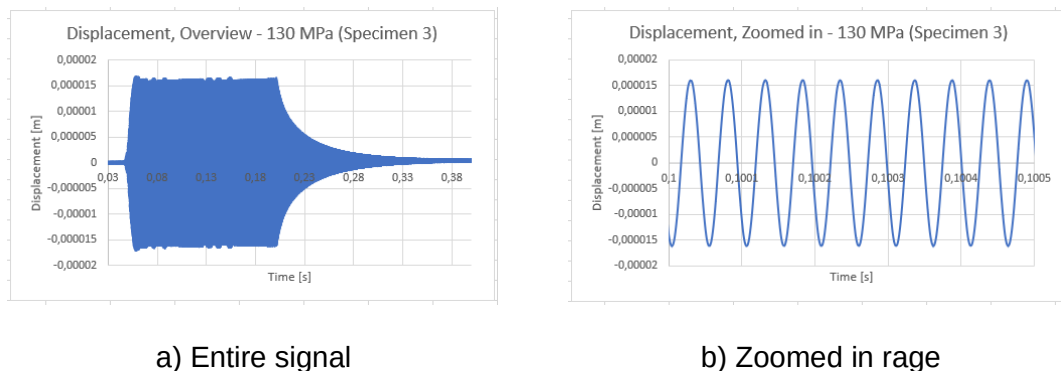


Figure 4-31 – Time-displacement chart for a stress amplitude of 130 MPa for Specimen 3 (Design B).



Figure 4-32 – Specimen 3 after the fracture.

The unexpected fracture location near the bottom gripper could be attributed not only to the influence of the persistent bending vibrational mode, which was previously observed during the testing of Specimens 1 and 2, but also to the possible involvement of a fretting-fatigue process. Fretting, caused by micro-movements at the contact surfaces, particularly where the gripper secures the specimen, can increase stress concentrations, boosting the specimen's exposure to failure. This interaction most likely increased the localised stress, which contributed to the premature fracture in this unintentional zone. The convergence of these factors - vibrational mode excitation and fretting fatigue - highlights the importance of attention to specimen mounting and interface conditions in high-frequency fatigue testing.

Table 4-1 summarises the experimental conditions and results for each specimen, providing a clear overview of the testing process and the challenges encountered, particularly the excitation of undesirable vibrational modes and the discrepancies between expected and actual fracture locations.

Table 4-1 – Summary table of the trials for each specimen.

Specimen	Calibration Test Frequency (kHz)	Final Test Stress Amplitude (MPa)	Horn Side Amplitude for 20% Output (μm)	Horn Side Amplitude for 40% Output (μm)	Number of Cycles to Failure	Observations
1	19.540	140	11.52	24.18	~200,000	Fracture at the specimen's centre, indicating stress concentration at intended zone.
2	19.600	130	10.36	20.77 (iterated)	-	Notable bending in the specimen's thickness direction due to the excitation of a bending vibrational mode.
3	19.560	130	10.96	21.77 (iterated)	4.28 million	Fracture occurred near the bottom gripper, likely due to the influence of a bending vibrational mode and potential fretting-fatigue processes.

5. CONCLUSIONS AND FUTURE WORK

5.1. Conclusions

To conclude the dissertation, the refined experimental setup, which was an improvement from the original design, proved to be successful for one out of the three specimens tested, highlighting the obstacles and susceptibility to inconsistencies in VHCF testing of thin-plate specimens. Those results also demonstrated the importance of precision in every step of the testing process, including experimental design, simulation, and careful specimen fabrication.

The refinement of the design focused mainly on increasing stress concentrations in the critical section of the specimen for the same amplitude displacement, a key objective that was met with different levels of success across the specimens. This effort was essential aimed to ensure reliability of test results and push the limits of understanding material behaviour under extreme cyclic loading.

An important outcome of this study was the ease with which other vibrational modes could be inadvertently excited due to a variety of uncontrollable minor factors, such as the specimen's absolute symmetry being compromised by bolt placements or the possibility of looseness between the gripper and specimen. As mentioned, these factors may result in the excitation of other modes, as bending and torsion, with frequencies that are very close to the desired longitudinal mode observed in simulation. The close proximity of these vibrational modes in frequency highlights the delicate nature of the experimental setup, where even minor unintended changes in geometry can cause significant deviations in results.

Furthermore, the differences between simulations and experimental setups demonstrated the importance of experimental validation on material testing. For example, the simulated ties between the gripper and the specimen, which represented idealised constraints, were unable to accurately represent the complexities of physical interactions in the real-world context, where friction and the gripping mechanism play critical roles. This discrepancy was particularly evident in the possibility of the specimen sliding or loosening within the grip, challenging the experimental results.

To summarise, this thesis not only provides an understanding of the complex dynamics of VHCF testing of thin-plate specimens, but it also serves as a cautionary narrative for future research in this field. The findings highlight the importance of a rigorous, detail-oriented approach to experimental design and execution, in which every minor detail, from specimen fabrication to setup alignment, can have a significant impact on the validity and reliability of the results.

5.2. Future work

This master's dissertation represents the culmination of a thorough investigation into the VHCF behaviour of the AlSi10MnMg alloy, and there are several intriguing areas for future research that might further develop our understanding of materials under cyclic loading. The tasks described below are advised to improve the findings and investigate the broader landscape of VHCF.

- Focus on refining the experimental setup and conducting a detailed characterization of the AlSi10MnMg alloy to learn about its behaviour under cyclic loading.
- Extend the experimental programme to evaluate the AlSi10MnMg alloy at frequencies other than 20 kHz. Use machinery that can accommodate different frequencies to identify frequency-dependent differences in VHCF behaviour. This investigation will yield vital information into the material's behaviour to various vibrational situations.
- Diversify the scope of study by evaluating additional alloys at the defined 20 kHz frequency. The inclusion of different alloys will allow for comparison investigations, revealing how VHCF behaviour varies among materials. Consider alloys with different compositions and mechanical qualities to gain a better knowledge of their performance in the ultra-high cycle regime.
- Conduct fatigue testing at 20 kHz frequency with varying load ratios. Systematically investigate the effect of different load conditions on the VHCF behaviour of the AlSi10MnMg alloy. This will provide critical insights into how load ratios affect fatigue life, crack initiation, and propagation, allowing for a more complete understanding of the material's mechanical reaction.

6. BIBLIOGRAPHY

- Altan, T., & Saan, G. E. D. (2008). Advanced High Pressure Die Casting Processes. *CIRP Annals - Manufacturing Technology*, 57(2), 583-604.
- ASTM International. (2013). *ASTM 1823-2013: Standard Terminology Relating to Fatigue and Fracture Testing* American Society for Testing and Materials, West Conshohocken, PA. Retrieved from <https://www.astm.org/e1823-13.html>
- Bathias, C. (1999). There is no infinite fatigue life in metallic materials. *Fatigue & Fracture of Engineering Materials & Structures. The International Journal for Structural Integrity*, 22(7), 559-565.
- Bathias, C. (2012). Gigacycle fatigue of bearing steels. *Materials Science and Technology*, 28(1), 27-33.
- Bathias, C. (2014). Fatigue Limit in Metals. In *Focus Series*. London, England: ISTE Ltd.
- Bathias, C., & Paris, C. (2004). Gigacycle fatigue in mechanical practice. *CRC Press*.
- Brugger, C., Palin-Luc, T., Osmond, P., & Blanc, M. (2016). Gigacycle fatigue behavior of a cast aluminum alloy under biaxial bending: experiments with a new piezoelectric fatigue testing device. *21st European Conference on Fracture*, 1174-1180.
- Budynas, G. R., & Nisbett, K. J. (2011). Shigley's mechanical engineering design, volume 9. Retrieved from https://www.academia.edu/37397805/Shigleys_Mechanical_Engineering_Design_9th_Edition
- Budynas, R., & Nisbett, K. (2008). *Shigley's Mechanical Engineering Design* (Vol. 10): McGraw-Hill.
- Callister, W. D., & Rethwisch, D. G. (2018). *Materials Science and Engineering: An Introduction*. New York: Wiley.
- Chegg Inc. (2015). Re: QUESTION 4 a) The diagram below shows S-N curves for three materials at a stress ratio $R = -1$. State whether each material would fail or be safe if you were to apply one million cycles at a stress amplitude of 200 MPa. Message posted to

<https://www.chegg.com/homework-help/questions-and-answers/question-4-diagram-shows-s-n-curves-three-materials-stress-ratio-r-1-state-whether-material-q7693171>

- Costa, P., Nwawe, R., Soares, H., Reis, L., Freitas, M., Chen, Y., & Montalvão, D. (2020). Review of Multiaxial Testing for Very High Cycle Fatigue: From 'Conventional' to Ultrasonic Machines *Máquinas* 2020, 8(2), 25. Retrieved from <https://doi.org/10.3390/machines8020025>
- Ebara, R., & Yamada, Y. (1987). Ultrasonic corrosion fatigue testing of 13 Cr stainless steel and Ti-6Al-4V alloys in Ultrasonic Technology. *Tokyo: MYU Research*.
- Fatemi, A. (2016). *Fatigue of Materials: Advances and Emergences in Understanding*. Springer.
- Fiorentin, F. K. (2020). *A Numerical and Experimental Study on Residual Stresses, Distortion and Fatigue Behaviour of Additive Manufactured Components*. (PhD thesis). Inegi, 2020,
- Fiorentin, F. K. (2023). *High-Cycle and Ultra-High Cycle Fatigue Behaviour of Directed Energy Deposited Inconel 625*. (PhD thesis). Retrieved from <https://hdl.handle.net/10216/149204>
- Furuya, Y., Matsuoka, S., Abe, T., & Yamaguchi, K. (2002). Gigacycle fatigue properties for high-strength low-alloy steel at 100 Hz, 600 Hz, and 20 kHz. *Scripta Materialia*, vol. 46, 157-162.
- Gouveia, M. (2022). *Giga-cycle Fatigue Behaviour of the Structural Steel S690*. (Master dissertation). Retrieved from <https://hdl.handle.net/10216/143741>
- Kolsky, H. (1964). Stress waves in solids. *Journal of Sound and Vibration*, 1(1), 88–110.
- Liu, X. (2019). *Predicting Fatigue Life of Welded Joints under Variable Amplitude Loading Spectra*. (Ph.D.). Brunel University London, London.
- Makhlouf, M., & Ali, S. (2017). *High Pressure Die Casting: Machines and Products*: Springer.
- Mirzadeh, H., & Razaghian, A. (2020). High-Cycle Fatigue Behavior of Aluminum Alloys: A Review. *Journal of Materials Engineering and Performance*, 29(10), 6127-6141.
- Mughrabi, H. (2006). Specific features and mechanisms of fatigue in the ultrahigh-cycle regime. *International Journal of Fatigue*, 28(11), 1501-1508.

- Murakami, Y. (2019). Metal fatigue: effects of small defects and nonmetallic inclusions (2nd edition). *Academic Press*.
- Murakami, Y., Nomoto, T., & Ueda, T. (1999). Factors influencing the mechanism of superlong fatigue failure in steels. *Fatigue & Fracture of Engineering Materials & Structures*, vol. 22, 581-590.
- Pacheco, F., M. (2016). *Projeto de máquina ultrassônica para testes em aços submetidos à fadiga giga-ciclo*. (Master's thesis). Retrieved from <https://www.maxwell.vrac.puc-rio.br/colecao.php?strSecao=resultado&nrSeq=30934@1>
- Pawar, R. P., & Choudhary, A. (2014). Process Parameters Optimization of HPDC for Thin Walled Casting. *Procedia Materials Science*, 6, 1355-1361.
- Peliteiro, D. (2023). *Giga-Cycle Fatigue Behavior of high pressure die-casting aluminum alloys*. (Master dissertation).
- Phung, N., Favier, V., Ranc, N., Valès, F., & Mughrabi, H. (2014). Very high cycle fatigue of copper: Evolution, morphology and locations. *International Journal of Fatigue*, vol. 63, 68-77.
- Polytec. (2019). *Polytec VibroOne Laser Vibrometer Data Sheet*. Retrieved from <https://www.atecorp.com/atecorp/media/pdfs/data-sheets/polytec-vibroflex-connect-laser-vibrometer-data-sheet.pdf?ext=.pdf>
- Pyttel, B., Schwerdt, D., & Berger, C. (2011). Very high cycle fatigue – is there a fatigue limit? *International Journal of Fatigue*, 33(1), 49–58. Retrieved from <https://www.sciencedirect.com/science/article/abs/pii/S0142112310001362>
- Rheinfelden. (2013). Rheinfelden Alloys. Retrieved from https://alumag.com/wp-content/uploads/2021/05/RHEINFELDEN-ALLOYS_Details-fur-den-Konstrukteur-uber-die-Duktilen-Aluminium-Druckgusslegierungen Sf-36 Ci-37 Ma-59.pdf
- Rheinfelden. (s.d). Alloys. *Silafont*. Retrieved from <https://rheinfelden-alloys.eu/en/alloys/silafont/>
- Song, D., Zhu, S., Lu, L., & Lu, K. (2018). Very high cycle fatigue properties of high-strength steels with various microstructures. *International Journal of Fatigue*, 106, 136-145. doi:10.1016/j.ijfatigue.2017.09.020

- Stanzl-Tschegg, S. (2014). Very high cycle fatigue measuring techniques. *International Journal of Fatigue*, 60, 2–17.
- Stanzl-Tschegg, S., Mayer, H., & Stich, A. (2002). Variable amplitude loading in the very high-cycle fatigue regime. *Fatigue & Fracture of Engineering Materials & Structures*, 25(8-9), 887–896.
- Stephens, R., Fatemi, A., Stephens, R., & Fuchs, O. (2000). Metal fatigue in engineering. *John Wiley & Sons*. Retrieved from https://books.google.pt/books?hl=pt-PT&lr=&id=B2aAPVa1TloC&oi=fnd&pg=PA1&dq=Metal+fatigue+in+engineering,+John+Wiley+%26+Sons,+2000.&ots=AAbywZk8Ct&sig=CwhktxNmYUtbTeTuyio_Y80QcyQ&redir_esc=y-v=onepage&q=Metal%20fatigue%20in%20engineering%2C%20John%20Wiley%20%26%20Sons%2C%202000.&f=false
- Tofique, M. W., Bergström, J., Hallbäck, N., & Burman, C. (2018). Fatigue initiation and strength of duplex stainless steel strip specimens in the very high cycle fatigue regime. *6th International Conference on VHCF*, 1-12.
- TSF. (s.d). Table of Preloads and Tightening Approximate Values. Retrieved from <https://www.tsftsh.com/docs/documentation/Precargas Web.pdf>
- Tsutsumi, N., Murakami, Y., & Doquet, V. (2009). Effect of test frequency on fatigue strength of low carbon steel. *Fatigue & Fracture of Engineering Materials & Structures*, 32(6), 473–483.
- Vernon, P., & Mackin, T. (2001). Fatigue failure analysis of a leg press exercise machine. *Failure analysis case studies, vol. II*, 255-266.