

FACULDADE DE ENGENHARIA DA UNIVERSIDADE DO PORTO

A Tool for Designing Eco-Friendly Adhesively Bonded Structures for Vehicles

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Master in Mechanical Engineering

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Abstract

The automotive sector is witnessing a growing popularity of adhesive connections due to their numerous advantages. Their key benefits include cost reduction, weight reduction of the overall structure, and subsequent fuel consumption reduction. Adhesive joints offer these advantages through several key factors, such as a more uniform stress distribution and the elimination of the need for additional elements such as screws or welding beads, making the manufacturing process easily automated. Moreover, adhesive connections do not require the creation of cut holes or any other geometric alterations in the substrate material. Additionally, they provide excellent vibration damping properties and can also act as effective seals, making them ideal for joining dissimilar materials.

This project aims to study the behavior of adhesive bonds with natural materials. More specifically, joints with wood and natural-based adhesive. Furthermore, it is intended to make the pre-design process of these eco-friendly adhesively bonded structures automatic and available for use.

To accomplish the primary objective, a website has been developed to provide a convenient platform for predicting the stress distribution in the overlap region for various types of joints. These include single lap joints, double lap joints, L-joints, corner-joints, T-joints, patch-joints, peel-joints, and inclined-joints. Users are empowered to customize the simulation by defining loading conditions and specifying geometric dimensions for the joint. The website also offers flexibility in terms of material definition as it provides a wide range of material type options, suitable for both conventional and natural materials. Additionally, users can input material properties based on the chosen material type. To further enhance the functionality and reliability of the simulation, the website grants users the option to apply a failure criterion. The results obtained through the website were compared with numerical results for different geometries and different types of materials. It was verified that the results were very close for most of the study cases.

In light of the limited knowledge concerning the performance of natural materials and the predominant emphasis on studying the commonly employed single lap joint (SLJ), it is of utmost importance to investigate other configurations, including L-joints, particularly within industries such as the automotive. Hence, experimental testing was performed on L-joints to gain in-depth insights into their characteristics. This investigation specifically targeted two types of wood: pine and oak. A key factor considered in the experiments was the orientation of the wood fibers, which varied between the test conditions. Subsequently, it was confirmed that oak wood with a vertical configuration exhibited superior strength capabilities. The experimental results were then compared with numerical simulations, and remarkably similar outcomes were observed between the two approaches when considering pine wood.

Keywords: Analytical modeling, finite element analysis, wooden bio-adhesive joint, web tool

Resumo

Atualmente o sector automóvel assiste a uma popularidade crescente das ligações adesivas devido às suas inúmeras vantagens. Estas vantagens-chave incluem a redução de custos, a redução do peso da estrutura e a subsequente redução do consumo de combustível. As juntas adesivas oferecem estas vantagens através de vários factores-chave, tais como uma distribuição mais uniforme das tensões e a eliminação da necessidade de elementos adicionais tais como parafusos ou cordões de soldadura, tornando o processo de fabrico facilmente automatizado. Além disso, as ligações adesivas não requerem a criação de furos ou quaisquer outras alterações geométricas no material do substrato. Adicionalmente, proporcionam excelentes propriedades de amortecimento de vibrações e podem também actuar como vedantes eficazes, tornando-as ideais para unir materiais dissimilares.

Este projecto tem como objectivo estudar o comportamento de ligações adesivas com materiais naturais. Mais concretamente, as uniões com madeira e adesivo de base natural. Além disso, pretende-se tornar o processo de pré-concepção destas estruturas adesivas ecológicas automático e disponível para utilização.

Para atingir o objectivo principal, foi desenvolvido um website que fornece uma plataforma conveniente para a simulação de vários tipos de juntas. Estas incluem juntas de sobreposição simples, juntas de sobreposição dupla, juntas em L, juntas de canto, juntas em T, juntas de remendo, juntas de arrancamento e juntas inclinadas. Os utilizadores podem personalizar a simulação definindo as condições de carga e especificando as dimensões geométricas da junta. O website também oferece flexibilidade em termos de definição de materiais, uma vez que fornece uma vasta gama de opções de tipos de materiais, adequados tanto para materiais convencionais como naturais. Além disso, os utilizadores podem introduzir as propriedades do material com base no tipo de material escolhido. Para melhorar ainda mais a funcionalidade e a fiabilidade da simulação, o website dá aos utilizadores a opção de aplicar um critério de falha. Os resultados obtidos através do website foram comparados com resultados numéricos para diferentes geometrias e diferentes tipos de materiais. Verificou-se que os resultados eram muito próximos para a maioria dos casos de estudo.

Tendo em conta os conhecimentos limitados sobre o desempenho dos materiais naturais e a ênfase predominante no estudo da junta sobreposta (SLJ), é da maior importância investigar outras configurações, incluindo as juntas em L, particularmente em indústrias como a automóvel. Assim, foram efectuados ensaios experimentais em juntas em L para obter uma visão aprofundada das suas características. Esta investigação incidiu especificamente sobre dois tipos de madeira: pinho e carvalho. Um fator-chave considerado nas experiências foi a orientação das fibras da madeira, que variou entre as condições de ensaio. Posteriormente, confirmou-se que a madeira de carvalho com uma configuração vertical apresentava capacidades de resistência superiores. Os resultados experimentais foram depois comparados com simulações numéricas, tendo sido observados resultados notavelmente semelhantes entre as duas abordagens quando se considerou

a madeira de pinho.

Keywords: Métodos analíticos, análise de elementos finitos, junta bio-adesiva de madeira, website

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*“Engineering is the closest thing to magic
that exist in the world.”*

Elon Musk

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Acronyms

CFRP	Carbon Fiber Reinforced Polymers
CLPT	Classical Laminated Plate Theory
CZM	Cohesive Zone Model
DCB	Double Cantilever Beam
DLJ	Double Lap Joint
ENF	End-Notched Flexure
FEA	Finite Element Analysis
FSDT	First Order Shear Deformation Theory
SLJ	Single Lap Joint
THM	Thermo-hydrromechanical
WFP	Wood Failure Percentage

Chapter 1

Introduction

1.1 Background and motivation

Throughout history, adhesives have been utilized as a means of joining materials. A variety of industries, including the automotive and aerospace sectors, have increasingly adopted adhesive joining techniques. Adhesive bonding has become a highly competitive and attractive replacement for conventional mechanical joining techniques such as bolting, riveting, welding, and brazing in various applications. The rationale for this shift is the numerous special benefits they provide over traditional methods. These benefits include the ability to connect different materials, improved stiffness, excellent load transmission capabilities, homogeneous stress distribution, increased design freedom, and capacity to produce regular shapes and reduced costs.

The transition towards vehicle electrification and global sustainability requires the development of lightweight and ecologically-friendly structures capable of supporting essential loads. However, commonly used materials such as metal alloys and synthetic composites have a high carbon footprint and require more energy for production when compared to renewable alternatives [1]. Consequently, there is an increasing interest in the utilization of naturally sourced substrates and adhesives in various applications. Specifically, within the adhesive industry, bio-based adhesives present a viable alternative to oil-based synthetic adhesives. Bio-based adhesives offer a promising and sustainable solution, derived from renewable resources, as a substitute for their synthetic counterparts. This trend reflects a desire to reduce the environmental impact of vehicle production and foster a more sustainable future.

The demand for eco-friendly adhesively bonded structures in automotive applications has increased. Therefore, it is crucial to provide resources that include bio-based materials to aid and promote the introduction of sustainable materials. This will ensure that structures can be designed rapidly and with ease in an effective and reliable way.

1.2 Objectives

The main objective of this work is to enhance the understanding of adhesively bonded wooden joints, particularly in the context of bonded vehicle structures. Therefore, the objectives of this work are twofold:

- Deepen knowledge in wood joint behavior.
- Develop a web application for designing adhesively bonded wooden joints.

The web application intends to expand the design possibilities for structures. The web application will empower users by offering a wide selection of materials, including bio-based options, and enabling them to specify the overall structural dimensions of the vehicle. The methodology employed in developing the web application draws inspiration from the successful *JointDesigner* tool, which optimizes the design of commonly used joint shapes. By adopting a similar approach, the proposed web application for bonded vehicle structures aims to be faster, more adaptable, and capable of facilitating the use of sustainable materials. Furthermore, it seeks to promote the introduction of bio-based materials, contributing to a more sustainable future.

Additionally, this work will involve conducting experimental tests to directly observe the behavior of bio-joints. These tests will provide valuable insights into the real-world performance of the joints, which will complement and validate the findings obtained through numerical simulations.

1.3 Research methodology

In order to achieve the aforementioned objectives, it was necessary to establish a specific methodology, which is as follows:

- Conduct a thorough literature review on adhesive joints and wood structures for predicting joint behavior.
- Study carefully analytical methods suitable for performing the necessary calculations in the pre-design phase of adhesive structures, encompassing both conventional and bio-based materials.
- Analyze the website that was previously implemented in the AJPU - the *JointDesigner* and develop a new one following a similar approach.
- Validate the new design tool through numerical simulations, for effective and reliable use.
- Carry out numerical analysis along with experimental tests to determine key properties and predict the behavior of wood and bio-adhesives.

1.4 Thesis outline

This thesis is divided into six chapters. This introduction section provides the background information and motivation for the study. As well as a summary of the problem currently being addressed and a description of the goals of the study.

Chapter 2 presents a comprehensive literature review, offering valuable insights into adhesive joints, wooden structures, classical analytical methods, and available adhesive joint design software.

Chapter 3 introduces the analytical methods implemented in the research. Both methods were carefully selected for their suitability in developing the website, with one method exhibiting greater versatility compared to the other and therefore chosen to be implemented. Additionally, the chapter presents the validation process conducted to ensure the accuracy and reliability of the implemented model.

Chapter 4 focuses on the development of the web application and presents the final interface results. In this chapter, the process of developing the web application is described in detail.

Chapter 5 presents the testing procedures conducted on L-joints with the objective of deepening the understanding of wood behavior. This chapter provides detailed insights into the materials employed, the manufacturing process of the joints, the experimental setup, and the corresponding numerical simulations. Furthermore, a thorough comparison is conducted between the experimental and numerical data to evaluate their consistency and reliability.

Chapter 6 serves as the concluding chapter of the project, summarizing the key findings and outcomes. It provides a comprehensive overview of the achievements of the project and highlights the significance of the research that has been conducted. Additionally, this chapter provides suggestions and recommendations for potential future work.

Finally, the appendices in this work provide supplementary information to support the study conducted. Specifically, they include all the implemented coding necessary for the analytical models explored in Chapter 3. Additionally, the appendices include the two scientific papers written within the framework of this thesis.

Paper 1: Analytical modeling of single lap joints, L-joints, and T-joints with bio-based materials for the automotive industry

AMS Couto, CSP Borges, P Tsokanas, S Jalali, EAS Marques, RJC Carbas, LFM da Silva (2023). Analytical modeling of single lap joints, L-joints, and T-joints with bio-based materials for the automotive industry. *International Journal of Solids and Structures*. Submitted.

Embracing sustainable practices can be greatly enhanced by exploring the potential of bio-based materials for adhesive joints. However, ensuring the successful implementation of these materials necessitates the provision of suitable tools and resources to facilitate effective design processes. The existing literature primarily focuses on methodologies for studying specific types of joint configurations. The approach of Bigwood and Crocombe stands out because it can be applied to a wider range of joint types, as long as end-loading values in the overlap edges are known. This work was focused on the automotive industry, where the most common joint geometries are single-lap joints, L-joints and T-joints. Thus, the problem of these joints under tensile loading was analyzed. An analytical study is performed using the methodology developed by Bigwood and Crocombe. The results in terms of peel and shear stress were compared with the ones obtained numerically to validate the results. This work aims to validate the accuracy of the results before creating a novel modular method that facilitates a fast and flexible design process for bonded vehicle structures. This approach would give users the ability to specify the overall dimensions of the structure and select the bio-based materials to be employed. The ultimate software employs a method analogous to that used by the JointDesigner web tool, which has been successfully developed to make it easier to design the most widely used joint geometry. The analytical method used proves its suitability for studying bio-based joints by accurately predicting critical stresses in different joint configurations, such as the SLJ, L-joint, and T-joint.

Paper 2: Experimental and numerical study on eco-friendly adhesive joints for the automotive industry

AMS Couto, CSP Borges, P Tsokanas, S Jalali, EAS Marques, RJC Carbas, LFM da Silva (2023). Experimental and numerical study on eco-friendly adhesive joints for the automotive industry.

In the mobility industry, there is an increasing recognition of the environmental consequences related to adhesive joints. As a result, efforts are being made to explore and implement environmentally friendly materials. Some examples of these materials include wooden substrates and bio-adhesives. This study focus on adhesive L-joint, which is a commonly employed configuration in the automotive sector for creating bonded structural components with significant bending rigidity. The investigation utilized pine wooden substrates and bio-based adhesives to examine the behavior of these joints. Two distinct configurations were studied, differing solely in the fiber orientation of the wood. The research combined experimental testing and finite element modeling to analyze the strength of the joints and determine their failure mode when subjected to tensile loading conditions. The findings indicate that the configuration of the joint plays a crucial role in its overall performance, with the vertical configuration demonstrating higher strength. Additionally, a high degree of agreement was observed between the experimental and numerical analyses.

Chapter 2

Literature review

2.1 Adhesive joints

2.1.1 General considerations

The history of adhesives dates back to pre-historic times when the majority of adhesives were created from organic sources. Around 1900, synthetic polymer-based adhesives were introduced, which led to the expansion of their usage. Nowadays, adhesives are used frequently in a variety of industries. The aerospace industry was a pioneer in utilizing adhesives, particularly with the introduction of composite materials in airplanes. Additionally, the automotive and rail industries are investing in this technology to produce lighter vehicles. Other industries that extensively use adhesives include civil, electronics, biology, and medicine [2].

In 1987, Kinloch et al. [3], defined adhesives as substances that, when applied to materials' surfaces, may join them together and resist separation. As a result, the word "adhesive" is used to describe a variety of substances, including glue, cement, paste, and others. Furthermore, an adhesive joint is made by joining two surfaces, whether they are made of the same material or not. Substrates are the general name for the materials intended to be bonded together with adhesives. Nevertheless, after the bonding has occurred, the word adherend is frequently used to describe the bonded components [2].

Adhesion, as described by Possart et al. in 2006 [4], refers to the attraction between two substances due to intermolecular interactions. Unlike cohesion, which involves forces within a single substance, adhesion involves the interaction between two different substances. Both adhesion and cohesion are primarily driven by intermolecular forces of the Van der Waals type. The level of adhesion can also be influenced by mechanical, electrical, and diffusion events.

Adhesively bonded joints have become increasingly popular in various applications, replacing traditional mechanical joints. They offer several advantages over mechanical fasteners. Firstly, they ensure a more uniform distribution of stress across the bonded area, resulting in enhanced load transmission and rigidity (see Figure 2.1). Consequently, this leads to weight reduction and cost savings. Adhesively bonded joints also exhibit superior damping properties and excellent fatigue strength. They are well-suited for joining thin plates and can effectively bond dissimilar

materials. While the strength of adhesives may be lower compared to metals, they are particularly suitable for structural applications involving thin plates with significant bearing regions. The application of adhesives is efficient and can be easily automated, while the joint design allows for adaptability, enabling the implementation of innovative concepts like sandwich structures. Additionally, adhesive bonding offers the advantage of extremely smooth surface finishes without holes or welding marks. The close contact between bonded surfaces enhances corrosion resistance and structural stability [2].

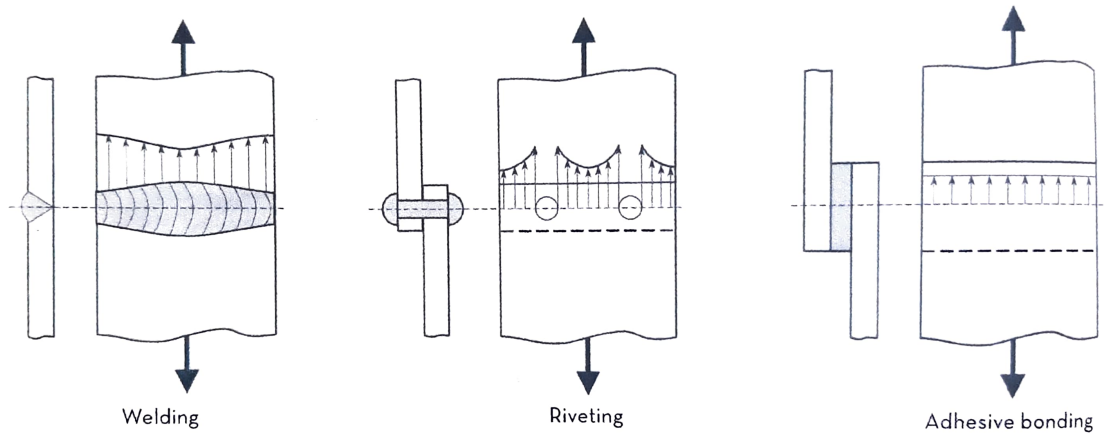


Figure 2.1: Stress fields obtained using three distinct joining methods [5].

However, adhesive bonding does come with certain drawbacks that need to be considered. One of the most significant challenges is the requirement for careful and appropriate surface preparation. Improper surface preparation can have a severe impact on joint strength and potentially lead to premature joint failure [5]. Another important aspect to consider is the sensitivity of adhesive joints to the loading direction. Concentrated loads in a small area can weaken the joint due to peeling and cleavage stresses. Furthermore, adhesives have limited resistance to extreme temperatures and humidity conditions due to their polymeric nature. Additionally, bonding does not occur instantaneously and may necessitate the use of specialized tools to maintain substrate positioning during the curing process. Moreover, certain adhesives require temperature for curing, which can result in additional costs. Quality control is also crucial but can be challenging, particularly for complex joint designs. Despite the availability of nondestructive techniques for quality control, there is still some skepticism regarding the long-term strength of adhesive bonding [2].

2.1.2 Adhesive classification

Adhesives may be classified in different manners. Some of them according to their polymer base, their function in the polymer "backbone", their physical form, their chemical composition, and their functional types [2].

Regarding their polymer base, they can be classified based on their use of natural or synthetic resources. On the one hand, adhesives that are derived from natural sources encompass vegetable and animal-based materials as well as natural gums (e.g. natural rubber, animal glue, casein-based, and protein-based adhesives). On the other hand, adhesives that have a synthetic origin comprise acrylics, epoxies, silicones, and polyesters [2, 6].

In terms of their function in the polymer "backbone", adhesives can be thermoplastic, thermoset, elastomeric and hybrid. Thermoplastic adhesives are cured by melting or solvent loss and can be melted without experiencing significant changes in their properties. This is due to the presence of secondary bonds between chains, which typically also makes these materials very ductile. Polymers that cure through chemical reactions at room or high temperature and cannot be melted after the initial cure, and have covalent bonds between polymeric chains, are referred to as thermosetting adhesives. These materials are usually stronger and stiffer. Elastomeric polymers are characterized by having a higher elongation, toughness, and peel strength. Lastly, hybrid adhesives are produced by combining the characteristics of the aforementioned types [2, 6].

Adhesives may assume different physical states, among them liquid form, paste, film, or powder. Their physical form is very important because it typically determines their method of application. Liquid adhesives can be applied using rolls, sprays, or brushes and they possess excellent gap-filling capabilities. Paste adhesives have high viscosity and little tendency to drip, making them suitable for use on vertical surfaces. Film adhesives ensure uniform bondline thickness and are easy to handle. Finally, powder adhesives usually require heating or dissolution into a solvent to transform them into a liquid form for application [2, 6].

Structural or non-structural may be another way to categorize adhesives, according to the load levels they can resist, i.e. regarding their functional type. Additionally, adhesives can be classified according to their characteristics, such as hot-melt, pressure-sensitive, water-based, ultraviolet/-electron beam cured and conductive characteristics [2].

Lastly, there are various classifications based on their chemical families. Adhesives can be classified as epoxy, acrylic, polyurethane among others. Epoxy adhesives are considered the most extensively used for structural applications because of their mechanical performance. They are thermosetting adhesives that are strong and rigid but can be modified to be more flexible and tough without compromising their tensile strength. Acrylics offer quick curing and high strength but their downside is that they are more expensive than epoxies. Polyurethane adhesives remain flexible and durable even at low temperatures, but they are sensitive to moisture and heat [2, 5].

2.1.3 Loading modes and joint design

The behavior of joints is closely linked to the type of mechanical loading they experience. Therefore, to ensure a successful design for adhesive joints, it is crucial to understand the various types of stress they can be subjected to during service, and how it affects the adhesive layer [7]. These stresses can be classified into four fundamental modes of loading: tension, shear, peel, and cleavage, as shown in Figure 2.2. However, in most real situations these loading modes may occur in combination [8].

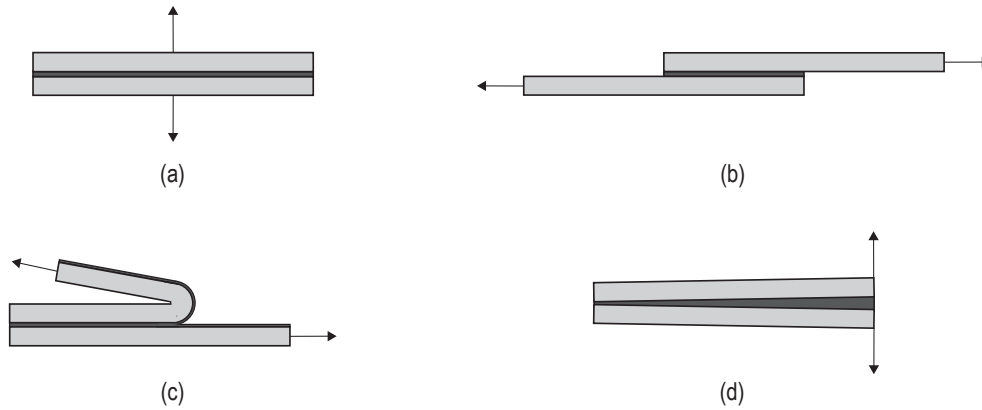


Figure 2.2: Modes of loading in the adhesive joints: (a) tension, (b) shear, (c) peel, (d) cleavage.

The extent of each stress is usually determined by various factors such as the manner in which the load is applied, the properties of the adhesive and substrates, and the joint's geometry. Therefore the selection of the appropriate adhesive must consider the design of the bonded joint, for two primary reasons. Firstly, the adhesive must be capable of handling the fundamental modes of loading that the joint will be subjected to. Secondly, the working life of the adhesive must be long enough to provide sufficient time for adhesive application, particularly when dealing with large joints [2]. Among the commonly used joint geometries, the single lap joint is popular because of its effectiveness in stress transfer and ease of manufacture [9]. Other frequently utilized joint configurations include double-lap joints, scarf joints, stepped-lap joints, strap joints, butt joints, bevel joints and tubular joints, as represented in Figure 2.3.

2.1.4 Failure modes and joint design

Failure modes in adhesively bonded joints can be categorized into adhesive failure, cohesive failure in the adhesive, mixed failure (adhesive-cohesive) and substrate failure. Figure 2.4 provides a visual representation of all these different failure modes [2]. Adhesive failure occurs at the interface between an adherend and the adhesive, often due to inadequate surface preparation or improper degreasing processes. Cohesive failure, on the other hand, typically occurs when the internal cohesion of a material tear, and both adherends remain coated with the adhesive layer. Mixed failure involves a combination of adhesive and cohesive failure on the same fracture surface. For the special case of composite or timber substrates, an additional failure mode may arise due to their low transverse tensile strength. This can lead to peak peel stresses at the ends of the overlap, resulting in interlaminar failure by delamination [8].

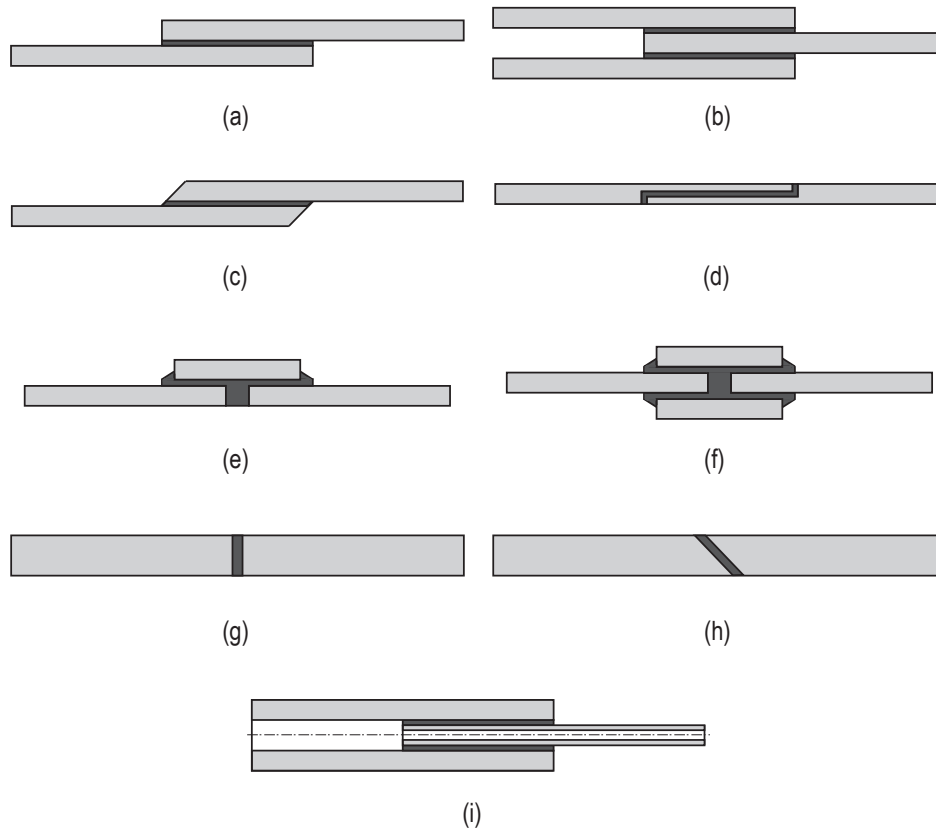


Figure 2.3: Most common joint configurations: (a) single lap joint, (b) double lap joint, (c) scarf joint, (d) step joint, (e) strap joint, (f) double strap joint, (g) butt joint, (h) bevel joint, (i) tubular joint.

2.1.5 Strength prediction approaches

Adhesive joints may be subjected to multiple loading conditions, as exposed previously. Peel and shear stresses are typically the most significant aspects to analyze when considering a strength analysis. In terms of joint design, the main goal is to ensure that the tensions within the adhesive layer are primarily governed by in-plane shear stresses. However, in practical scenarios, applied loads often generate bending moments that lead to substantial peel stresses, which must not be disregarded. Peel and shear stresses exhibit peak values at the edges of the overlap region. These stresses tend to approach infinity as the adhesive thickness approaches zero. Consequently, the initiation of joint failure commonly occurs in this region. When assessing the strength of adhesive joints, three main approaches are commonly employed: continuum mechanics, fracture mechanics, and damage mechanics. Continuum mechanics provides a broad framework for analyzing joint behavior. Fracture mechanics focus on the propagation of cracks and offers valuable insights into crack growth and failure. Damage mechanics combines aspects of both approaches to capture the evolution of damage and its influence on joint performance [10]. More details can be found in

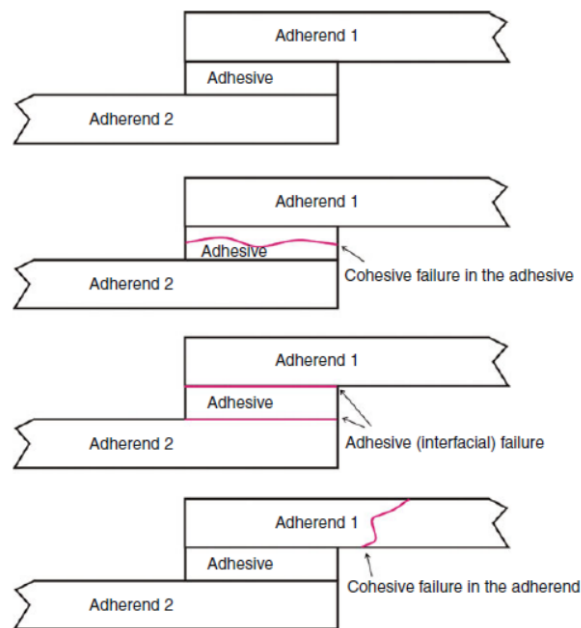


Figure 2.4: Common failure modes observed in bonded joints [2].

the following sections.

Continuum mechanics

The continuum mechanics approach considers both the adhesive and adherends as continuous materials, assuming a perfect bond at the interface while disregarding its specific properties [11]. This approach primarily focuses on studying the stresses, deformations, and maximum loads that can be applied to the bonded parts. The failure criterion in this model relies on two parameters: stress and strain. However, it is not suitable for accurately predicting adhesive joint failure due to the presence of singularities at the overlap edges. The stress values at these edges increase with mesh refinement in finite element analysis, making it challenging for the model to converge. Although some researchers have addressed this issue, their methods are only applicable to continuous structures with well-defined geometries, limiting the scope of the approach [11].

Fracture mechanics

The fracture mechanics approach is specifically designed to account for the presence of cracks and other defects within materials, including manufacturing defects, by treating them as discontinuities. This approach enables quantitative analysis of the behavior of structures containing cracks, allowing for the assessment of crack propagation and determining if the crack size exceeds the critical failure size. The criteria utilized in this approach typically involve a stress intensity factor

criterion and an energy release rate. However, a limitation of this model is its inability to accurately depict the initiation of damage. It is primarily suited for studying the propagation of damage rather than the initial stages of damage formation [11].

Damage mechanics

To address the limitations observed in the aforementioned approaches, various methods have been developed to incorporate both damage initiation and propagation. One of the most widely employed methods to predict damage is the cohesive zone model (CZM), which typically employs interface elements with no thickness to establish bonding properties between two solid surfaces. Each element incorporates a damage variable that is used to simulate material degradation. Once a stress criterion is met, a relaxation process based on a fracture energy criterion is implemented. This method successfully circumvents stress singularities and does not necessitate the presence of pre-existing cracks or any form of intervention to initiate propagation [11].

Traction-separation laws are utilized to apply this technique for reproducing adhesive behavior, as illustrated in Figure 2.5. Typically, a triangular or bilinear law is used due to its simplicity and efficacy in modeling adhesives with brittle failure behavior. In contrast, a trapezoidal or trilinear law is more suitable for ductile adhesives.

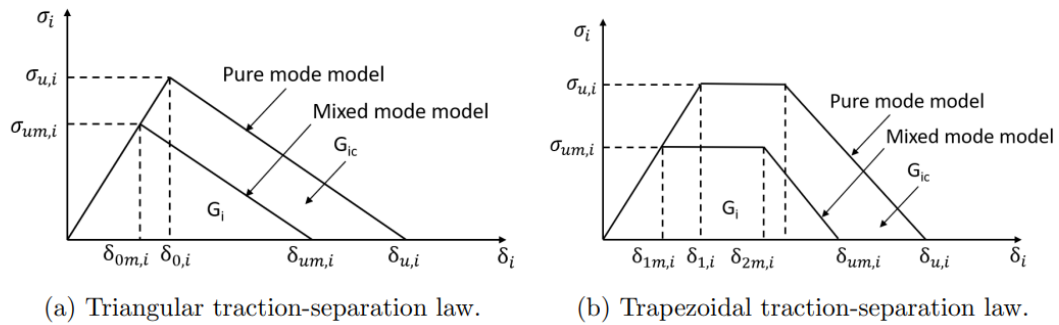


Figure 2.5: Schematic representation of traction separation laws, $i = I, II$ [12].

2.2 Wood substrates

Since ancient times, humanity has used wood for a variety of daily functions due to its renewability, ability to be cut into different shapes, durability, good mechanical strength, abundance in many regions, reasonable price, excellent thermal and acoustic insulation properties and attractive appearance. Additionally, its lightweight nature and resilience in various environmental conditions have also contributed to its widespread use, making it a highly sought-after material [13, 14].

The inherent property of wood to change its shape in response to environmental factors such as moisture, temperature, and loading has paved the way for the creation of intricate and refined geometric configurations in various objects. Through industrial processes such as sawing, cutting,

gluing, pulping, and chemical treatments, wood can be transformed into a diverse range of products. This has given rise to an extensive array of industrial wood products, including plywood, medium-density fiberboard, and laminated timber. The processability of wood plays a crucial role in its widespread application across multiple industries [13]. Wood's unique characteristics have found applications in numerous fields such as furniture, sports products, and vehicles, which can be ships, trains and aircraft. Moreover, wood is being recognized as a means to increase carbon sequestration capacity and reduce energy consumption. As a naturally grown and engineered material, wood contributes to the reduction of greenhouse gases by storing carbon absorbed from the air during tree growth. The production and processing of wood products require less energy compared to most traditional building materials, resulting in a lower carbon footprint (see Figure 2.6) [13, 1].

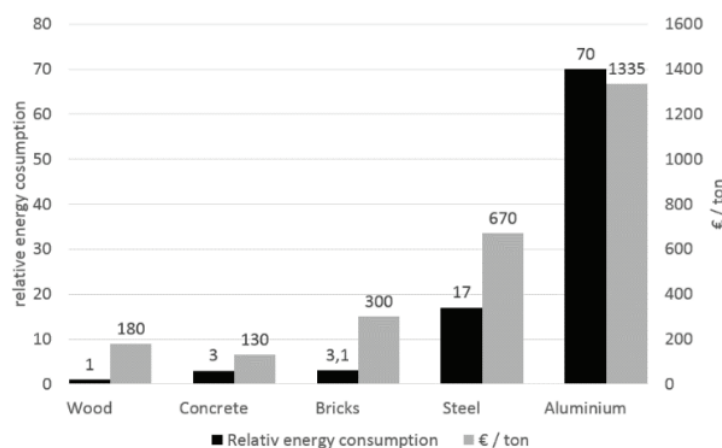


Figure 2.6: Average relative energy consumption of the production of a tonne of selected raw materials, with wood as reference and prices [1].

In terms of nomenclature, it is worth noting that the terms "timber" and "lumber" are often used interchangeably to refer to wood. However, it's important to distinguish between them based on their processing stages. In general, "lumber" refers to newly cut wood with the bark intact, while "timber" refers to wood that has been sawn into an intermediate product.

2.2.1 Structure of wood

Understanding wood's microstructure is important because it is a key factor in determining the characteristics of wood substrates. Figure 2.7 depicts the essential microstructure components, showcasing the progression from the outer part (left side) to the inner part (right side), encompassing outer bark, inner bark, vascular cambium, sapwood, heartwood, and the pith.

In the very outer part, it is found the outer bark, which acts like a protective layer for the inner bark. Its main function is preventing excessive water loss through evaporation. In turn, the inner bark plays a vital role in transporting sugars produced through photosynthesis from the leaves to the tree's roots and growing areas. Situated between the bark and the wood, the vascular cambium

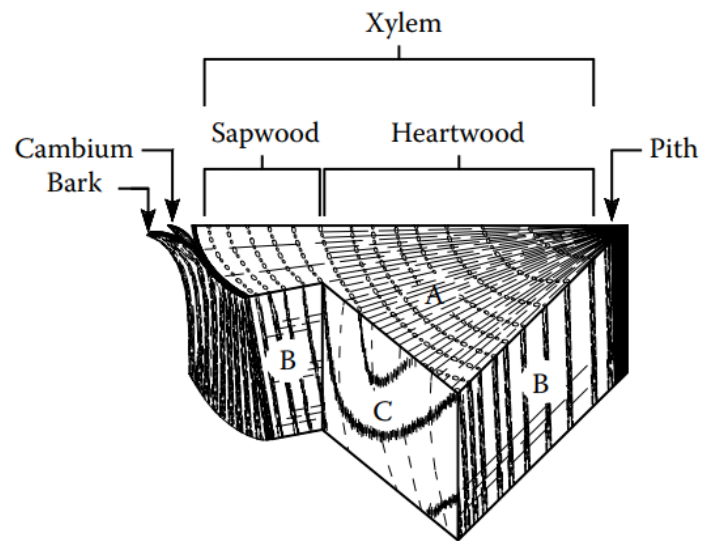


Figure 2.7: Schematic representation of wood section cross section, illustrating: (A) transverse surface, (B) radial surface, (C) tangential surface [13].

is responsible for generating both of these tissues, annually. The sapwood, which is the active and living wood, carries water from the roots to the leaves. Oppositely, the heartwood is a zone of inactive cellular tissue that no longer conducts water and is usually darker than the sapwood. Finally, the pith, located at the center, represents the residual tissue from the early growth stage before the wood formation even occurred [14, 15].

The stem or also named the trunk is composed of concentric annual rings. The number of annual rings present indicates the age of the tree. Within each annual ring, the light-colored concentric rings formed early in the growing season are referred to as earlywood, while the darker and denser concentric rings are referred to as latewood. In contrast, tropical trees do not have distinguishable annual rings due to minimal seasonal climate variation [15].

Softwood and hardwood are the two main categories of timber, with the former derived from gymnosperms like spruce and pine species, and the latter from angiosperm trees such as beech and oak [14]. These two types differ significantly in their properties, which can be attributed to their fundamentally different anatomies or component cells [14, 16]. When comparing softwoods and hardwoods, it becomes evident that softwoods possess simpler basic structures, characterized by only two types of cells with minimal variation within these cell types. In contrast, hardwoods exhibit greater structural complexity due to a larger number of basic cell types and a significantly higher degree of variability within these cell types. The most significant distinguishing feature between these two wood categories is the presence of vessel elements in hardwoods, which are absent in softwoods. However, an important cellular similarity between softwoods and hardwoods is that, in both types, the majority of cells become non-living as they mature, even in sapwood. The small portion of cells that remain alive at maturity is referred to as parenchyma cells and can be found in both softwoods and hardwoods [14].

In practical applications, softwood is commonly utilized for structural building purposes, whereas hardwoods have been less frequently employed as construction materials. This is primarily due to the challenges associated with processing and the limited availability of design methods. However, it is worth noting that hardwoods hold great potential for structural applications, given their abundant availability in Central Europe and their remarkable mechanical resistance [17].

Cellulose, hemicellulose, and lignin are the three primary biopolymers of wood. Cellulose microfibrils are embedded in hemicellulose matrices and reinforced by lignin in both the primary and secondary walls (Figure 2.8). Lignin is the primary constituent of the intercellular regions, serving as a binding material that confers stiffness to the wood against external loads and growth stresses. The cellulose microfibril plays the role of fiber in this bio-composite material, consisting of multiple linear polymer chains linked together through hydrogen bonding and Van der Waals forces. These microfibrils are combined to create larger fibrils and lamellae, which confer high tensile strength to the cell wall. Hemicelluloses are branched-chain polymers made up of various sugar units that link cellulose to lignin. They are capable of permeating water and thus provide flexibility and support to the cell wall. Lignin, a phenylpropanoid polymer, is a crucial component that ensures the structural integrity of the material by contributing to the stiffness and strength of the cell wall. Additionally, it shields the other two constituents from the assault of enzymes and microbial organisms [18, 19].

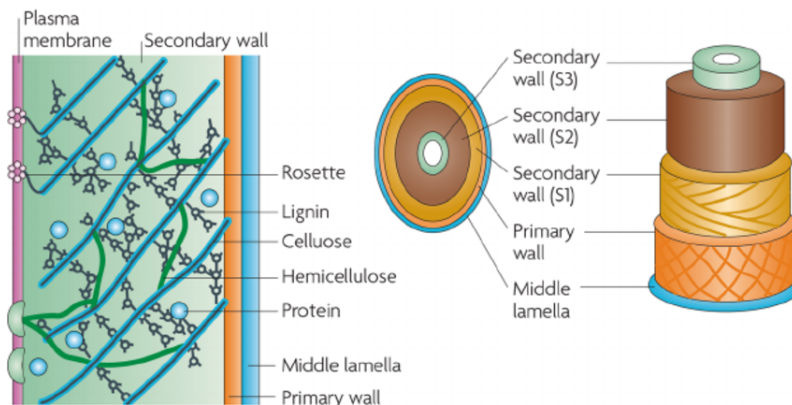


Figure 2.8: Structure of the wood cell wall [20].

2.2.2 Mechanical properties of wood

The anisotropic nature of wood derives from its intricate anatomical structure and the arrangement of cellulose within the cell wall. Wood can be considered a cylindrical orthotropic material with three principal axes of orthotropy: longitudinal (parallel to the fibers), radial (normal to the growth rings), and tangential (perpendicular to the grain and tangent to the growth rings). Figure 2.9 illustrates these principal axes of orthotropy in wood. The longitudinal grain direction aligns the wood cells predominantly, resulting in a dominant axis with higher stiffness and strength. Conversely, the other directions exhibit significantly weaker characteristics. As a consequence,

wood tends to fail along planes parallel to its fibers, often initiated by tension perpendicular to the grain or shear loading.

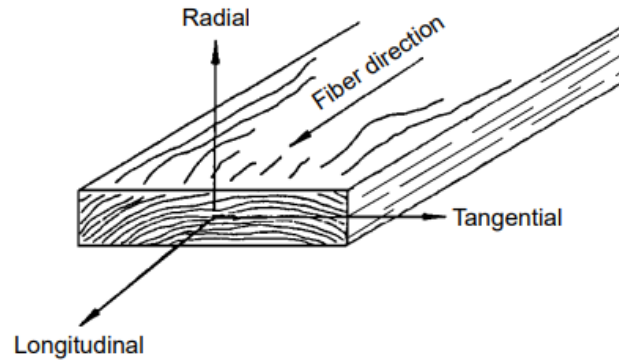


Figure 2.9: Principal directions defined for anisotropic structure of wood [14].

The orthotropic elasticity theory utilizes the generalized Hooke's law to describe the relationship between stresses (σ) and strains (ϵ). This law requires nine independent constants - three Young's modulus (E_L, E_R, E_T), three Poisson's ratios ($\nu_{LT}, \nu_{LR}, \nu_{TR}$) and three shear modulus (G_{LR}, G_{LT}, G_{TR}) to fully characterize the elastic behavior of the material. These constants can be experimentally determined through various testing methods [13, 17]. In certain cases, wood can be approximated as transversely isotropic, where equal material properties are assumed in the radial and tangential directions for the sake of simplicity [15, 17, 21].

2.2.3 Methods of wood delignification and densification

The utilization of timber, especially from fast-growing species, has gained significant interest as an environmentally conscious alternative. However, these species often possess lower wood density and mechanical properties, which restrict their applications [22, 23]. To overcome these limitations, increasing the density of these materials can prove beneficial. By doing so, they have the potential to become high-value, high-performance products, serving as substitutes for harder species or conventional alloys in demanding applications [18].

In recent years, thermo-hydrmechanical (THM) treatments have emerged as a method to enhance the density and mechanical properties of wood [18]. The viscoelastic nature of wood makes its mechanical properties dependent on moisture, temperature, and time. Wood's lignin and hemicellulose components exhibit a glass transition temperature (T_g), which marks the transition from stiff and brittle behavior to a rubbery and flexible state. This transition occurs due to the mobility of polymers and the resulting molecular reorganization when temperatures exceed the T_g of these polymers. This phenomenon allows for significant deformation under compression through cell wall buckling rather than fracture [24, 25, 26]. Moisture plays a crucial role in T_g ,

as its increase leads to a decrease in T_g . Mainly because moisture acts as a plasticizer, reducing secondary bonding between polymer chains and increasing polymer molecule mobility [25].

The densification process, however, faces a significant challenge related to its susceptibility to irreversible swelling caused by moisture. This phenomenon, referred to as "set-recovery", occurs due to the release of elastic energy stored in cellulose macromolecules and the presence of unbroken covalent and hydrogen bonds between polymers [25, 27, 28]. To address this issue, chemical pre-treatments based on delignification processes can be employed to enhance the rate of densification [23]. These methods involve partial or complete elimination of lignin through the use of boiling alkaline solutions, which facilitate intricate reactions. These delignification methods break down lignin-hemicellulose bonds and degrade and solubilize lignin and hemicellulose in an alkaline environment. The reduction in lignin content increases the hydrophilicity of the material, as lignin is the hydrophobic component of wood. The removal of lignin results in increased flexibility, malleability, and porosity of the material while maintaining the fibrous cell wall structure. The decreased transverse rigidity of the cell walls, achieved by lignin removal, enables smooth compression without flaws, thereby enhancing the mechanical properties of the material [29, 30, 31, 32]. Despite the absence of a substantial lignin matrix, which typically acts as a binding and structural component, densified wood retains its integrity due to the formation of fixed hydrogen bonds between cellulose fibrils facilitated by the presence of free hydroxyl (OH) groups during compression drying. Additionally, the physical interlocking of cellulose fibers contributes to this phenomenon [29, 31].

2.2.4 Strength failure criteria

Wood materials utilized in structural applications often encounter multiaxial loading, resulting in a complex stress state. Accurately predicting wood failure under such conditions necessitates the application of appropriate failure criterion. As mentioned previously, wood is categorized as an orthotropic material. Hence, failure criteria developed for other orthotropic materials (e.g. composites), can be employed. Among these criteria, a notable distinction exists between non-interactive criteria and interactive criteria. The former primarily analyze individually the stress and strain components, offering the benefit of promptly identifying the mode of failure. Conversely, the latter takes into account the more complex stress state that arises from the combined effects within the drawback of more complex calculations. Non-interactive criteria encompass the maximum stress criterion and maximum strain criterion. In turn, interactive criteria comprise the empirical Hankinson, Killmann and Cotê, Tsai-Hill, Tsai and Wu [13]. These criteria will be detailed in the subsequent subsections.

2.2.4.1 Maximum stress criterion

The maximum stress criterion determines failure when one of the stress tensors within the symmetry system of the material reaches the respective strength value.

$$\begin{aligned}
\sigma_i &= \sigma_{ui}^t \quad \text{for } \sigma_i > 0; i = L, R, T \\
\sigma_i &= \sigma_{ui}^c \quad \text{for } \sigma_i < 0; i = L, R, T \\
|\sigma_{ij}| &= \sigma_{uij}; \quad ij = RT, LT, LR
\end{aligned} \tag{2.1}$$

where ‘t’ and ‘c’ represent the tensile and compressive strengths, respectively.

2.2.4.2 Maximum strain criterion

The maximum strain criterion determines failure when one of the strain components in the material symmetry system reaches its corresponding failure value. The given criterion can be formulated based on stresses utilizing constitutive equations, and it yields comparable outcomes when compared to the maximum stress criterion.

$$\begin{aligned}
\varepsilon_i &= \varepsilon_{ui}^t \quad \text{for } \varepsilon_i > 0; \quad i = L, R, T \\
\varepsilon_i &= \varepsilon_{ui}^c \quad \text{for } \varepsilon_i < 0; \quad i = L, R, T \\
|\varepsilon_{ij}| &= \varepsilon_{uij} \quad \text{where } ij = RT, LT, LR
\end{aligned} \tag{2.2}$$

where ‘t’ and ‘c’ represent the tensile and compressive strengths, respectively.

2.2.4.3 Empirical Hankinson

A well-known interactive criterion for wood, widely utilized in the industry, is the empirical Hankinson criterion [33]. This criterion enables the prediction of wood strength when subjected to uniaxial compressive loading that may not be aligned with the grain orientation. This criterion is strictly empirical but has demonstrated a consistent agreement with experimental results. Considering the case of the LR plane, it follows:

$$\sigma_\theta = - \frac{\sigma_{uL}^c \sigma_{uR}^c}{\sigma_{uL}^c \sin^2 \theta + \sigma_{uR}^c \cos^2 \theta} \tag{2.3}$$

where θ is the angle between grain orientation and the load applied.

2.2.4.4 Kollmann and Coté

In situations where wood is subjected to uniaxial tensile loading in a direction that is not aligned with the grain orientation, an alternative formulation based on the empirical Hankinson criteria can be employed to estimate its strength. This methodology, initially proposed by Kollmann and Coté [34], provides an expression that allows for the prediction of wood strength in such specific scenarios:

$$\sigma_\theta = \frac{\sigma_{uL}^t \sigma_{uR}^t}{\sigma_{uL}^t \sin^n \theta + \sigma_{uR}^t \cos^n \theta} \tag{2.4}$$

where n denotes a constant value between 1.5 and 2.0.

2.2.4.5 Tsai-Hill criterion

The Tsai-Hill [35] criterion, derived from the Hill expression initially formulated for metals, is an invaluable tool when incorporating wood into complex structures. This criterion proves particularly useful because it allows for the utilization of more comprehensive criterion specifically tailored for orthotropic materials, enabling the establishment of failure limits. It is given by:

$$A(\sigma_R - \sigma_T)^2 + B(\sigma_T - \sigma_L)^2 + C(\sigma_L - \sigma_R)^2 + 2D\sigma_{RT}^2 + 2E\sigma_{LT}^2 + 2F\sigma_{LR}^2 = 1 \quad (2.5)$$

where A, B, C, D, E, F are constants depending on failure strengths. They are determined as follows:

$$\begin{aligned} B + C &= \frac{1}{\sigma_{uL}^2}; & A + C &= \frac{1}{\sigma_{uR}^2}; & A + B &= \frac{1}{\sigma_{uT}^2}; \\ 2D &= \frac{1}{\sigma_{uRT}^2}; & 2E &= \frac{1}{\sigma_{uLT}^2}; & 2F &= \frac{1}{\sigma_{uLR}^2} \end{aligned} \quad (2.6)$$

When examining the scenario of plane stress in the LR plane, the Tsai-Hill criterion is utilized to predict failure. It determines failure by verifying whether the following expression holds:

$$\frac{\sigma_L^2}{\sigma_{uL}^2} - \frac{\sigma_L \sigma_R}{\sigma_{uL}^2} + \frac{\sigma_R^2}{\sigma_{uR}^2} + \frac{\sigma_{LR}^2}{\sigma_{uLR}^2} = 1 \quad (2.7)$$

2.2.4.6 Tsai and Wu

In 1971, Tsai and Wu [36] introduced a tensorial quadratic polynomial criterion that incorporates linear terms to account for the possibility of material failure under a hydrostatic stress state. The quadratic terms in the Tsai-Wu criterion define an ellipsoid in the stress space, capturing the interactions between normal stresses. By including linear terms, the criterion accommodates different tensile and compressive strengths, resulting in a more accurate representation of the behavior of composite materials. The required strengths can be determined through uniaxial tests. However, one remaining parameter that needs to be determined is the interaction stress coefficient F_{LR} . Precise measurement of this coefficient would necessitate biaxial loading tests involving stresses σ_L and σ_R , which are particularly challenging to perform for wood. Additionally, the inherent material inhomogeneity contributes to significant scatter in the data, leading to considerable uncertainty surrounding the value of F_{LR} . This criterion acknowledges the importance of addressing such conditions and presents the following expression:

$$F_{ij}\sigma_{ij} + F_{ijkl}\sigma_{ij}\sigma_{kl} = 1; \quad i, j, k, l = L, R, T \quad (2.8)$$

where F_i and F_{ij} are strength tensors determined experimentally.

Considering orthotropic materials subjected to plane stress in the LR plane, the Tsai-Wu criterion can be expressed as follows:

$$\left(\frac{1}{\sigma_{uL}^t} - \frac{1}{\sigma_{uL}^c}\right)\sigma_L + \left(\frac{1}{\sigma_{uR}^t} - \frac{1}{\sigma_{uR}^c}\right)\sigma_R + \frac{\sigma_L^2}{\sigma_{uL}^t \sigma_{uL}^c} + \frac{\sigma_R^2}{\sigma_{uR}^t \sigma_{uR}^c} + 2F_{LR}\sigma_L\sigma_R + \frac{\sigma_{LR}}{\sigma_{uLR}} = 1 \quad (2.9)$$

2.2.5 Wood for structural vehicle components

The utilization of wood in the automotive sector has gained significant attention and interest. Companies like Volkswagen [37] have expressed their interest in exploring wood materials as viable options for structural applications in the automotive industry. Mainly because wood offers numerous advantages due to its appealing potential and technological possibilities [1]. However, it also presents challenges, particularly in terms of mechanical compatibility, reliability, process compatibility, and reproducibility.

Reducing the overall weight of a vehicle is crucial in minimizing its CO_2 emissions, and substituting denser materials such as steel with wood-based materials can potentially achieve weight reductions of approximately 15-20% while maintaining performance [1]. Apart from the aforementioned advantages, wood also offers practical feasibility compared to other materials. With the expected increase in demand for finite resources, especially in developing countries, the scarcity of raw materials may lead to higher prices. In contrast, wood is a renewable and abundant material that offers reliability and price stability.

In the automotive industry, the process chain has mainly focused on the processing of metal materials, hence its process chain is mainly adapted to the processing of steel-dominated structures, posing a serious challenge. In terms of design flexibility, the introduction of wood comes with certain drawbacks as it is difficult to obtain complex geometries. In those cases, steel cannot be directly substituted by wood. The material properties and parameters strongly influence the design. As wood is a heterogeneous natural material, there are many factors that influence its properties such as growth, location and time of harvest. Furthermore, the number of layers, orientation of the fibers and choice of adhesive will influence a lot the final properties of the assembly. Therefore, it requires a careful study of the tensile, bending, and shear strength and modulus [1].

The conventional method of joining wooden components involves using mechanical fasteners such as bolts, and screws [17, 38, 39]. However, these techniques have a major disadvantage, as they require drilling holes into the substrates, which damages the wood fibers and creates areas of high stress concentration, ultimately leading to structural failure. To tackle these issues, adhesive bonding has emerged as a promising alternative for joining wood components.

The introduction of bio-adhesives is a very good way of obtaining an overall bio-joint. There are different applications of bio-adhesives, the most common the wound closure and joining wood panels [40]. Also, there are various adhesive families used synthetic to eco-friendly adhesives, with the latter being preferred due to their lower environmental impact. Moreover, they are divided between structural and non-structural applications. It should be noted that the bondability of wood is affected by its surface properties, porosity, density, moisture content, and dimensional changes

[41]. Wood with a lower density is easier to bond because the adhesive can penetrate it more easily, resulting in better mechanical interlocking.

Regarding the mechanical and fracture toughness properties of an adhesive joint, they may be determined through strength and fracture tests. To assess the joint's performance, the wood failure percentage (WFP) value should be evaluated [42]. It represents the percentage of the overlap area that failed on the wood substrate. Therefore, a higher WFP indicates a stronger bond, provided that the adhesive layer was not the primary failure region. If the failure occurs entirely on the wood substrate, the joint's limit coincides with the limit of the substrate, which is the optimal scenario.

In terms of strength tests, the most common is the single lap joint test. It is used to characterize the combination of both adhesive and substrate materials [43]. The reason behind it is that there are numerous conditions that can be assessed from this type of joint, including adhesive/substrate bondability, joint performance under varying temperatures, humidity conditions, loads, and more.

The double cantilever beam (DCB) and end-notched flexure (ENF) tests are the most frequently used fracture tests for adhesive joints, including wood adhesive joints. These tests are carried out to characterize the adhesive and determine the joint's mode I and mode II fracture toughness.

There are some conditions that influence the behavior of the wood joint, such as moisture and fatigue loading. The first is related to the environmental conditions and provokes the wood to absorb moisture and make it shrink or swell. This effect promotes higher stress on the adhesive layer. Since the moisture content of wood varies continuously with environmental conditions, moisture can impact the strength of the joint. This could potentially cause wood delamination, or in more severe cases, result in macro-scale joint failure. In regard to fatigue loading wood structures are highly vulnerable. If long-term cyclic loads accumulate at stress levels below the static strength of the material, structures can experience premature failure over time [44].

2.3 Analytical methods

Analytical methods have been developed primarily to predict the stress and strength of adhesively bonded joints, providing a valuable means of evaluating joint performance. These methods utilize mathematical expressions to calculate the stress levels within the adhesive joint. In the initial stages of design, analytical approaches can be advantageous as they offer quick and relatively simple solutions that are reasonably accurate. However, their viability is dependent on the simplicity of the joint geometry, load scenario, and adhesive behavior [5, 7].

It should be noted that there are specific conditions under which analytical solutions are valid, including the linearity or nonlinearity of the adhesive and adherends, as well as the isotropic or orthotropic behavior of the adherends' material. Analytical models can be implemented in calculation processes and software, but more advanced analytical techniques often require extensive computations to obtain a solution and should only be employed when necessary.

However, when it comes to optimizing joint strength and obtaining accurate results, numerical approaches are always required. Numerical methods offer the flexibility to handle complex

geometries, nonlinearity, and various loading conditions, making them essential for achieving optimal joint designs [7, 43].

Several classical analytical methods have been developed for analyzing adhesively bonded joints, including the Volkersen, Goland and Reissner, Hart-Smith, and Adams methods.

In 1938, Volkersen [45] formulated a straightforward method for analyzing adhesive joints, which involves computing only the shear stress. The method simplifies the adherend as a rod that undergoes axial deformation exclusively, while the adhesive is modeled as a continuous shear spring. Due to these assumptions, the method disregards the moment that may naturally arise in some types of joints. The strain reduction in the adherends along the overlap and the continuity of the adhesive/adherend interface result in a not evenly distribution of shear strain and stress in the adhesive layer. More specifically, the shear stress is highest at the ends of the overlap and decreases towards the middle, where it is significantly lower.

In 1944, Goland and Reissner [46] created a model based on Volkersen's formulation while introducing the bending effect. This results in the consideration of joint rotation, where the applied loads tend to align along an imaginary horizontal line. As a result, the bending moment decreases as the joint rotates further, leading to a nonlinear geometric problem that requires accounting for the significant deflections of the adherends. The method resorts to a bending moment factor and a transverse force factor to establish the relationship between the applied tensile load per unit width to the bending moment and transverse force at the ends of the overlap. Furthermore, in order to simplify the solution, Goland & Reissner's method disregards shear and normal deformations in the adherends. This method emphasized the critical role of the bending moment, highlighting the importance of accounting for peel stresses in the analysis. Subsequently, Ojalvo & Eidinoff [47] rectified the Goland and Reissner approach.

In 1973, Hart-Smith [48] introduced a model that accounted for the plastic deformation of the adhesive material, going beyond its elastic response. This model recognized that plastic yielding at the edges of the overlap region does not necessarily result in immediate joint failure. It demonstrated that a significant portion of the adhesive layer can still bear additional loads even after plastic deformation occurs. However, the practical application of this model is limited by the requirement for accurate knowledge of the elastoplastic properties of the adhesive, which are often not readily available. As a result, its widespread use is constrained.

In 1997, Adams [49] proposed a straightforward analytical approach that enables the determination of the failure load for adhesive global yielding and adherend yielding. In cases where substrates yield, a plateau is observed at a specific overlap value corresponding to adherend yielding, making it easy to predict joint strength. However, for intermediate or brittle adhesives and non-yielding adherends, the analysis is less robust, and the author recommends using either the finite element method or a more comprehensive analytical solution. The failure load of the adhesive joint with elastic adherends corresponds to the total plastic deformation of the adhesive, i.e., yielding everywhere, while the maximum load that can be sustained without causing adherend yielding corresponds to the failure load of the adhesive joint [50, 51].

In 1989, Bigwood and Crocombe [52] proposed an extension of the Goland-Reissner model

[46] that allows for its application to various joint geometries. This model requires the specification of end-loading values on each overlapping edge, making it suitable for analyzing different joint configurations using a sandwich-type analysis. More recently, Weißgraeber et al., in 2014, introduced a versatile sandwich-type model [53], which builds upon the method proposed by Bigwood and Crocombe. The innovation lies in its more accurate approach to capturing the shear deformations of the adherends, achieved by incorporating the First Order Shear Deformation Theory (FSDT) into the model. As a result, this enhanced model offers a wider range of applications, facilitating the analysis of diverse joint configurations and suitable for predicting the behavior of adhesive joints with both isotropic and arbitrary laminated adherends.

2.4 Adhesive joint design software

The widespread use of structural adhesives in typical mechanical projects is significantly limited due to the lack of confidence in the prediction of their strength. Currently, joint strength calculations rely on technical data sheets provided by adhesive manufacturers, which primarily consist of data from industry-standard single lap tests. Another approach to perform joint strength calculations is by utilizing finite element software, which enables Finite Element Analysis (FEA). However, this method requires substantial computational resources, especially for achieving high precision, and necessitates skilled personnel. Consequently, experts and researchers may use it for more complex geometries or research purposes [54]. Another potential approach is the implementation of analytical models. However, as the analytical equations become more complex in an effort to account for variables such as adhesive thickness, plasticity, thermal effects, and composite materials, solving them necessitates significant computational resources. To address the technical demand, a computation tool of moderate complexity that can assist designers in their standard bonding practice would be the ideal solution.

Some commercial PC-based analysis/design software packages that already exist to assist in the design of efficient joints are: *Bolt*, *BISEPSLOCO*, *BISEPSTUG*, *CoDA*, *DLR*, *Feloco*, *PAL*, *RETCALC*, *ESDU*, *JointDesigner*, *JointCalc*. Appendix A presents more details about them, retrieved from [54]. It should be noted that the software packages are often narrow in scope and typically only cover one or two joint geometries. Additionally, only a few models are suitable for complex materials (e.g. composites).

JointDesigner

The *JointDesigner* is an innovative software package designed to provide accurate predictions of adhesive joint behavior. Accessible through a user-friendly website, it offers a range of powerful features and capabilities. Users have the flexibility to define various aspects of the joint, including material behavior (e.g. elastic or plastic) and material type (e.g. isotropic or anisotropic). The software allows users to input specific load intensities, material properties (e.g. Young's modulus, Poisson's ratio, tensile strength, and shear strength), and material geometry parameters (including thickness, overlap length, and width).

Currently, the software includes two joint configurations: single lap and double lap joints. It enables the selection of different analytical models, namely Volkersen, Goland and Reissner, Hart-Smith, Bigwood and Crocombe, and Adams. All of these models are suitable for studying the elastic behavior of the joints. However, when it comes to studying the plastic behavior of the material, only the Hart-Smith and Adams models are applicable. More specifically, the Hart-Smith model is suitable for analyzing plastic adhesives with elastic adherends, while the Adams model can handle full plastic joints.

Upon completing the analysis, the *JointDesigner* website provides users with visual representations of peel and shear stress distributions, allowing for a better understanding of the joint's performance. Additionally, users have the option to save or download the calculated results in various file formats, such as *Excel*, *CSV*, and *PDF*, ensuring easy access and documentation of the analysis outcomes.

Chapter 3

Analytical analysis

This section aims to provide an overview of the analytical models used in this thesis. Bigwood and Crocombe's method was utilized as the first approach to predict the adhesively joint's behavior. Specifically, this method was applied within the simplified approach and documented in Appendix D. However, to broaden the scope of analysis and modeling a wide range of materials, from the simplest (e.g. isotropic) to more complex ones (e.g. asymmetric layups), a more extensive solution was adopted. Consequently, this section will first present Bigwood and Crocombe's method [52], followed by the exposition of the general sandwich-type model for adhesive joints, proposed by Weißgraeber et al. [53].

3.1 Bigwood and Crocombe's method

Bigwood and Crocombe's model was developed in 1989 in response to the limitations seen in earlier analytical models. Prior models were restricted to specific joint configurations, necessitating the use of multiple computer implementations to determine the optimal configuration for a given application. Consequently, accurately predicting adhesive stress using these methods was challenging. To overcome these limitations, Bigwood and Crocombe's analytical model was developed, providing a versatile and reasonably accurate approach to designing bonded joints.

The analytical method developed by Bigwood and Crocombe et al. [52] is specifically designed to be applied to a wide range of joint configurations under various loading conditions. Figure 3.1 provides an illustration of the different joint configurations that are suitable for the application of this analytical method. The analysis focuses exclusively on the overlap region, where the adherend ends experience tensile and shear forces, as well as bending moments. To simplify the analysis, the joint is represented as a basic adhered-adhesive-adhered sandwich structure, as shown in Figure 3.2. It is worth noting that the model can be further expanded to incorporate material non-linearity, as discussed by Bigwood and Crocombe et al. [55].

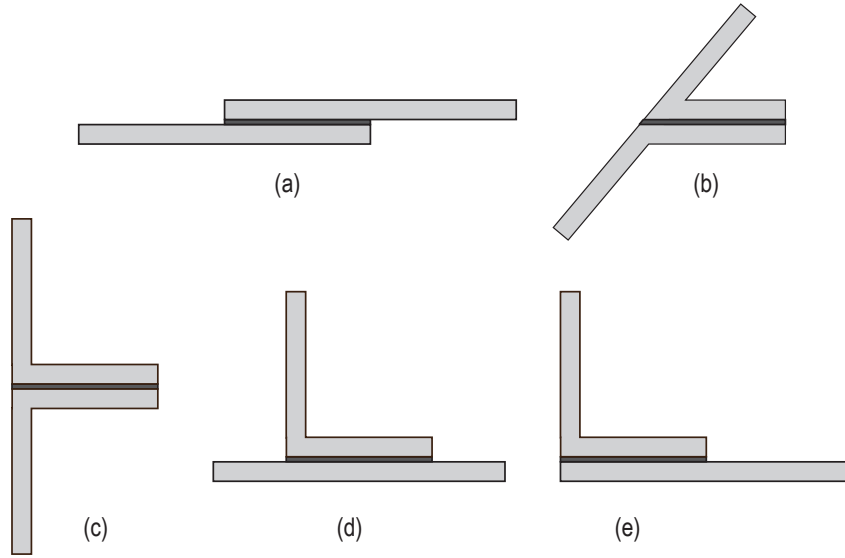


Figure 3.1: Adhesive joint configurations appropriate for study within Bigwood and Crocombe's method: (a) single lap joint, (b) inclined T-peel joint, (c) peel-joint, (d) L-joint, (e) corner-joint.

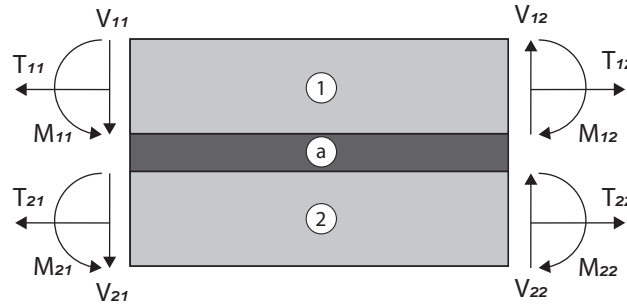


Figure 3.2: The general adherend-adhesive-adherend sandwich with general loading.

The presented analytical model provides a comprehensive analysis based on the assumption of full elastic behavior. It considers an elemental length of a general adherend-adhesive-adherend sandwich within the overlap region, as depicted in Figure 3.3. The model assumes that the ends are subjected to tensile, shear, and moment loading. The load is transmitted through the adhesive layer, which experiences transverse tension and shear. To establish equilibrium equations, the model assumes that the longitudinal direct stress in the adhesive is negligible compared to the corresponding stress in the adherends. The adherends are assumed to be isotropic and can possess varying thicknesses from one another, as long as these thicknesses remain constant throughout their length. Similarly, the adhesive is considered isotropic with a much smaller thickness than the

adherends. Under bending conditions, the adherends are handled as flat plates, assuming a state of plane stress in the $x-z$ plane and neglecting normal stresses along the thickness. Furthermore, a plane strain assumption is made in the $x-y$ plane to complete the analysis.

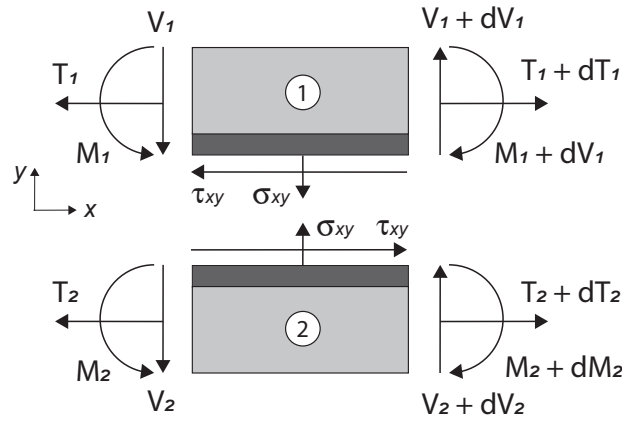


Figure 3.3: Free-body cut of an element of the adherend-adhesive-adherend sandwich under general loading.

Bigwood and Crocombe's modeling approach encompasses two distinct methods: the general and the simplified approach. The general approach considers coupled equations that determine the distribution of peel and shear stresses. It can be applied to analyze isotropic joints with similar or dissimilar adherends. Although it involves complex mathematical calculations, it offers greater versatility and accuracy, making it suitable for a wide range of applications. In contrast, the simplified approach provides a more straightforward analysis, as its name suggests. The simplified approach is generally suitable for isotropic adherends, but its applicability to joints with dissimilar adherends must be carefully analyzed for each particular case.

3.1.1 General elastic analysis

The equilibrium equations have been developed to ensure that the forces applied to an adhesive joint are in equilibrium with the adhesive stresses they produce. This allows for accurate predictions of the joint's behavior under different loading conditions. However, these equations rely on certain assumptions. First, it is assumed that the longitudinal direct stress in the adhesive is insignificant compared to the stress in the adherends. Additionally, the adherends and adhesive are assumed to be isotropic, with constant thicknesses. For this approach, the adherends may have different thicknesses.

This strategy is used to demonstrate that the differential equations describing the shear and transverse stress distributions in the adhesive layer have solutions of the general types as follows:

$$\begin{aligned}\tau_{xy} = & C_1 \cosh(m_1 x) + C_2 \sinh(m_1 x) + C_3 \cosh(n_1 x) \cosh(n_2 x) \\ & + C_4 \cosh(n_1 x) \sin(n_2 x) + C_5 \sinh(n_1 x) \cos(n_2 x) + C_6 \sinh(n_1 x) \sin(n_2 x) + C_7\end{aligned}\quad (3.1)$$

$$\begin{aligned}\sigma_y = & D_1 \cosh(m_1 x) + D_2 \sinh(m_1 x) + D_3 \cosh(n_1 x) \cos(n_2 x) \\ & + D_4 \cosh(n_1 x) \sin(n_2 x) + D_5 \sinh(n_1 x) \cos(n_2 x) + D_6 \sinh(n_1 x) \sin(n_2 x)\end{aligned}\quad (3.2)$$

It should be noted that the 13 constants (C_1 – C_7 and D_1 – D_6) must be calculated in order to fully solve the problem.

3.1.2 Simplified analysis

3.1.2.1 Simplified peel analysis

The simplified analysis evaluates the scenario of dissimilar adherends subjected to general tensile, shear, and bending moment loading per unit width while in a state of plane strain in the x – y plane. Initially, the assumption is made that the shear stress in the adhesive layer, τ_{xy} , does not affect the flexural behavior of the adherends, and thus is neglected. Alternatively, it can be interpreted that the adhesive functions solely as a series of tensile springs, with no shear load transmission taking place.

The method used in the prior case is employed, but with the assumption that the adhesive shear is zero. This results in the development of a differential equation that describes the peel stress experienced by the adhesive layer.

$$\frac{d^4 \sigma_y}{dx^4} + 4K_5^4 \sigma_y = 0 \quad (3.3)$$

where:

$$K_5^4 = \frac{E_a}{4t} \left(\frac{1}{D_1} + \frac{1}{D_2} \right) \quad (3.4)$$

The generic solution to the differential equation mentioned above can be used to fully describe the transverse stress distribution in the adhesive layer, as follows:

$$\begin{aligned}\sigma_y = & A_1 \cos(K_5 x) \cosh(K_5 x) + A_2 \cos(K_5 x) \sinh(K_5 x) \\ & + A_3 \sin(K_5 x) \cosh(K_5 x) + A_4 \sin(K_5 x) \sinh(K_5 x)\end{aligned}\quad (3.5)$$

where A_1 – A_4 and K_5 are the elastic explicit constants for the full simplified peel stress analysis, and their detailed calculation can be found in Appendix B.

The peak stress reveals itself at the end of the overlap where the load is applied. Considering that the load is applied at the left end, the transverse stress equation is given by:

$$\sigma_y = A_1 \quad (3.6)$$

To further simplify, it is possible to analyze each type of loading separately. In order to accomplish it, two additional variables known as peel compliance factors are included, which indicate the relative stiffness of the adhesive and adherends. They are defined as:

$$\beta_1 = \frac{12E_a(1-\mu_1^2)}{E_1h_1^3t} \quad [\text{mm}^{-4}] \quad (3.7)$$

$$\beta_2 = \frac{12E_a(1-\mu_2^2)}{E_2h_2^3t} \quad [\text{mm}^{-4}] \quad (3.8)$$

Assuming all the aforementioned conditions and considering peel compliance factors, it is feasible to calculate compressive transverse stress. Equation 3.9 is produced by the shear force per unit width, denoted by V , in the positive direction to the upper adherend, while Equation 3.10 is produced by a separate positive bending moment per unit width, M , applied at the same point. The total peak stress is estimated by considering the superposition of the component peak stresses.

$$\sigma_V = \frac{-(2)^{0.5}\beta_1V}{(\beta_1 + \beta_2)^{0.75}} \quad (3.9)$$

$$\sigma_M = \frac{-\beta_1M}{(\beta_1 + \beta_2)^{0.5}} \quad (3.10)$$

3.1.2.2 Simplified shear analysis

Simplified shear analysis is a method that can be utilized to determine the distribution of longitudinal shear stress in an adhesive layer. Within this simplification, the adhesive transverse stress is disregarded and only the in-plane deformation of the adhesive is taken into account. The general analysis method is employed. The distribution of shear stress in the adhesive layer is described by:

$$\frac{d^3\tau_{xy}}{dx^3} - K_6^2 \frac{d\tau_{xy}}{dx} = 0 \quad (3.11)$$

where:

$$K_6^2 = \frac{4G_a}{t} \left(\frac{(1-\mu_1^2)}{E_1h_1} + \frac{(1-\mu_2^2)}{E_2h_2} \right) \quad (3.12)$$

The shear transverse stress distributions in the adhesive layer are defined as:

$$\tau_{xy} = B_1 \cosh(K_6x) + B_2 \sinh(K_6x) + B_3 \quad (3.13)$$

where B_1 - B_3 and K_6 are the elastic explicit constants for the full simplified shear stress analysis, and their detailed calculation can be found in Appendix B.

Similar to the simplified peel analysis, additional simplifications are applied to develop design expressions for calculating the peak shear stress at the end of the adhesive overlap. These equations take into account the specific type of applied loading. The simplifications make it possible to calculate the peak shear stress at the adhesive overlap's end. The peak shear stress component induced for a tensile force per unit width, T , at the end of the overlap is given by:

$$\tau_T = \frac{-\alpha_1 T}{2(\alpha_1 + \alpha_2)^{0.5}} \quad (3.14)$$

where α_1 and α_2 are shear compliance factors which are a measure of the relative shear stiffness of the adhesive and the stiffness of the adherends. They are determined as:

$$\alpha_1 = \frac{G_a(1 - \mu_1^2)}{E_1 h_1 t} \quad [\text{mm}^{-2}] \quad (3.15)$$

$$\alpha_2 = \frac{G_a(1 - \mu_2^2)}{E_2 h_2 t} \quad [\text{mm}^{-2}] \quad (3.16)$$

The shear stress produced for a unit width shear force load, V , is obtained by:

$$\tau_V = \frac{3V}{4h_1} \quad (3.17)$$

While the shear stress produced for a unit width bending moment, M , applied at the same point, is given by:

$$\tau_M = \frac{3\alpha_1 M}{h_1(\alpha_1 + \alpha_2)^{0.5}} \quad (3.18)$$

The peak shear stress produced by any combination of tensile, shear, and bending moment loading can be determined using the superposition of the stress components.

3.1.3 Application of the model

Bigwood and Crocombe's method was employed for the development of this thesis, specifically utilizing the simplified analysis. The study focused on pine wood substrates bonded with a novel polyurethane bio-based adhesive. Three joint configurations were simulated: SLJs, L-joints, and T-joints. All of these joints were studied under tension loading. After implementing the analytical model, an FEA was performed in order to compare and validate the results obtained.

One of the main challenges encountered in applying the model was the requirement of prior knowledge regarding the internal forces and moments within the overlap region. Typically, only the global external forces or moments applied to the joint are known. To address this issue, the linear statics approach was employed as previously done by Liu et al. [56] and Weißgraeber et al. [53]. This approach considers that the structure is a non-deformable body and that it is in equilibrium. To establish that principle, the sum of the forces and moments acting on the structure

must be equal to zero. Therefore the reaction forces in the supports are found through the following equations:

$$\sum F_x = 0 \quad (3.19)$$

$$\sum F_y = 0 \quad (3.20)$$

$$\sum M_z = 0 \quad (3.21)$$

where x represents the horizontal direction, y the vertical direction and z the out-of-plane direction as in the Cartesian coordinate system. Once the forces and moments relevant to the problem are all determined, the internal forces within the region of interest (e.g. overlap edges) can be obtained through the analysis of isolated sections of the structure.

However, it is important to note that this approach has certain limitations, namely, neglecting nonlinear effects resulting from large bending deformations of the adherend, which can alter load-lines. To capture the nonlinear effects of axially loaded SLJs, different approaches that use moment factors have been proposed [46, 57, 58, 59]. In the present work, the analysis of the SLJ incorporates the utilization of bending moment factors, as described by Zhao et al. [59]. For the remaining cases, the linear statics approach is employed as a means of obtaining the internal forces and moments in the edges of the overlap region.

Moreover, the stress distributions within the adhesive joint were accurately predicted through a combination of analytical and numerical techniques. Emphasis was placed on evaluating the peel and shear stresses, as these factors play a crucial role in assessing the strength of the joints. The objective was to compare the results and, in the case of similar outcomes, validate the analytical approach for future use in the preliminary design of adhesively bonded structures with bio-materials.

Regarding the peel stress distribution, it was observed that the comparison was not very accurate. This discrepancy must be attributed to the specific characteristics of the simulated bio-adhesive, which cures at zero thickness. This factor greatly impacts the overall results, particularly for the peel stress distribution. Despite the relatively inaccurate peel stress distribution in the bio-adhesive joints under consideration, the critical stresses located at the edges of the overlap were found to be very similar. This demonstrates the effectiveness of the analytical method in predicting joint behavior. On the other hand, for the analysis of shear stress distributions, the results were quite similar. For more comprehensive details, the study can be referred to in Appendix D.

Abaqus/CAE customization

Abaqus plugins offer a powerful solution for customizing *Abaqus/CAE*, enabling efficient execution of repetitive tasks. These tasks can encompass both pre-processing and post-processing activities, which are very time-consuming. By utilizing plugins, it is possible to automate these tasks, aiming for productivity and saving valuable time.

Plugins utilize the *Abaqus Scripting Interface* and *Abaqus GUI Toolkit* commands to allow customization of *Abaqus/CAE* according to the user's requirements. These plugins come in two types:

- *Kernel* plugins: These plugins consist of functions developed using the *Abaqus Scripting Interface*. They enable customization by executing scripting commands directly within the *Abaqus kernel*.
- *GUI* plugins: These plugins are built using the *Abaqus GUI Toolkit* and are responsible for creating user interfaces that interact with the *kernel*. They facilitate customization by generating an interface and sending commands to the *kernel* as needed.

Within the scope of this thesis, a user-friendly *Abaqus* plugin was developed to enable the quick and convenient construction of multiple models, with a specific focus on adhesively bonded L-joint configurations. A simple *GUI* plugin displays a dialogue box (Figure 3.4) to enter the geometric parameters, loading conditions, and desired mesh size for an L-joint configuration. The mesh incorporates refinements at the overlap edges and adhesive area, allowing the user to adjust the mesh intensity as needed (Figure 3.5 (a)). When the *OK* button is pressed, the dialog box is closed, and the *GUI* plugin constructs a command that is sent to the *kernel* for execution. The *kernel* function builds the model using the information entered in the dialog box. The model is then generated and includes the corresponding geometry, material properties, sections, sets, boundary conditions (Figure 3.5 (b)), steps, and meshing (Figure 3.5 (a)). It is worth mentioning that the materials used in this study were specifically defined to explore bio-based joints with wood substrates.

Once the necessary configurations are complete, the job initiates automatically, and, upon completion, it starts the post-processing operations. It exports the stress distributions in the adhesive to an *Excel* file, encompassing both peel and shear stresses. Additionally, a new text file is generated on the computer, containing the internal forces and moments at the ends of the overlap for both adherends. The main *Python* scripting build for this study can be found in Appendix E.

Initially, the main goal was to access the internal forces and moments within the ends of the overlap region. The primary step was to conduct a thorough investigation into how the geometric parameters (e.g. total length, overlap length, adherend thickness, adhesive thickness, etc) and variations in loading affected the internal forces and moments. Through this investigation, it was observed that certain factors, such as adhesive thickness and the vertical length of the upper adherend (outside the overlap region), had no significant influence on the internal forces and moments within the region of interest. Furthermore, the remaining aspects either had a negligible effect on some of the internal forces and moments or exhibited a linear correlation with changes in length or intensity. Based on the findings of this study, it was concluded that the linear statics approach was appropriate for accurately identifying the internal forces and moments within the structure in the specific areas of interest.

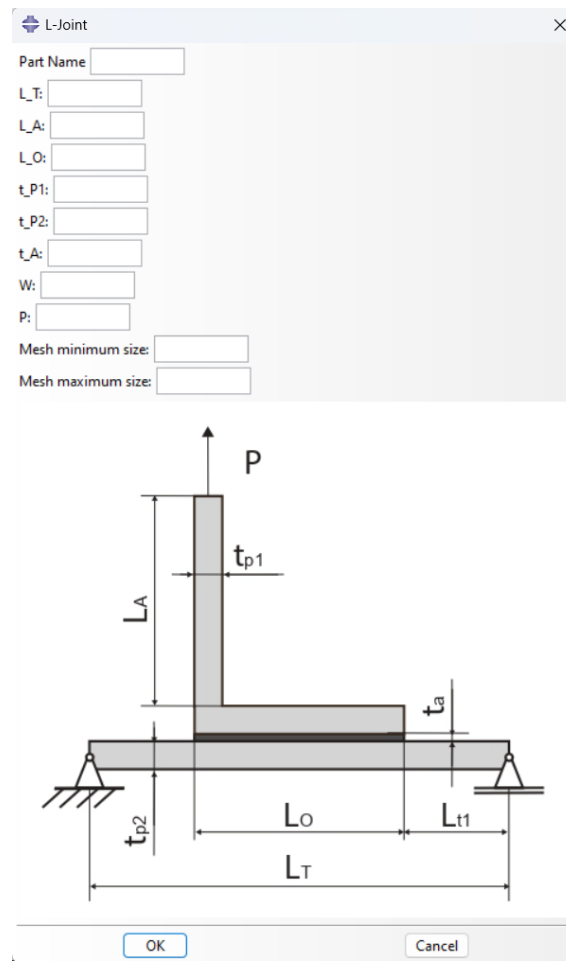
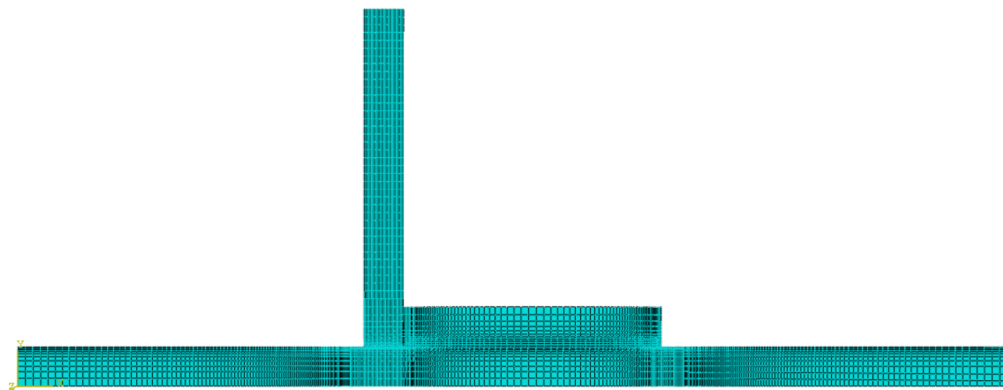


Figure 3.4: Abaqus plugin for L-joint configuration.

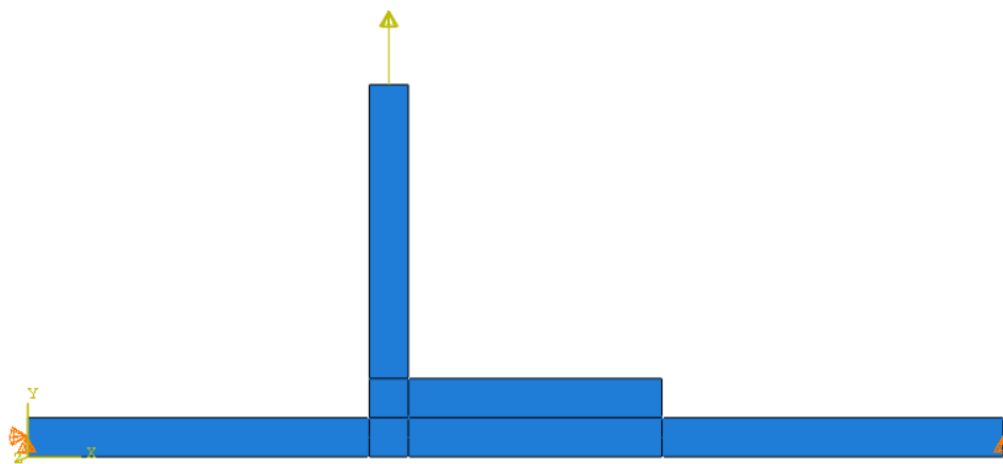
3.2 General sandwich-type model

The general sandwich-type model for adhesive joints with composite adherends approach was proposed by Weißgraeber et al. [53], in 2014. Prior to that point, a comprehensive and efficient model for stress analysis in bonded joints with laminated adherends had not yet been developed. The model is based on the well-established methodology pioneered by Bigwood and Crocombe et al. [52]. However, certain modifications have been incorporated into the model, which are exposed in this section.

The analysis adopts a two-dimensional approach, utilizing a general sandwich-type model that is well-suited for studying adhesive joints under the plane-strain assumption. This model, similar to the approach proposed by Bigwood and Crocombe, focuses on studying the overlap region of an adhesive joint. Consequently, it can effectively analyze various joint configurations, as outlined in Figure 3.1 for the previous model. In addition, this general sandwich-type model introduces additional joint geometries: double lap joint, inclined-joint, and patch-joint (Figure 3.6).



(a)



(b)

Figure 3.5: *Abaqus* plugin results: (a) mesh, (b) boundary conditions.

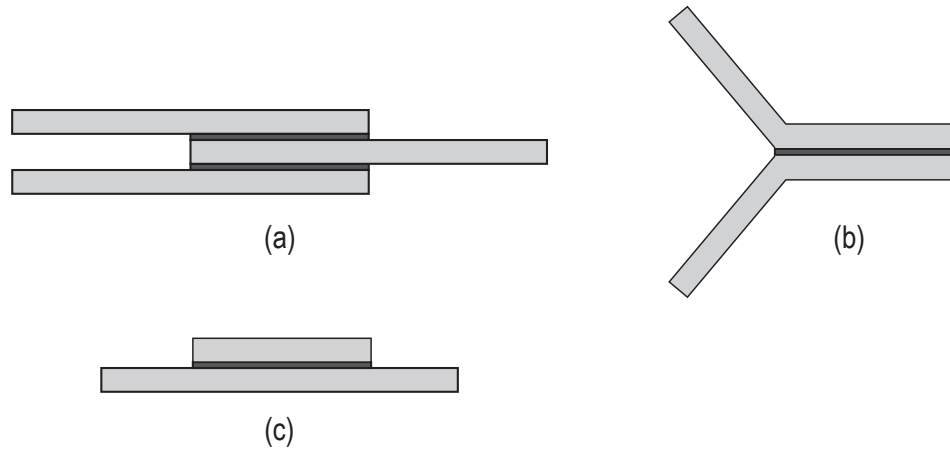


Figure 3.6: Additional adhesive joint configurations suitable for study within the general sandwich-type model: (a) double lap joint, (b) inclined-joint, (c) patch-joint.

Similar to the previous method, this general sandwich-type model focuses on an adherend-adhesive-adherend sandwich configuration to analyze the overlap region. However, it employs a different general loading convention, where the positive directions for transverse and moments applied to the overlap edges are reversed compared to the previous model (Figure 3.7).

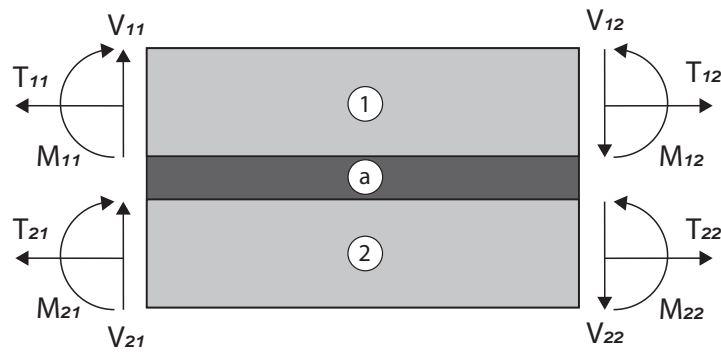


Figure 3.7: The general adherend-adhesive-adherend sandwich with general loading considered for the general sandwich-type model.

The free-body cut of an element with length dx of the sandwich follows a similar approach as seen in the previous analytical method (Figure 3.3). It is important to note that in this analytical model, the positive directions for shear forces (V), moments (M) and shear stress (τ_{xy}) are reversed compared to the previous model. The shear stress $\tau(x)$ and peel stress $\sigma(x)$ are considered in the

adhesive layer. Equilibrium equations of force and moment arise from the analysis of the free-body cut. The analysis of the free-body cut leads to equilibrium equations that relate the internal forces and moments with the shear and peel stresses and also the thickness of both adherends and adhesive. One notable distinction of this method compared to the previous one is the inclusion of the adhesive thickness in the equilibrium equations, whereas the previous model disregarded it.

In situations where the adherends possess relatively low transverse and normal stiffness, it becomes crucial to consider the shear deformation of the adherends. The conventional Kirchhoff-Love assumptions, such as those employed in the Classical Laminated Plate Theory (CLPT), may not be adequate to capture the deformations of such adherends. Thus, the First Order Shear Deformation Theory (FSDT) was utilized to accurately model the behavior of the adherends for this analytical model [60].

The plane-strain condition is assumed, allowing for the relationship between section forces and moments with the mid-plane strain and curvature of the adherends to be established as follows:

$$\begin{bmatrix} N \\ M \end{bmatrix} = \begin{bmatrix} A_{11} & B_{11} \\ B_{11} & D_{11} \end{bmatrix} \begin{bmatrix} u'_0 \\ \psi' \end{bmatrix} \quad (3.22)$$

$$V = kA_{55}\gamma_{xz} = kA_{55}(w'_0 + \psi) \quad (3.23)$$

where A_{11} is the extensional stiffness, B_{11} is the bending-extension coupling stiffness, D_{11} the flexural rigidity, A_{55} is the shear stiffness and k denotes the shear correction factor. The determination of these parameters is exposed in Section 3.2.1.

The thin adhesive layer is modeled as a weak interface and the shear stress in the adhesive layer is influenced by the deformations of the adherends, as indicated by Equation 3.24. When the adhesive layer is in a plane-strain state with negligible strain ϵ_x , the constitutive relation for the peel stress in the adhesive layer can be described by Equation 3.25.

$$\tau_0 = G_a \left(\frac{u_2 - u_1}{t} + \frac{1}{2} (w'_{0,2} + w'_{0,1}) \right) \quad (3.24)$$

$$\sigma_0 = E_a^* \frac{w_{0,2} - w_{0,1}}{t} \quad (3.25)$$

where the Young's modulus for plane-strain is given by:

$$E_a^* = \frac{E_a(1 - \nu_a)}{(1 + \nu_a)(1 - 2\nu_a)} \quad (3.26)$$

The governing coupled differential equation for the peel and shear stress in the adhesive layer can be expressed as follows:

$$\tau_0''' - \alpha_1 \tau_0' = -\alpha_2 \sigma_0 + \alpha_3 \sigma_0'' \quad (3.27)$$

where:

$$\alpha_1 = \frac{Ga}{t} \left(\frac{D_{11}^{(2)} + (h_2 + t)B_{11}^{(2)} + \frac{(h_2 + t)^2}{4}A_{11}^{(2)}}{\Delta^{(2)}} + \frac{D_{11}^{(1)} - (h_1 + t)B_{11}^{(1)} + \frac{(h_1 + t)^2}{4}A_{11}^{(1)}}{\Delta^{(1)}} \right) \quad (3.28)$$

$$\alpha_2 = \frac{Ga}{t} \left(\frac{1}{\Delta^{(2)}} \left(B_{11}^{(2)} + \frac{h_2 + t}{2}A_{11}^{(2)} \right) + \frac{1}{\Delta^{(1)}} \left(B_{11}^{(1)} - \frac{h_1 + t}{2}A_{11}^{(1)} \right) \right) \quad (3.29)$$

$$\alpha_3 = \frac{Ga}{2} \left(\frac{1}{k^{(2)}A_{55}^{(2)}} - \frac{1}{k^{(1)}A_{55}^{(1)}} \right) \quad (3.30)$$

In turn, the second coupled differential equation for the peel and shear stress in the adhesive layer is given by:

$$\sigma_0'''' - \beta_1 \sigma_0'' + \beta_2 \sigma_0 = \beta_3 \tau_0' \quad (3.31)$$

where:

$$\beta_1 = \frac{E_a^*}{t} \left(\frac{1}{k^{(1)}A_{55}^{(1)}} + \frac{1}{k^{(2)}A_{55}^{(2)}} \right) \quad (3.32)$$

$$\beta_2 = \frac{E_a^*}{t} \left(\frac{A_{11}^{(1)}}{\Delta^{(1)}} + \frac{A_{11}^{(2)}}{\Delta^{(2)}} \right) \quad (3.33)$$

$$\beta_3 = \frac{E_a^*}{t} \left(\frac{1}{\Delta^{(1)}} \left(B_{11}^{(1)} - \frac{h_1 + t}{2}A_{11}^{(1)} \right) + \frac{1}{\Delta^{(2)}} \left(B_{11}^{(2)} + \frac{h_2 + t}{2}A_{11}^{(2)} \right) \right) \quad (3.34)$$

The coupled differential equations for the shear and peel stress (Equations 3.27 and 3.31) constitute a system of seventh-order ordinary differential equations with constant coefficients. To solve the equations, some appropriate dimensionless variables and parameters are added along with a system matrix, as follows:

$$\vec{z} = \left[\tau_0 \quad \tau_0' \quad \tau_0'' \quad \sigma_0 \quad \sigma_0' \quad \sigma_0'' \quad \sigma_0''' \right]^T \quad (3.35)$$

$$\mathbf{A} = \begin{bmatrix} 0 & 1 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 1 & 0 & 0 & 0 & 0 \\ 0 & \alpha_1 & 0 & -\alpha_2 & 0 & \alpha_3 & 0 \\ 0 & 0 & 0 & 0 & 1 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 1 \\ 0 & \beta_3 & 0 & -\beta_2 & 0 & \beta_1 & 0 \end{bmatrix} \quad (3.36)$$

Finally, a system of seven differential equations of the first order is found:

$$\vec{z}' = \mathbf{A}\vec{z} \quad (3.37)$$

The system matrix $\mathbf{A}$ has a total of seven eigenvalues. These eigenvalues consist of a single zero eigenvalue, as well as six eigenvalues that can be either real or complex, depending on the specific joint configuration being considered. The complete solution of the differential Equation (3.37) can be expressed as a linear combination of solutions associated with these eigenvalues.

The determination of the seven boundary conditions is crucial for determining the free constants of the solution. These boundary conditions are derived by considering the horizontal equilibrium of forces (Equation 3.38) and ensuring kinematic compatibility at the ends of the overlap region (Equations 3.39, 3.40, 3.41). It is important to note that the six boundary conditions related to kinematic compatibility are obtained only by three equations that must be satisfied at each end of the overlap. More specifically, for the left side $x = -L_O/2$ along with index $i = 1$, and for the right side $x = L_O/2$ with index $i = 2$.

$$b \int_{-L_O/2}^{L_O/2} \tau_0 dx = N_{11} - N_{12} \quad (3.38)$$

$$\begin{aligned} \tau_0'(x) = & \frac{G_a}{t} \left(\frac{1}{\Delta^{(2)}} \left(\left(D_{11}^{(2)} + \frac{h_2+t}{2} B_{11}^{(2)} \right) N_{2i} - \left(B_{11}^{(2)} + \frac{h_2+t}{2} A_{11}^{(2)} \right) M_{2i} \right) \right. \\ & + \frac{1}{\Delta^{(1)}} \left(\left(-D_{11}^{(1)} + \frac{h_1+t}{2} B_{11}^{(1)} \right) N_{1i} - \left(-B_{11}^{(1)} + \frac{h_1+t}{2} A_{11}^{(1)} \right) M_{1i} \right) \\ & \left. + \frac{G_a}{2} \left(\frac{1}{k^{(2)}A_{55}^{(2)}} - \frac{1}{k^{(1)}A_{55}^{(1)}} \right) \sigma_0(x), \right. \end{aligned} \quad (3.39)$$

$$\begin{aligned} \sigma_0''(x) = & \frac{E_a^*}{t} \left(\frac{1}{\Delta^{(1)}} \left(-B_{11}^{(1)} N_{1i} + A_{11}^{(1)} M_{1i} \right) - \frac{1}{\Delta^{(2)}} \left(-B_{11}^{(2)} N_{2i} + A_{11}^{(2)} M_{2i} \right) \right) \\ & + \frac{E_a^*}{t} \left(\frac{1}{k^{(1)}A_{55}^{(1)}} + \frac{1}{k^{(2)}A_{55}^{(2)}} \right) \sigma_0(x) \end{aligned} \quad (3.40)$$

$$\begin{aligned} \sigma_0'''(x) = & \frac{E_a^*}{t} \left(\frac{1}{\Delta^{(1)}} \left(B_{11}^{(1)} - \frac{h_1+t}{2} A_{11}^{(1)} \right) + \frac{1}{\Delta^{(2)}} \left(B_{11}^{(2)} + \frac{h_2+t}{2} A_{11}^{(2)} \right) \right) \tau_0(x) \\ & + \frac{A_{11}^{(1)}}{\Delta^{(1)}} V_{1i} - \frac{A_{11}^{(2)}}{\Delta^{(2)}} V_{2i} + \left(\frac{1}{k^{(1)}A_{55}^{(1)}} + \frac{1}{k^{(2)}A_{55}^{(2)}} \right) \sigma_0'(x) \end{aligned} \quad (3.41)$$

3.2.1 Material properties

As previously discussed, the general sandwich-type model offers numerous advantages, including exceptional versatility in modeling various material types, from single-layer to complex laminated

plates. This model is well-suited for studying isotropic materials, specially orthotropic, general orthotropic, and anisotropic materials, making it applicable to a wide range of materials. Additionally, it can study both symmetric and asymmetric laminates effectively.

In the present analytical model, the calculation of certain material properties is required, which vary depending on the material type. These properties are then incorporated into Equations 3.22 and 3.23. This section presents the mathematical formulations for determining these properties, as exposed by Reddy et al. [60].

3.2.1.1 General considerations

The arrangement of plies in a laminate is commonly represented as $(\alpha / \beta / \gamma / \dots)$, where α represents the orientation of the first ply, β represents the orientation of the second ply, and so on. The plies are numbered in the positive z direction, with an upward orientation. It should be noted that, unless stated otherwise, this notation assumes that all layers have the same thickness and are made of the same material.

There are two types of stacking sequences: symmetric and asymmetric. In a symmetric laminate, the ply stacking sequence, material composition, and ply thicknesses exhibit symmetry concerning the midplane of the laminate (Figure 3.8(a)). Conversely, an asymmetric laminate lacks this symmetry (Figure 3.8(b)).

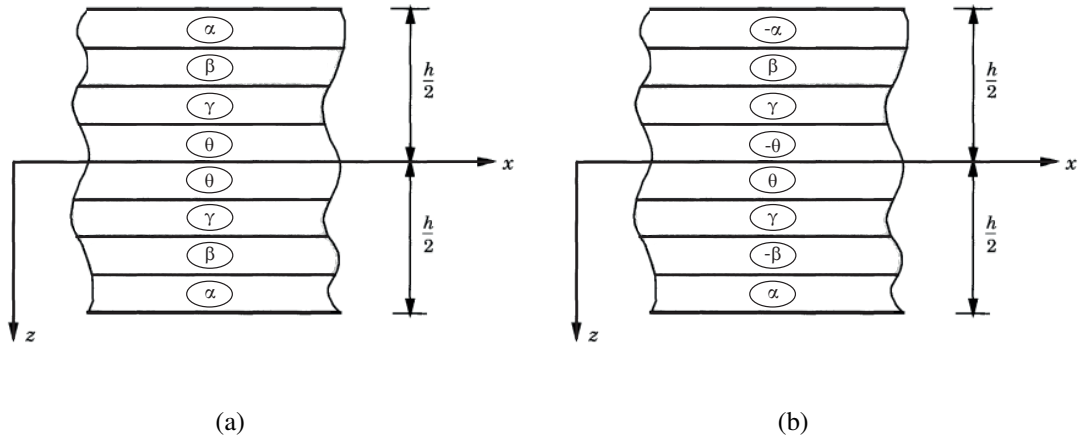


Figure 3.8: Laminates: (a) generic symmetric, (b) example of asymmetric adapted from [60].

Furthermore, some terminology related to laminates should be defined. A general laminate refers to a structure composed of layers with various orientations denoted by θ , where $-90^\circ \leq \theta \leq 90^\circ$. The term general angle-ply laminates consist of ply orientations between $0^\circ \leq \theta \leq 90^\circ$, with at least one layer having an orientation other than 0° or 90° (e.g. $(15/-30/0/90/45/-45)$). Cross-ply laminates are characterized by ply orientations of 0° or 90° (e.g. $(0/90/90/0/0/90)$).

The stiffnesses of a laminate, denoted by A_{ij} , depend only on the thicknesses and stiffnesses of the individual layers. They are not affected by the arrangement of the layers within the laminate. Conversely, the stiffnesses denoted by D_{ij} are influenced not only by the layer thicknesses and

stiffnesses but also by their specific placement relative to the midplane of the laminate. Additionally, the stiffnesses denoted by B_{ij} are influenced by the layer thicknesses, stiffnesses, and their location relative to the midplane. It is important to note that these stiffnesses can vary and may even be negative, depending on the lamination scheme and the number of layers in the laminate.

3.2.1.2 Single-layer plates

Single-layer plates include isotropic, specially orthotropic, general orthotropic, and anisotropic adherends. Specially orthotropic adherends refer to the materials where the principal material coordinates coincide with those of the plate. On the other hand, general orthotropic adherends refer to the cases where the principal material coordinates do not coincide with those of the plate. It should be noted that for all single-layer plates, the bending coupling is zero, $B_{ij} = 0$.

Isotropic

Considering isotropic single-layer plates, the extensional stiffnesses (A_{ij}), the bending stiffnesses (D_{ij}), and the bending-extensional coupling stiffnesses (B_{ij}) are given by:

$$A_{11} = \frac{Eh}{1 - \nu^2} \quad (3.42)$$

$$B_{11} = 0 \quad (3.43)$$

$$D_{11} = \frac{Eh^3}{12(1 - \nu^2)} \quad (3.44)$$

$$A_{55} = \frac{Eh}{2(1 + \nu)} \quad (3.45)$$

where E is the Young's modulus, ν is the Poisson's ratio and h is the thickness of the adherend.

Specially orthotropic

For the case of specially orthotropic single-layer plates, the extensional stiffnesses (A_{ij}), the bending stiffnesses (D_{ij}), and the bending-extensional coupling stiffnesses (B_{ij}) are given by:

$$A_{11} = \frac{E_1 h}{1 - \nu_{12}\nu_{21}} \quad (3.46)$$

$$B_{11} = 0 \quad (3.47)$$

$$D_{11} = \frac{E_1 h^3}{12(1 - \nu_{12}\nu_{21})} \quad (3.48)$$

$$A_{55} = G_{13}h \quad (3.49)$$

General orthotropic

In the case of general orthotropic material, the procedure becomes more complex due to the requirement of considering the transformation between the principal material coordinates and the plate coordinate system. Starting by defining the material compliance parameters:

$$\mathbf{S} = \begin{bmatrix} S_{11} & S_{12} & S_{13} & 0 & 0 & 0 \\ S_{12} & S_{22} & S_{23} & 0 & 0 & 0 \\ S_{13} & S_{23} & S_{33} & 0 & 0 & 0 \\ 0 & 0 & 0 & S_{44} & 0 & 0 \\ 0 & 0 & 0 & 0 & S_{55} & 0 \\ 0 & 0 & 0 & 0 & 0 & S_{66} \end{bmatrix} \quad (3.50)$$

where:

$$S_{11} = \frac{1}{E_1}, \quad S_{12} = -\frac{\nu_{12}}{E_1}, \quad S_{13} = -\frac{\nu_{13}}{E_1} \quad (3.51)$$

$$S_{22} = \frac{1}{E_2}, \quad S_{23} = -\frac{\nu_{23}}{E_2}, \quad S_{33} = \frac{1}{E_3} \quad (3.52)$$

$$S_{44} = \frac{1}{G_{23}}, \quad S_{55} = \frac{1}{G_{13}}, \quad S_{66} = \frac{1}{G_{12}} \quad (3.53)$$

Q_{ij} , called the plane stress-reduced stiffnesses, are given by:

$$Q_{11} = \frac{S_{22}}{S_{11}S_{22} - S_{12}^2} = \frac{E_1}{1 - \nu_{12}\nu_{21}} \quad (3.54)$$

$$Q_{12} = \frac{S_{12}}{S_{11}S_{22} - S_{12}^2} = \frac{\nu_{12}E_2}{1 - \nu_{12}\nu_{21}} \quad (3.55)$$

$$Q_{22} = \frac{S_{11}}{S_{11}S_{22} - S_{12}^2} = \frac{E_2}{1 - \nu_{12}\nu_{21}} \quad (3.56)$$

$$Q_{66} = \frac{1}{S_{66}} = G_{12} \quad (3.57)$$

To facilitate the transformation between the principal material coordinates and the plate coordinate system, the utilization of the following transformation matrix is necessary:

$$\mathbf{T} = \begin{bmatrix} \cos^2(\theta) & \sin^2(\theta) & 0 & 0 & 0 & -\sin^2(2\theta) \\ \sin^2(\theta) & \cos^2(\theta) & 0 & 0 & 0 & \sin^2(2\theta) \\ 0 & 0 & 1 & 0 & 0 & 0 \\ 0 & 0 & 0 & \cos(\theta) & \sin(\theta) & 0 \\ 0 & 0 & 0 & -\sin(\theta) & \cos(\theta) & 0 \\ \sin(\theta)\cos(\theta) & -\sin(\theta)\cos(\theta) & 0 & 0 & 0 & \cos^2(\theta) - \sin^2(\theta) \end{bmatrix} \quad (3.58)$$

Subsequently, the transformation should be carried out to map the local plane stress-reduced stiffness to the global coordinate system. This transformation can be achieved using a transformation matrix, as follows:

$$\bar{Q} = TQT^T \quad (3.59)$$

Thus, the extensional stiffnesses (A_{ij}), the bending stiffnesses (D_{ij}), and the bending-extensional coupling stiffnesses (B_{ij}) for a single generally orthotropic layer are given by:

$$A_{ij} = \bar{Q}_{ij}h, \quad D_{ij} = \frac{\bar{Q}_{ij}h^3}{12}, \quad B_{ij} = 0 \quad (3.60)$$

for $i, j = 1, 2, 3, 4, 5$ and 6 .

Anisotropic

For a single anisotropic layer, the stiffnesses are expressed in terms of the coefficients (C_{ij}) and thickness (h) as follows:

$$A_{ij} = C_{ij}h, \quad D_{ij} = \frac{C_{ij}h^3}{12}, \quad B_{ij} = 0 \quad (3.61)$$

for $i, j = 1, 2, 3, 4, 5$ and 6 . The stiffness coefficients (C_{ij}) are given by:

$$C_{11} = \frac{S_{22}S_{33} - S_{23}^2}{S}, \quad C_{12} = \frac{S_{13}S_{23} - S_{12}S_{33}}{S}, \quad C_{13} = \frac{S_{12}S_{23} - S_{13}S_{22}}{S} \quad (3.62)$$

$$C_{22} = \frac{S_{33}S_{11} - S_{13}^2}{S}, \quad C_{23} = \frac{S_{12}S_{13} - S_{23}S_{11}}{S}, \quad C_{33} = \frac{S_{11}S_{22} - S_{12}^2}{S} \quad (3.63)$$

$$C_{44} = \frac{1}{S_{44}}, \quad C_{55} = \frac{1}{S_{55}}, \quad C_{66} = \frac{1}{S_{66}} \quad (3.64)$$

with:

$$S = S_{11}S_{22}S_{33} - S_{11}S_{23}^2 - S_{22}S_{13}^2 - S_{33}S_{12}^2 + 2S_{12}S_{23}S_{13} \quad (3.65)$$

3.2.1.3 Symmetric laminates

As stated previously, the term "symmetric laminate" refers to a laminate in which the qualities of the material, its locations, and the arrangement of layers are symmetrical about the mid-plane (Figure 3.8(a)). The symmetry of the material coefficients (Q_i), distances (Z_k), and thicknesses (h_k) for each layer about the mid-plane of the laminate results in zero coupling stiffness (B_{ij}) for symmetric laminates.

Isotropic layers

Symmetric laminates consist of multiple isotropic layers with potentially distinct material properties and thicknesses, all arranged symmetrically in terms of both their geometric configuration and material characteristics.

The laminate stiffnesses A_{ij} , B_{ij} and D_{ij} are defined by:

$$A_{ij} = \sum_{k=1}^N \bar{Q}_{ij}^{(k)} (z_{k+1} - z_k) \quad (3.66)$$

$$B_{ij} = \frac{1}{2} \sum_{k=1}^N \bar{Q}_{ij}^{(k)} (z_{k+1}^2 - z_k^2) \quad (3.67)$$

$$D_{ij} = \frac{1}{3} \sum_{k=1}^N \bar{Q}_{ij}^{(k)} (z_{k+1}^3 - z_k^3) \quad (3.68)$$

where:

$$\begin{aligned} \bar{Q}_{11}^{(k)} = \bar{Q}_{22}^{(k)} &= \frac{E^k}{1 - \nu_k^2}, & \bar{Q}_{16}^{(k)} = \bar{Q}_{26}^{(k)} &= 0, \\ \bar{Q}_{12}^{(k)} &= \frac{\nu_k E^k}{1 - \nu_k^2}, & \bar{Q}_{44}^{(k)} = \bar{Q}_{55}^{(k)} = \bar{Q}_{66}^{(k)} &= \frac{E^k}{2(1 + \nu_k)} \end{aligned} \quad (3.69)$$

General orthotropic

A laminate comprising multiple symmetrically arranged specially orthotropic layers demonstrates no coupling between and extension, denoted by $B_{ij} = 0$. The laminate stiffnesses A_{ij} and D_{ij} defined by Equations 3.66, 3.67 and 3.68. For this case, the plane stress-reduced stiffnesses are given by:

$$\begin{aligned} \bar{Q}_{11}^{(k)} &= \frac{E_1^k}{1 - \nu_{12}^k \nu_{21}^k}, & \bar{Q}_{12}^{(k)} &= \frac{\nu_{21}^k E_1^k}{1 - \nu_{12}^k \nu_{21}^k}, & \bar{Q}_{22}^{(k)} &= \frac{E_2^k}{1 - \nu_{12}^k \nu_{21}^k}, \\ \bar{Q}_{16}^{(k)} = \bar{Q}_{26}^{(k)} &= 0, & \bar{Q}_{66}^{(k)} &= G_{12}^k, & \bar{Q}_{44}^{(k)} &= G_{23}^k, & \bar{Q}_{55}^{(k)} &= G_{13}^k \end{aligned} \quad (3.70)$$

Multiple generally orthotropic layers

Laminates are commonly constituted by orthotropic layers, where the principal material directions and the principal plate axes make an angle, i.e. these layers are aligned with the laminate axes at an angle of θ degrees. If the layers in the laminate have symmetric thicknesses, locations, and material properties about the midplane, then the bending-extensional coupling stiffnesses is zero ($B_{ij} = 0$).

In this condition, the stiffness of the laminate, denoted as A_{ij} and D_{ij} , are determined resorting to Equations 3.66, 3.67, and 3.68 in combination with Equations 3.70. The procedure is analogous to that explained for multiple specially orthotropic layers, with the distinction that the angles involved are not aligned with the principal material axes.

3.2.1.4 Asymmetric laminates

While symmetric laminates are preferred for analysis purposes, they may not always meet the design requirements in certain applications. Asymmetric laminates might have an even number of orthotropic laminae when the adjacent laminae have equal thicknesses and alternating orientations ranging from 0° to 90° . These laminates maintain symmetry in the thicknesses of each layer pair. However, it is important to note that the coupling stiffnesses (B_{ij}) in these laminates are not necessarily zero. Nevertheless, as the number of layers increases, these coupling stiffnesses tend to approach zero.

A special case of asymmetric laminates is the asymmetric cross-ply laminates. It considers the case of an even number of orthotropic layers with principal material directions alternate between 0° and 90° with respect to the laminate axes. An example of an asymmetric cross-ply laminate is (0/90/90/0/0/90). In terms of thickness, they might have all the same or the layers having thicknesses ($h_1/h_2/h_3/h_3/h_2/h_1$). It is important to note that for every 0° layer of a given thickness and position, there is a corresponding 90° layer of the same thickness and position on the opposite side of the midplane. Regarding the coupling stiffnesses of these laminates, they possess the following properties: $B_{22} = -B_{11}$, and all other components are equal to zero.

In turn the asymmetric angle-ply laminates also have an even number of orthotropic layers, where the principal material directions alternate between θ on one side of the midplane and $-\theta$ on the other side. For specific case of $\theta = 0$, $-\theta$ should be interpreted as 90° . The opposite case is also verified. An example of these types of laminates might be (-45/40/-15/15/-40/45).

3.2.2 Numerical validation

This section presents a comparison between stress distributions obtained from the analytical model presented and numerical results of detailed FEA, resorting to *Abaqus 2022*[®] software. A two-dimensional (2D) Finite Element Model of the whole joint was simulated. The mesh consists of 4-node biquadratic plane strain quadrilateral elements (CPE4), with a high mesh density chosen to ensure convergence stresses in the mid-plane of the adhesive layer. The number of elements

employed across the thickness of the adhesive layer varied based on the thickness of the adhesive. However, a minimum of four layers of elements were consistently utilized.

In terms of material properties, modeling isotropic single-layer materials was considered straightforward. They were simulated as simple elastic isotropic materials with uniform properties. On the other hand, orthotropic single-layer materials were more complex because their properties vary in different directions. Therefore, a homogeneous anisotropic layer was used. When considering laminated adherends, two approaches were employed. The first approach entailed modeling each layer of the laminate separately as homogeneous anisotropic layers. This method was applicable only when the orientation of the laminae matched the principal directions of the model (e.g. 0 or 90 °). The second approach was employed for all other cases. In such cases, a transformation was required to convert the local laminate properties to global properties. The Classical Laminate Plate Theory (CLPT) was used to facilitate this transformation and model the laminated adherends as a single layer. The properties resulting from the CLPT were directly inserted into *Abaqus* for the effect. The *Python* code used to perform these calculations is provided in Appendix F.

The materials used for the present study are shown in Table 3.1, with their respective properties. Carbon-fiber-reinforced plastics (CFRP) were considered to model different laminated adherends. Additionally, Figure 3.9 displays the geometries selected to perform this study, which include SLJ, L-joint and patch-joint.

The analysis presents comparison results for different joint configurations with different materials. Each diagram shows the Finite Element Analysis (FEA) results as a solid black line, while the results from the present analytical model are represented by a dotted gray line. For the case of axially loaded single lap joints (SLJs) the bending moment factors approach is employed as described by Zhao et al. [59] in order to obtain the loading on the overlap region. For all the other cases linear statics modeling was utilized. This approach is considered sufficient for the purpose of comparing the analytical results, despite not accounting for nonlinear effects resulting from significant bending deformations of the adherends and the associated changes in load-lines.

Table 3.1: Material properties used in the comparison of the present analytical model with FEA.

	Young's modulus [GPa]	Poisson's ratio [-]	Shear modulus [GPa]
Steel	$E = 210$	$\nu = 0.30$	$G = 80.8$
Aluminium	$E = 70.0$	$\nu = 0.33$	$G = 26.3$
Pine wood	$E_1 = 12.0$	$\nu_{12} = 0.47$	$G_{12} = 1.12$
	$E_2 = 1.90$	$\nu_{13} = 0.47$	$G_{13} = 1.04$
	$E_3 = 1.10$	$\nu_{23} = 0.57$	$G_{23} = 0.29$
CFRP	$E_1 = 135$	$\nu_{12} = 0.27$	$G_{12} = 5.00$
	$E_2 = 10.0$	$\nu_{13} = 0.27$	$G_{13} = 5.00$
	$E_3 = 10.0$	$\nu_{23} = 0.27$	$G_{23} = 5.00$
Epoxy	$E_a = 3.90$	$\nu_a = 0.36$	$G_a = 1.43$
Bio-adhesive	$E_a = 0.197$	$\nu_a = 0.33$	$G_a = 0.0741$

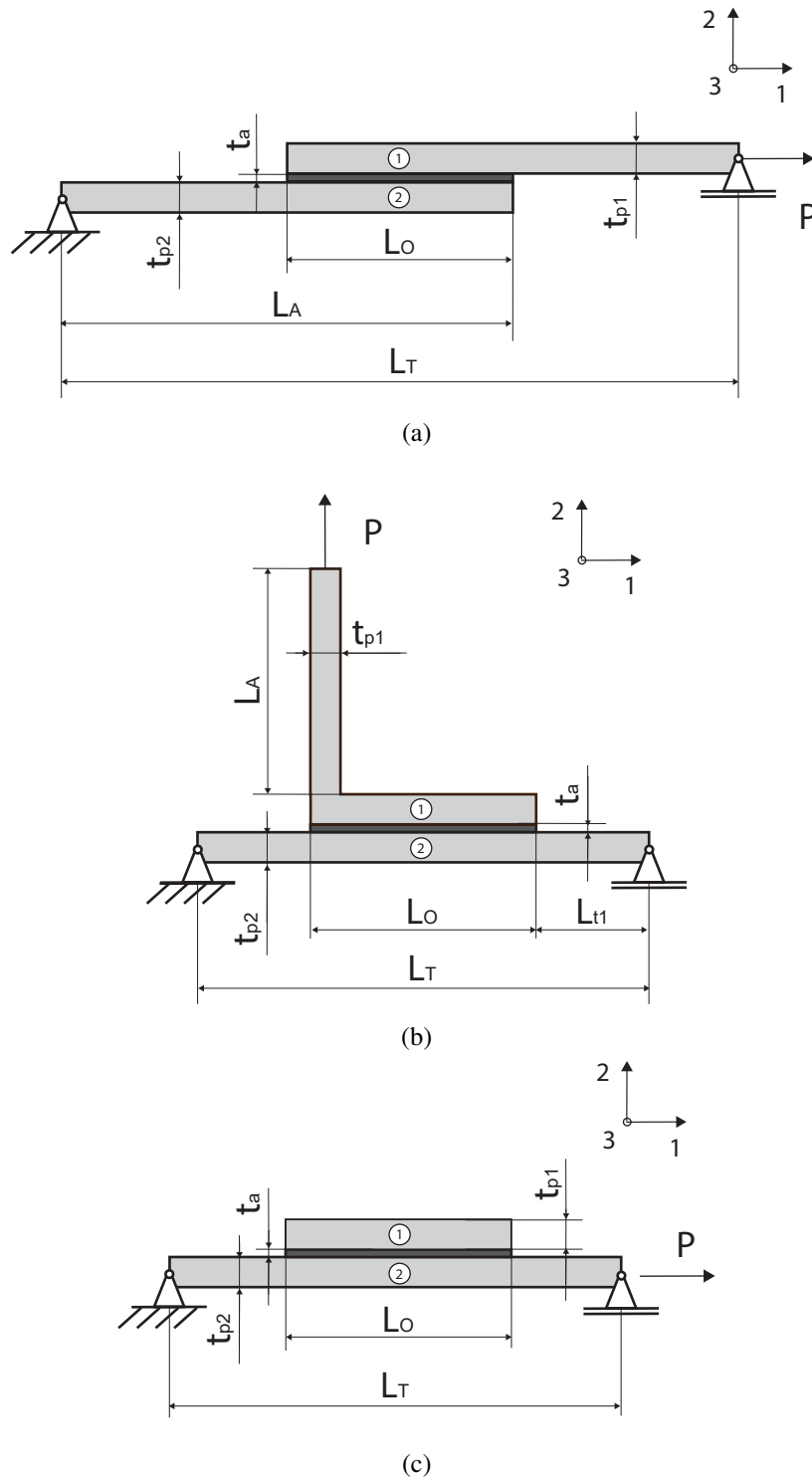


Figure 3.9: Geometries and boundary conditions: (a) SLJ, (b) L-joint, (c) patch-joint.

3.2.2.1 Single lap joint

For the case of the SLJ, three different cases were considered. The first one is the simplest case scenario with a symmetric joint and isotropic materials. Second, the case of a symmetric joint but

in this case with bio-based materials. Third, the case of a symmetric joint with different materials. The dimensions selected and loading conditions are exposed in Table 3.2.

Table 3.2: Joint geometry and loading conditions for SLJs.

	L_T [mm]	L_O [mm]	L_A [mm]	w [mm]	t_{p1} [mm]	t_{p2} [mm]	t_a [mm]	P [N]
Case 1	175	25	100	25	2	2	0.5	5000
Case 2	175	25	100	25	4	4	0.01	5000
Case 3	175	25	100	25	2	2	0.5	5000

Case 1: symmetric joint with isotropic materials

The materials selected for this study case consist of aluminum for the substrate and epoxy for the adhesive. Figure 3.10 presents the results obtained for both analyses. It is verified that for both cases the plots obtained present very good agreement.

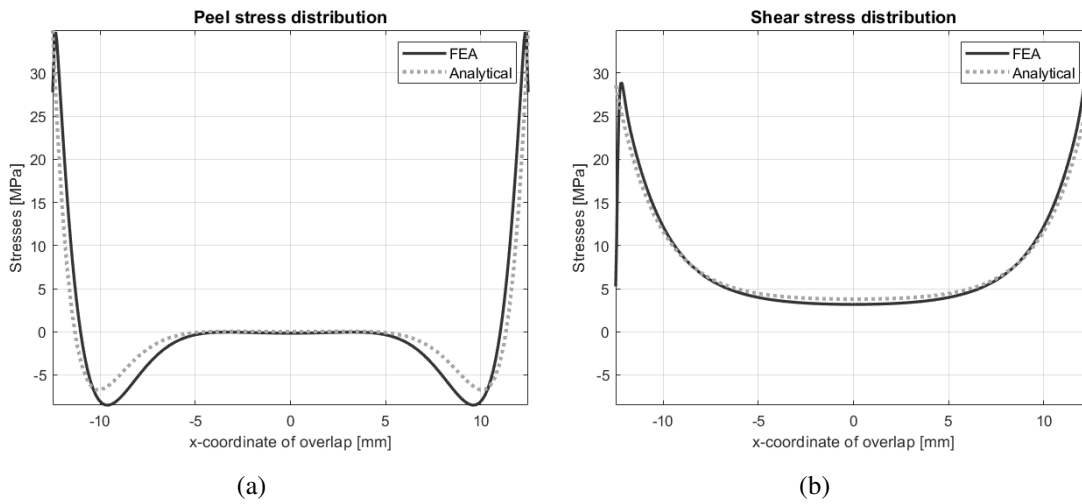


Figure 3.10: Stress distribution in a symmetric SLJ with aluminum adhered and epoxy: (a) peel stress, (b) shear stress.

Case 2: symmetric joint with bio-based materials

The materials chosen for this particular case study consist of wood for the substrate and a bio-adhesive. Figure 3.11 displays the obtained results for both peel and shear stress distributions. It is observed that the plotted stress distributions are not highly accurate. Despite that, for the peel stress distribution, the most critical stress obtained which are located at the ends of the overlap region, are very similar. However, throughout the overlap region, there are slight differences in the shape of the plots, although not very pronounced. Conversely, for the shear stress distribution,

there is a pronounced variation in the most critical values at the ends of the overlap region, but the overall shape of the plots is very similar. It is important to note that the analytical model yields higher stress values, indicating that it provides a conservative analysis. This confirms its suitability for preliminary joint design purposes.

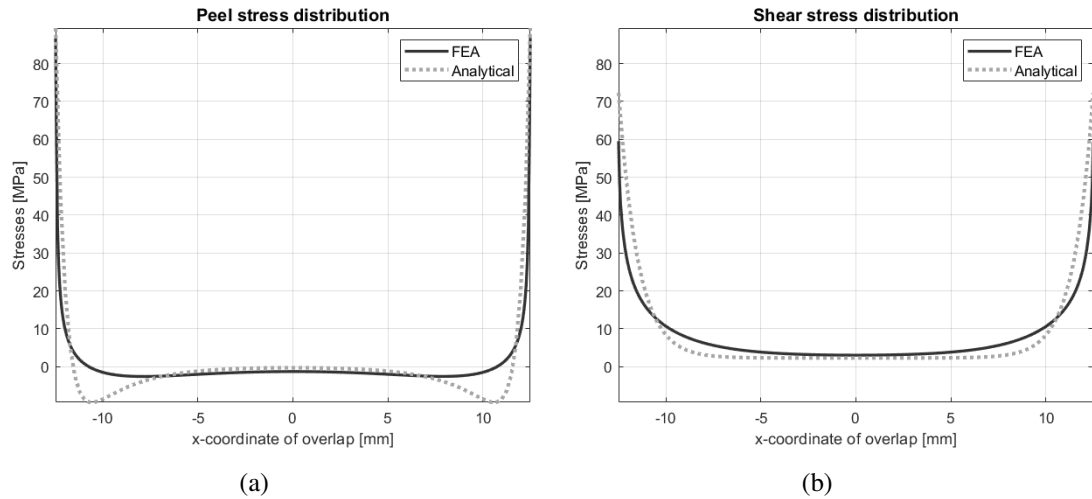


Figure 3.11: Stress distribution in a symmetric SLJ with pine wood adhered and bio-adhesive: (a) peel stress, (b) shear stress.

Case 3: symmetric joint with dissimilar materials

This third analysis presents a study of a joint composed of different substrates, the upper one CFRP with a stacking sequence of $[0/90]_s$ and the bottom one aluminum. Figure 3.12 presents the results obtained. It is verified a good agreement between the plots, even though not very accurate.

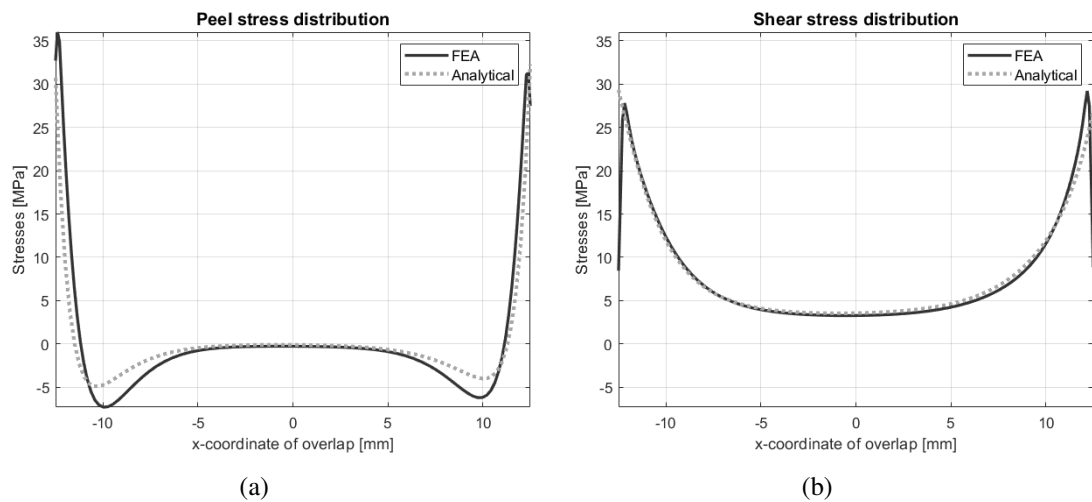


Figure 3.12: Stress distribution in a symmetric SLJ with CFRP and aluminum adhered and epoxy adhesive: (a) peel stress, (b) shear stress.

3.2.2.2 L-Joint

Considering the case of the L-joint, also three different scenarios were studied. The first one is a symmetric joint, with isotropic materials. Second, the case of a symmetric joint but in this case with bio-based materials. Third, the case of an asymmetric joint with different materials. The dimensions and loading conditions selected are exposed in Table 3.3.

Table 3.3: Joint geometry and loading conditions for L-joint.

	L_T [mm]	L_O [mm]	L_A [mm]	L_{t1} [mm]	w [mm]	t_{p1} [mm]	t_{p2} [mm]	t_a [mm]	P [N]
Case 1	100	30	30	35	25	4	4	0.5	750
Case 2	100	30	30	35	25	4	4	0.01	750
Case 3	100	30	30	35	25	4	4	0.2	750

Case 1: symmetric joint with isotropic materials

In this first case, an L-joint with substrates of equal thickness is examined. The substrates were simulated as aluminum, and the adhesive as epoxy. The obtained plots are displayed in Figure 3.13. Overall, there is a high level of agreement observed in the plots. However, it is worth noting that the peak stress values at the edges of the overlap region do not align perfectly in the peel stress analysis. Nevertheless, it is reassuring that the analytical model yields higher peak stress values, ensuring a conservative and safe design. The shear stress results appear to be accurate, showing good agreement between the analytical and numerical models.

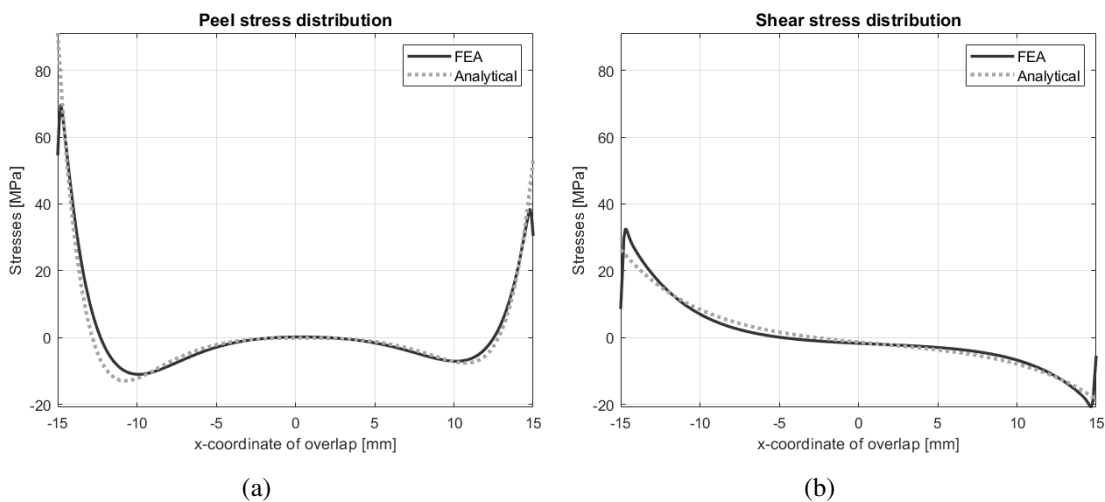


Figure 3.13: Stress distribution in a symmetric L-joint with aluminum substrate and epoxy adhesive: (a) peel stress, (b) shear stress.

Case 2: symmetric joint with bio-based materials

In Case 2, an L-joint with pine wood substrates of equal thickness and a bio-adhesive that cures at zero thickness is examined. The results obtained are presented in Figure 3.14. It is observed that there is a lack of agreement between the FEA and analytical model in terms of peel stress distributions and some inaccuracy regarding the peak peel stress. However, this discrepancy is not significant, and it is important to note that the analytical model provides conservative results. On the other hand, there is a high level of agreement in the shear stress distribution between the two models, indicating good accuracy.

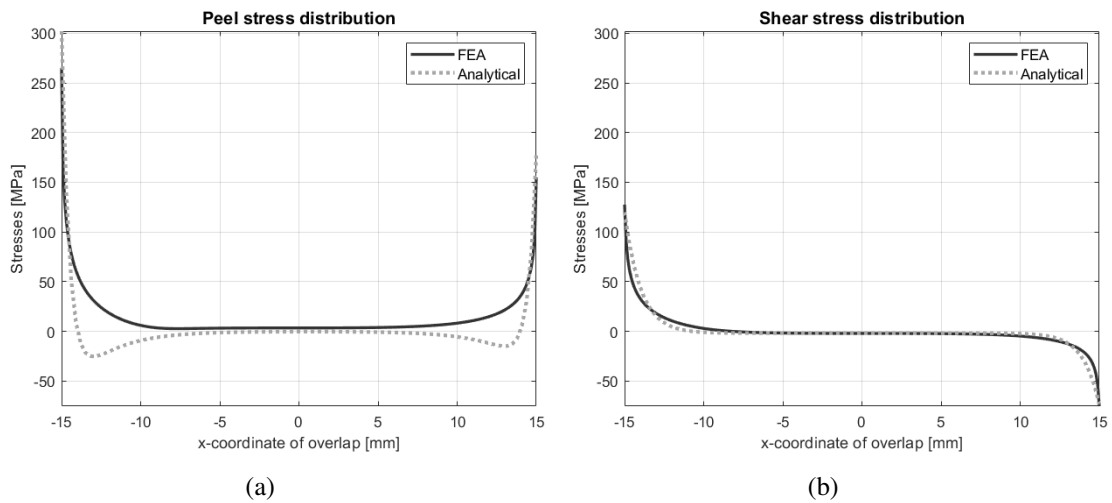


Figure 3.14: Stress distribution in a symmetric L-joint with pine wood adhered and bio-adhesive: (a) peel stress, (b) shear stress.

Case 3: symmetric joint with dissimilar materials

In Case 3, an L-joint with dissimilar materials but the same substrate thickness is investigated. The upper substrate was made of CFRP with a symmetric layup of $[45, -45, 0, 90]_s$, while the bottom substrate was composed of steel. The adhesive used in the simulation was an epoxy. The results obtained for this case are depicted in Figure 3.15. Upon analysis, it is evident that there is good agreement in the peak peel stress results on the right side of the joint. However, on the opposite side, there is less similarity between the FEA and analytical model outcomes. Despite this discrepancy, it is worth noting that the analytical model still provides valuable insights. Once again, the shear stress distribution demonstrates a high level of agreement between the FEA and analytical model, indicating accurate results.

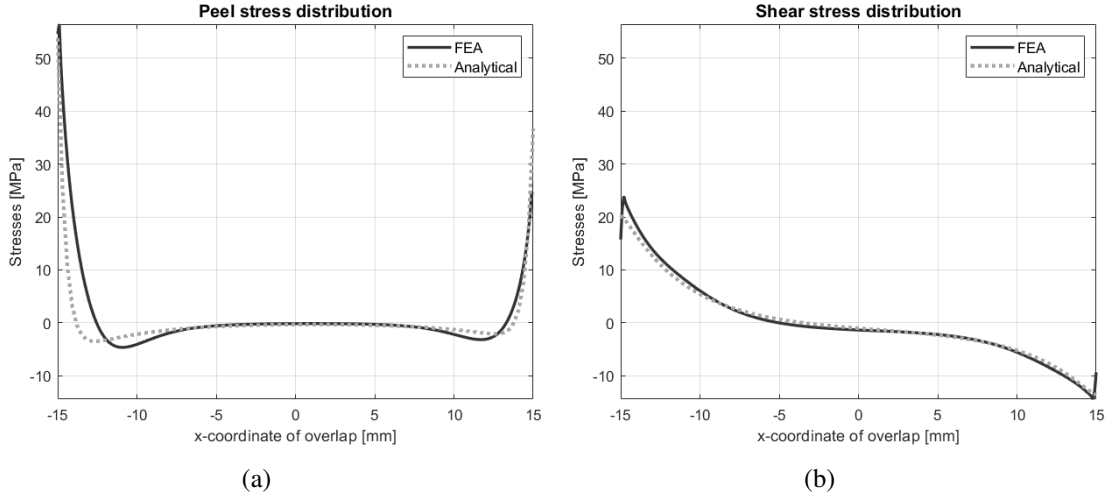


Figure 3.15: Stress distribution in a symmetric L-joint with CFRP $[45, -45, 0, 90]_s$, steel and epoxy: (a) peel stress, (b) shear stress.

3.2.2.3 Patch-joint

In the context of the patch-joint analysis, three different scenarios were investigated. This geometry was selected to validate different types of laminated substrates. The first scenario involved a symmetric joint with an asymmetric CFRP $[90/0]$, aluminum and epoxy. The second scenario examined an asymmetric joint, with CFRP $[0/90]$, aluminum and epoxy. Lastly, the third scenario focused on an asymmetric joint with bio-materials. The dimensions and loading conditions for these scenarios are presented in Table 3.4.

Table 3.4: Joint geometry and loading conditions for patch-joint.

	L_T [mm]	L_O [mm]	w [mm]	t_{p1} [mm]	t_{p2} [mm]	t_a [mm]	P [N]
Case 1	100	25	25	2	2	0.5	50000
Case 2	100	25	25	2	4	0.5	50000
Case 3	100	25	25	2	4	0.01	50000

Case 1: symmetric joint with dissimilar materials

In Case 1, a patch-joint with different materials for the substrate, CFRP with a $[90/0]$ layup and aluminum, and an epoxy adhesive were used. The thicknesses of the materials were considered equal, and this case accounted for bending-extension coupling. The obtained plots are depicted in Figure 3.16. In this study, the results show a lack of accuracy in terms of peak peel stress. There are discrepancies between the FEA and analytical model outcomes. However, as in previous cases, the analytical model tends to provide conservative results. Therefore, although the accuracy may not

be optimal in this case, the analytical model can still be applied. For the shear stress distribution, the results are not fully accurate but they are reasonably good.

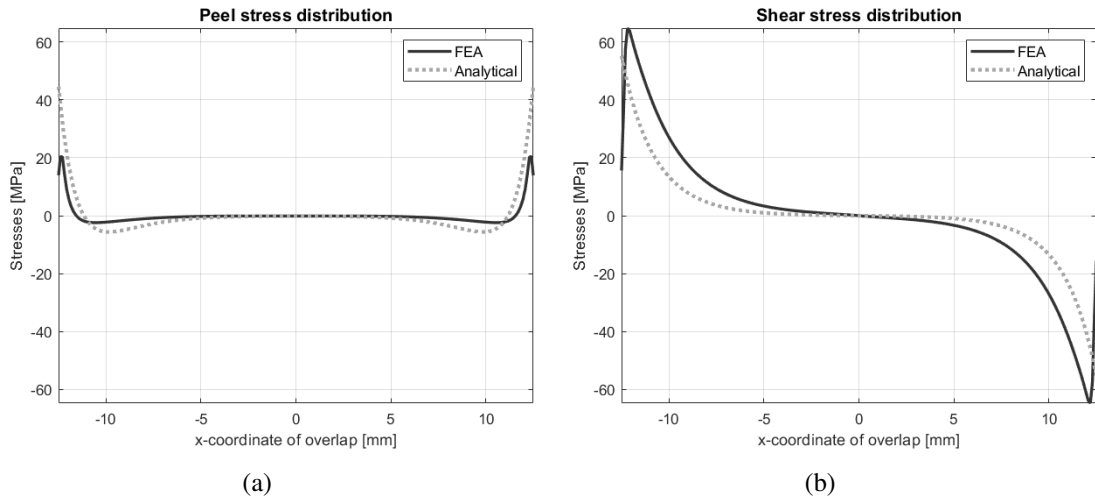


Figure 3.16: Stress distribution in a symmetric patch-joint with CFRP [90/0] and aluminum adhered and epoxy adhesive: (a) peel stress, (b) shear stress.

Case 2: asymmetric joint with dissimilar materials

In Case 2, a similar configuration to the previous case, but with a thicker bottom adherend and an inverted layup for the patch is studied. The obtained plots are displayed in Figure 3.17. This modification, particularly the increase in thickness of the bottom substrate, resulted in an almost perfect agreement for the peel stress distribution. However, there was a drawback observed in the shear stress distribution, which exhibited significant differences with distinct peak shear stresses between the FEA and analytical model, with the analytical model being more conservative.

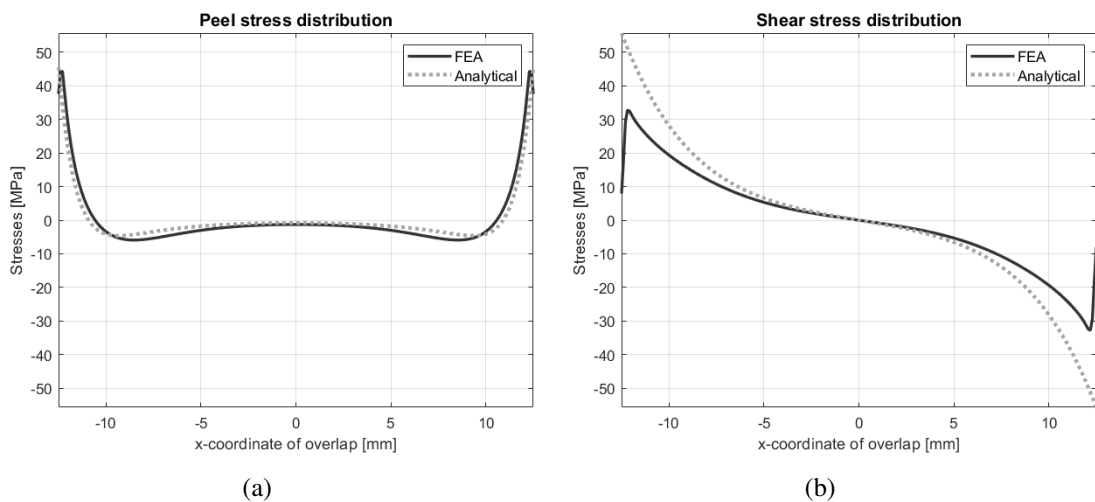


Figure 3.17: Stress distribution in an asymmetric patch-joint with CFRP [0/90] and aluminum adhered and epoxy adhesive: (a) peel stress, (b) shear stress.

Case 3: asymmetric joint with bio-based materials

To gain insights into bio-based joints, a joint configuration consisting of pine wood substrates and a bio-adhesive was simulated. The results of this analysis, shown in Figure 3.18, exhibit a high level of agreement between the analytical and numerical models for both peel and shear stress analysis.

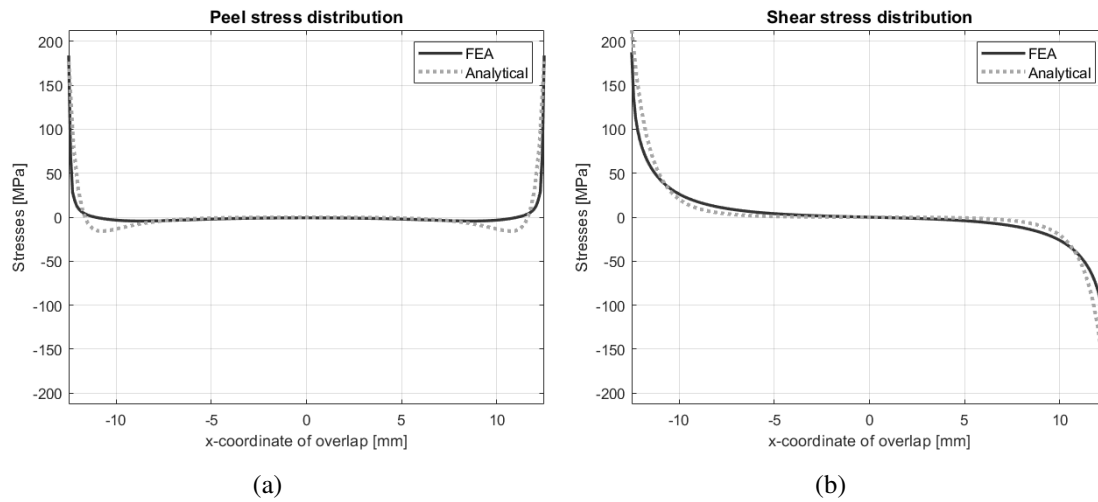


Figure 3.18: Stress distribution in an asymmetric patch-joint with pine wood adhered and bio-adhesive: (a) peel stress, (b) shear stress.

Chapter 4

Web application

This chapter is dedicated to the development of a website that incorporates the general sandwich-type model proposed by Weißgraeber et al. [53], which was described in detail in Section 3.2 and validated for numerous cases in Section 3.2.2. Therefore, this chapter contains an overview of the web application, discussing its technical aspects, design considerations, and key features.

4.1 General considerations

The aim of developing this web tool was to employ an analytical method capable of predicting the strength behavior in the context of adhesive joints. For this purpose, the general sandwich-type model proposed by Weißgraeber et al. [53] was chosen due to its high flexibility in studying various joint configurations and different material types, including arbitrary laminates. The following sections provide a detailed description of the user-defined options available for geometry selection, material definition, and failure criteria. Users have the freedom to define these parameters according to their specific model requirements.

4.1.1 Geometry

In the development of the web tool, a key aspect was to provide designers the flexibility to choose the geometry of the joint they want to analyze. This empowers designers to explore various joint configurations and assess their performance under different scenarios. Unfortunately, it was not feasible to allow users to define the boundary conditions for the joints. The boundary conditions, which determine the constraints applied to the joint, are pre-established based on each specific joint configuration. The joint configuration and boundary conditions are visually displayed in Figure 4.1.

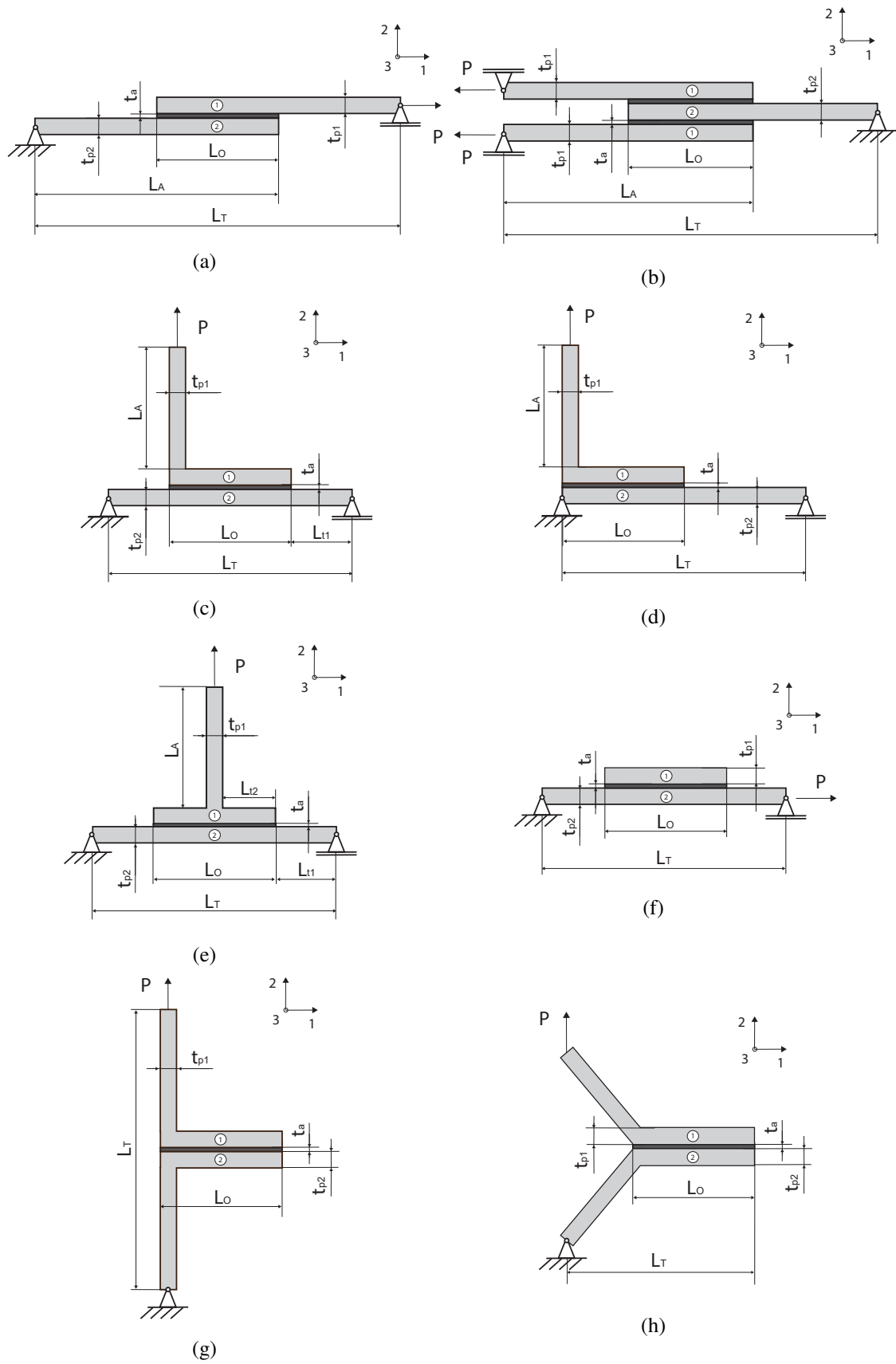


Figure 4.1: Loading and boundary conditions : (a) single lap joint, (b) double lap joint, (c) L-joint, (d) corner-joint, (e) T-joint, (f) patch-joint, (g) peel-joint, (h) inclined-joint.

4.1.2 Material

The implemented analytical model exhibits significant advancements in its capability to model a wide range of material types. These improvements include the incorporation of various material configurations, such as single-layer isotropic and orthotropic materials, as well as arbitrary laminated structures (Figure 4.4(a)). These laminates can be either symmetric or asymmetric, and they can include isotropic or orthotropic layers.

Section 3.2.1 presents the equations that encompass all material types, enabling the analysis and prediction of joint strength with great versatility. To make effective use of the model, the user needs to provide specific properties depending on the chosen material type.

For example, when opting for an isotropic material, the user is required to input Young's modulus, Poisson's ratio, and shear correction (Figure 4.4(b)). On the other hand, if a layup configuration is chosen, the user needs to provide Young's modulus, Poisson's ratio, Shear modulus values for all directions, and specify the number and angle of fiber orientations. It is also crucial for the user to specify the shear correction corresponding to the selected material.

4.1.3 Failure criterion

Furthermore, the web tool offers users the possibility of applying a failure criterion. The model utilizes the maximum failure stress criteria, which is explained in Section 2.2.4. To utilize this criterion, the user is required to define the maximum peel stress and maximum shear stress that the structure can withstand. It is important for the user to consider the direction of the applied load and the material properties of the utilized materials.

When the user enters stress values that are significantly higher than those calculated by the analytical model, no additional information is displayed in the final plots. However, if the user enters values lower than the calculated stresses, a visual indicator in the form of a dotted red line is presented. This dotted red line serves as a boundary that defines the failure region of the structure. It provides a clear visual representation, indicating the stress values beyond which the structure is predicted to fail (Figure 4.5).

4.2 Back-end development

The back-end of the system comprises the core processes that operate behind the scenes, ensuring seamless functionality of the web application. It encompasses various components, with one of the key elements being the implementation of the analytical model.

For the development of the web application, *Wampserver* was chosen as the local development environment. *Wampserver* includes *Apache*, *MySQL*, and *PHP*, making it a suitable solution for deploying web applications on a local machine, primarily for development and testing purposes.

The decision to utilize *PHP* as the server-side language was driven by its versatility and widespread support among web server hosting providers. *PHP* is a well-suited general-purpose language for web development tasks, offering the freedom to create interfaces and applications

tailored to specific requirements. With *PHP*, the possibilities are extensive, and the design of the interface/application is only limited by the *HTML* code that is delivered to the user's browser. Additionally, *PHP* applications can be accessed by users worldwide, regardless of their operating system or device, without encountering any limitations. Moreover, the availability of numerous plugins and packages for *PHP* greatly facilitates the development process. These resources encompass a wide range of functionalities, including plotting packages, navigation packages, and other valuable tools, which significantly contribute to the overall development and enhancement of the web application.

Given that the web application functions as a calculator, capable of handling heavy mathematical calculations, *Python* was also selected as a back-end language. *Python* offers a rich ecosystem of libraries and packages, such as *NumPy* and *SciPy*, which provide extensive mathematical and scientific computing capabilities. By utilizing these libraries, the web application is able to perform complex calculations with ease, speed and accuracy, meeting the requirements for efficient mathematical processing.

4.3 Front-end development

Front-end development involves creating the visual elements that are presented to the user, including the website layout, images, buttons, and other visual components. In this project, the website design was intentionally kept simple to prioritize the core focus, which is the joint analysis.

The user interface begins with the display of all available joint configurations, as shown in Figure 4.2. In this window, users can select the geometry of interest, enabling them to proceed with the analysis.

Once the geometry is selected, the user is directed to another window where they can define the dimensions of the joint geometry, specify the intensity of the applied loading, input material properties, and optionally apply the failure criteria. This window is illustrated in Figure 4.3. For a more comprehensive view of the user's input material options and requests, Figure 4.4 displays a detailed graphic representation.

After the user successfully completes all the input requests, the calculations are performed. As a final result, the web application displays the peel and shear stress distributions. Figure 4.5 provides a visual representation of the final results, serving as an illustrative example of the plots discussed.

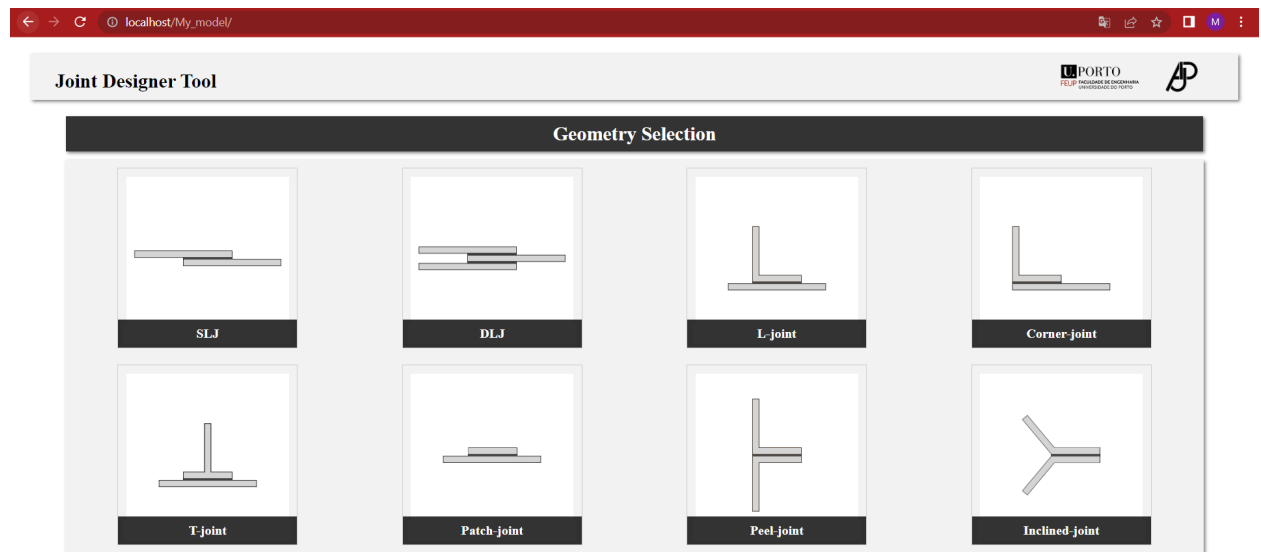


Figure 4.2: Geometry selection interface of the web application.

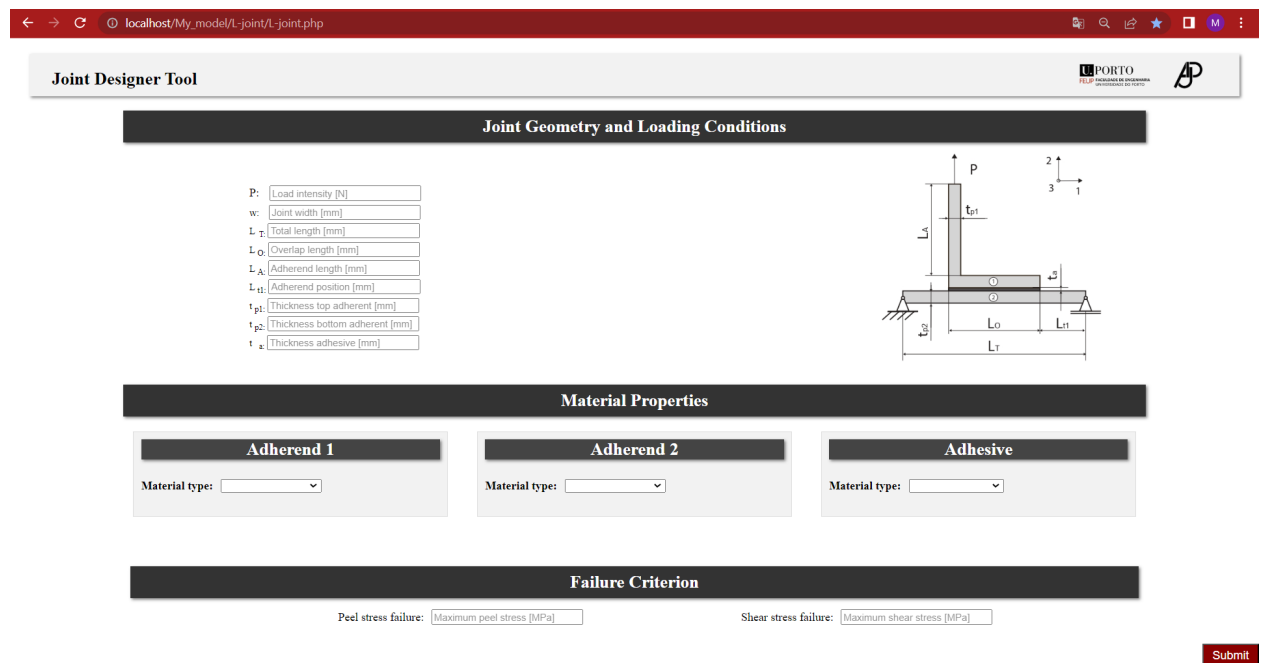
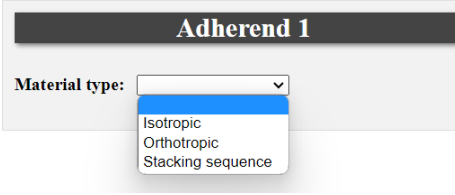


Figure 4.3: Main interface of the web application.



(a)

Adherend 1

Material type: Isotropic

Young's modulus:

Poisson's ratio:

Shear correction:

(b)

Figure 4.4: Material definition: (a) material type, (b) more detailed.

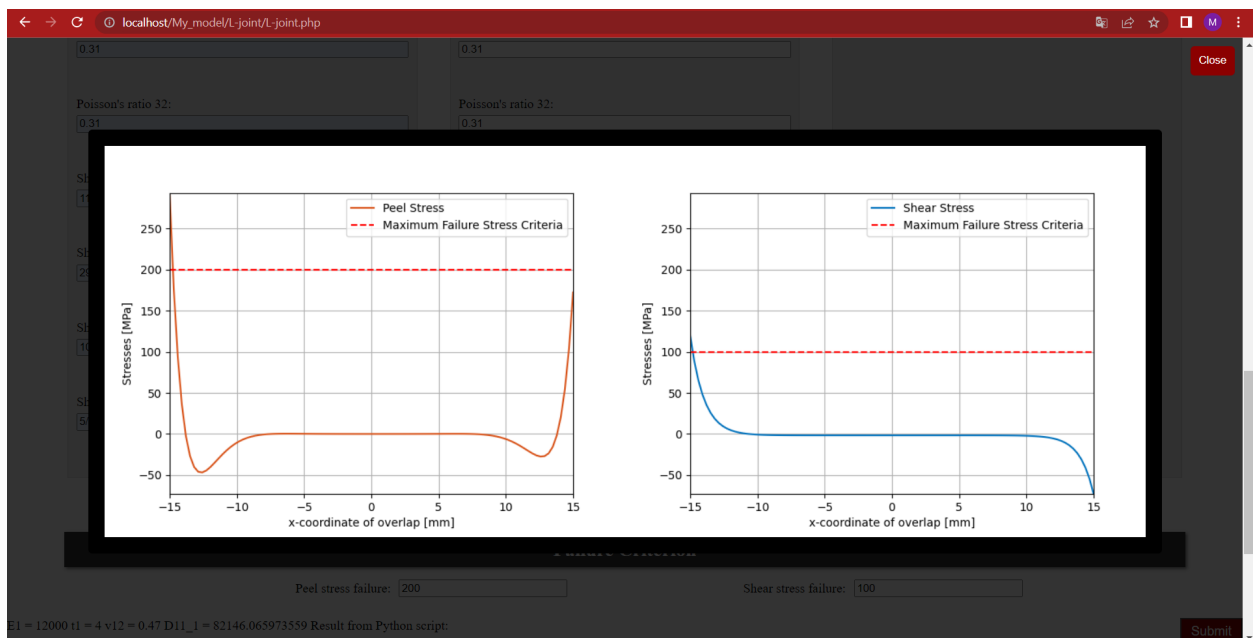


Figure 4.5: Final results: peel and shear stress distributions.

Chapter 5

L-joint testing procedures

This chapter aims to enhance the understanding of the behavior of adhesively bonded wooden L-joints. This is done by investigating the potential of using pine and oak wood for L-joint configurations, which are of great importance in the automotive industry. While some previous studies have explored the use of pine wood, there is limited research specifically focusing on its application in L-joint configurations. Moreover, there is a lack of research regarding the application of oak wood. Therefore, this study specifically examines adhesively bonded L-joints under tension loading conditions. To achieve this objective, an experimental study was conducted, considering two different configurations for each wood type. Additionally, a numerical study was performed to complement and support the findings obtained from the experimental study. The scientific paper written within this study can be found in [Appendix H](#).

5.1 Experimental testing

This section provides a detailed description of the materials used, including their properties, as well as the geometry and dimensions of the joint. The aim is to ensure transparency and reproducibility of the results by disclosing the testing conditions of all experiments. Finally, the section presents the conclusive findings obtained from the experiments.

5.1.1 Materials

The investigation of the L-joint configuration focused on utilizing bio-based materials. Wood materials were specifically chosen as the substrate, and a bio-based adhesive was selected for bonding purposes. Further details about these materials will be provided in the subsequent sections.

5.1.1.1 Wood substrate

In the experimental procedures, two types of wood were utilized: maritime pine timber (*Pinus pinaster* Ait.) and white oak wood (*Quercus alba*). Pine wood is a commonly used softwood in practical structural applications due to its cost-effectiveness, strength, and durability. On the other hand, oak wood is a hardwood with significant potential for real-world applications, primarily due

to its high density compared to other species. This characteristic makes oak wood exceptionally strong, heavy, and durable.

The elastic and strength properties of pine wood (*Pinus pinaster* Ait.) were determined through experimental testing conducted by Moura and Dourado et al. [13]. The elastic properties of oak wood (*Quercus alba*) were obtained from the Forest Products Laboratory (US) [14] and Reiterer et al. [61]. Table 5.1 provides the nominal elastic values, including Young's modulus (E_i), Poisson's ratios (ν_i), and shear modulus (G_i), for both pine and oak wood. In addition, Table 5.2 presents the nominal strength properties of pine and oak wood in different directions, denoted as L (longitudinal), R (radial), and T (transverse). Furthermore, Table 5.3 displays the cohesive parameters specifically for pine and oak wood.

Table 5.1: Nominal elastic properties of pine and oak wood [13, 14, 61].

	E_L [GPa]	E_R [GPa]	E_T [GPa]	ν_{LT} [-]	ν_{LR} [-]	ν_{TR} [-]	G_{LR} [GPa]	G_{LT} [GPa]	G_{TR} [GPa]
Pine	12.0	1.91	1.01	0.51	0.47	0.31	1.12	1.04	0.29
Oak	12.3	2.05	0.89	0.43	0.37	0.3	1.20	1.20	0.80

Table 5.2: Nominal strength properties of pine and oak wood [13, 14, 61].

	σ_{uL} [MPa]	σ_{uR} [MPa]	σ_{uT} [MPa]	σ_{uLR} [MPa]	σ_{uLT} [MPa]	σ_{uRT} [MPa]
Pine	97.5	7.90	4.20	16.0	16.0	4.50
Oak	105	13.8	5.50	-	-	-

Table 5.3: Cohesive parameters of pine and oak wood in the RL fracture system [13, 62, 63, 64, 61].

	G_{Ic} [N/mm]	G_{IIc} [N/mm]	$\sigma_{1,I}$ [MPa]	$\sigma_{1,II}$ [MPa]
Pine	0.264	0.94	5.34	9.27
Oak	0.348	-	-	-

5.1.1.2 Adhesive

In this study, a polyurethane-based bio-adhesive was utilized, which consists of 70% natural components. It is important to note that this bio-adhesive is currently in the prototype stage and has not yet been commercially released. The adhesive was developed by Professor João Bordado from Instituto Superior Técnico and was specifically chosen for its excellent compatibility and adhesion

to wood substrates. Unlike conventional adhesives, this bio-adhesive exhibits unique characteristics as it requires porous wood substrates for effective curing. Once cured, it forms an extremely thin adhesive layer with zero thickness.

The adhesive properties were experimentally characterized by Corte-Real et al. [65]. Regarding the mechanical characterization, testing using a modified thick adherend shear test (TAST) specimen revealed a minimum shear strength value of 12.4 MPa. The author also conducted tests using modified double-cantilever beam (DCB) and end-notched flexure (ENF) specimens, which provided G_{Ic} and G_{IIc} values of 0.16 N/mm. In a supplementary work concerning butt-joints, the tensile strength was determined to be 16.5 MPa.

5.1.2 Joint manufacturing

In the context of joint manufacturing, an important factor to consider is the orientation of wood rings. Depending on the substrate, different orientations were considered. Specifically, for the bottom substrate, the preferred ring orientation is depicted in Figure 5.1. However, it should be noted that achieving perfect symmetry of the wood rings in relation to a centered vertical axis was not always feasible. As for the upper substrate, two different configurations were studied and are presented in Figure 5.2. Consequently, distinct L-joint configurations were manufactured.

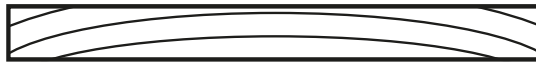


Figure 5.1: Illustration of the cross-section dimensions of the bottom substrate.

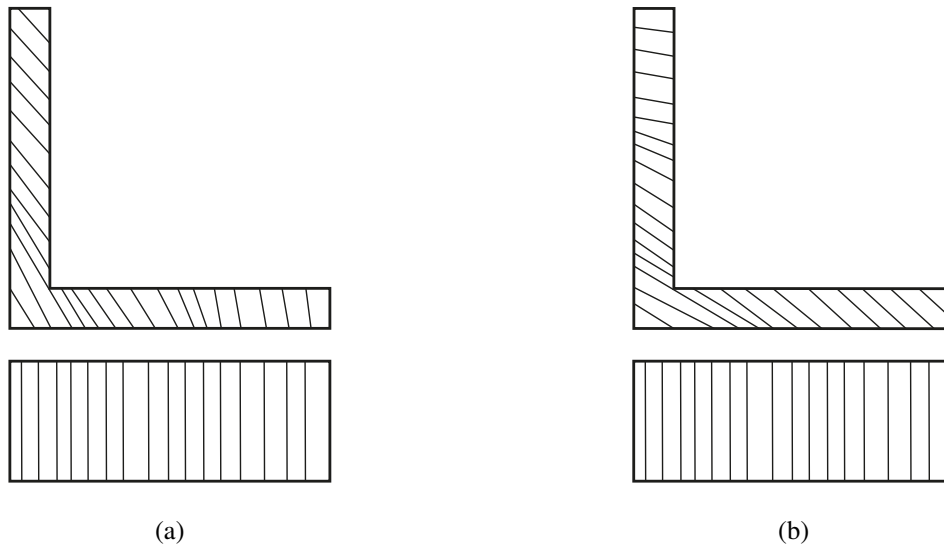


Figure 5.2: Illustration of the cross-section dimensions of the upper substrate: (a) vertical orientation, (b) inclined orientation.

The manufacturing process begins with the preparation of the substrates through a cleaning procedure. Subsequently, a thin layer of adhesive is applied specifically to the overlap region of both substrates. The substrates are carefully aligned and firmly pressed together. To ensure a robust bond, distributed pressure is applied to the overlap region using a plate and clamps. Following that, the assembled substrates are placed in an oven at a temperature of 100°C for 8 hours, allowing the adhesive to undergo the curing process. Lastly, the joints are retrieved, and any excess adhesive on the edges is cleaned, if necessary. The final assembly along with the overall dimensions of the joints can be observed in Figure 5.3.

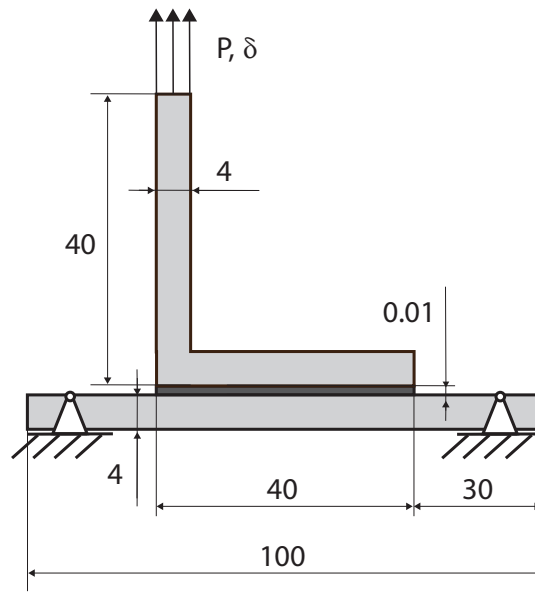


Figure 5.3: Dimensions of the L-joint tested.

5.1.3 Testing conditions

The L-joints manufactured for this study were subjected to tensile testing on the upper substrate using a quasi-static test machine. The traction tests were performed using an *INSTRON 3367*[®] universal testing machine with a load cell capacity of 30 kN. The machine recorded both the load and displacement during the tests at a constant rate of 1 mm/min.

Figure 5.4 illustrates the setup employed to perform the experimental procedures. It is worth noting that a minimum of five specimens were tested for each condition. The tests were conducted under laboratory conditions in terms of temperature and humidity.



Figure 5.4: Experimental setup for testing the L-joints.

5.1.4 Experimental results

A strength analysis was performed for both pine and oak wood, considering the two presented configurations. The observed variations among the studied cases were carefully examined. Additionally, this section includes an evaluation of the failure paths, encompassing an analysis of the behavior of damage initiation and the types of failure observed.

5.1.4.1 Strength analysis

The $P - \delta$ curves presented in this section were derived from experimental procedures, directly extracted from the testing machine. It is important to highlight that, in order to eliminate the influence of machine component measurements, the displacement values for the experimental curves were adjusted using a correction factor of 0.6.

Pine wood

The experimental results for the vertical and inclined configurations considering pine wood are illustrated in Figure 5.5 and Figure 5.6, respectively.

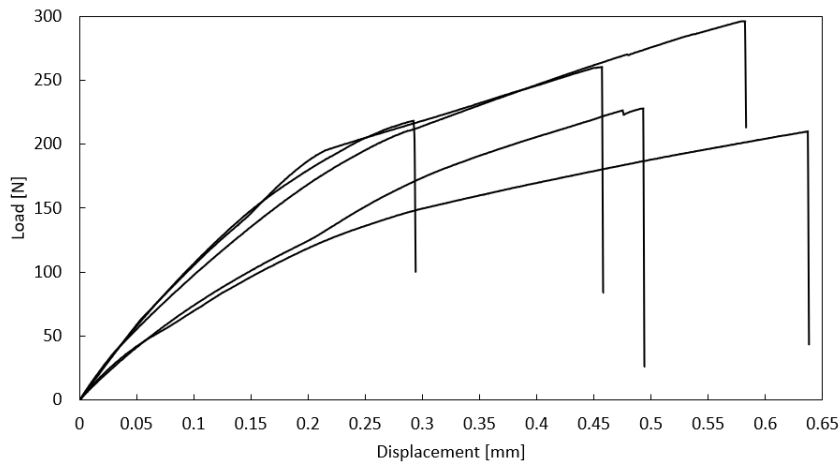


Figure 5.5: P- δ curve for pine wood with vertical configuration.

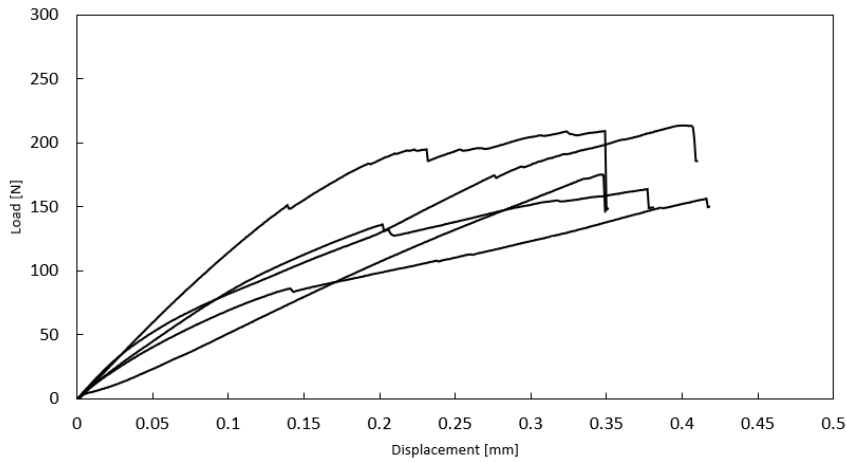


Figure 5.6: P- δ curve for pine wood with inclined configuration.

For the pine wood with a vertical fiber orientation of the upper substrate (Figure 5.5), the joint exhibited failure at a load of 242.9 ± 32.4 N, at a displacement of 0.49 ± 0.12 mm. In the case of the inclined configuration (Figure 5.6), the joint withstood a load of 183.2 ± 23.6 N before failure, resulting in a displacement of 0.38 ± 0.03 mm.

A comparison between these two cases indicates that the vertical configuration demonstrated approximately 25% higher strength compared to the other case. Furthermore, in terms of displacement, it exhibited approximately 22% more capacity for displacement before reaching failure.

Moreover, there is a noticeable disparity in the behavior observed in the two cases. In the vertical configuration, a change in the material's behavior is evident, indicated by a visible shift in the slope of the curve. Initially, the material demonstrates elastic behavior, but then transitions to a plastic behavior, allowing for more displacement before reaching failure. On the other hand, in the inclined configuration, there are small instantaneous decreases in the load along the curve, which

were not observed in the vertical configuration. It suggests the initiation of localized damage. Hence, it can be concluded that the two configurations not only display differences in strength and displacement capability but also exhibit distinct behaviors during the testing process.

It should be noted that superior characteristics are found for the vertical configuration. The inclusion of vertical fiber orientations in the overlap region allows the load to be applied along their strongest direction, thereby enhancing the stiffness and strength of the joint. The vertical alignment of the fibers maximizes their load-bearing capacity, resulting in improved overall performance of the joint.

Oak wood

The experimental results for the vertical and inclined configurations using oak wood are presented in Figure 5.7 and Figure 5.8, respectively.

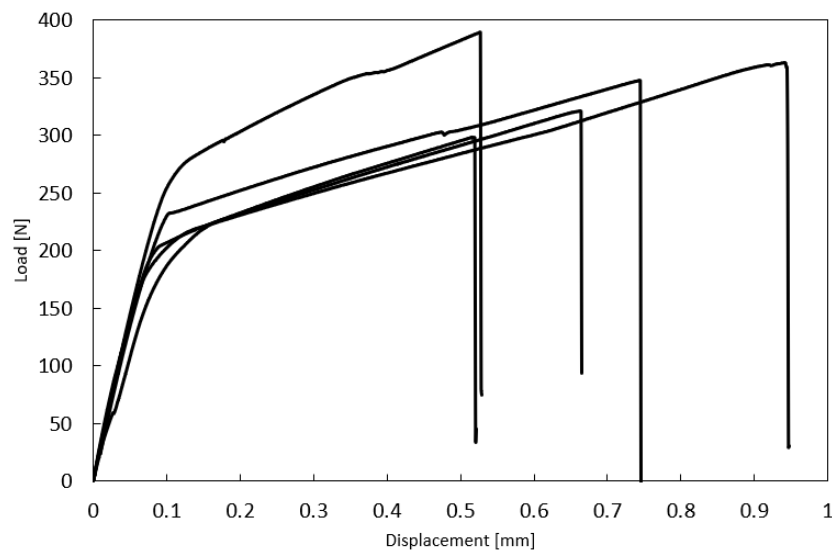


Figure 5.7: P- δ curve for oak wood with vertical orientation.

In the case of oak wood with a vertical fiber orientation of the upper substrate (Figure 5.7), the joint experienced failure at a load of 343.0 ± 33.0 N, with a displacement of 0.68 ± 0.16 mm. For the inclined configuration (Figure 5.8), the joint withstood a load of 214.9 ± 18.4 N before failure, resulting in a displacement of 0.21 ± 0.05 mm.

A comparison between these two cases reveals that the vertical configuration exhibited approximately 37% higher strength compared to the other case. Moreover, in terms of displacement, it demonstrated approximately 69% more capacity for displacement before reaching failure.

The curves obtained for oak wood exhibit a resemblance to the previously observed behavior in pine wood. When testing the vertical configuration, the wood shows elastoplastic behavior. For the inclined configuration, there are indications of localized damage initiation, as seen previously for pine wood.

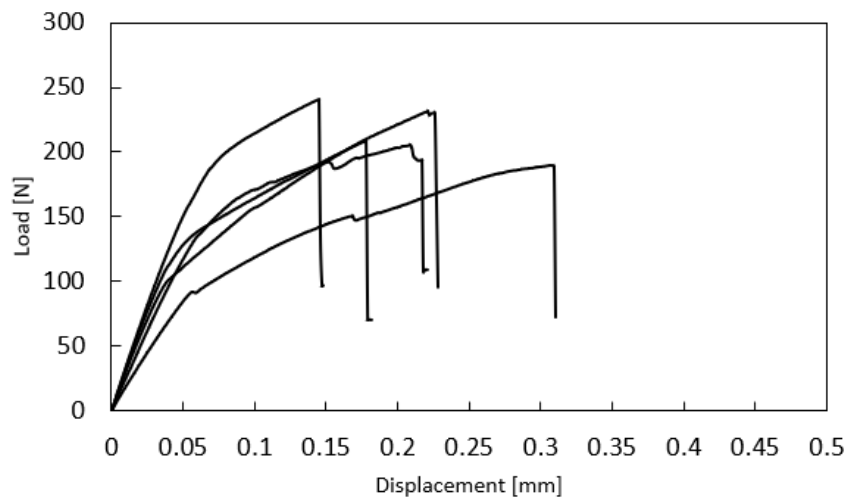


Figure 5.8: P- δ curve for oak wood with inclined orientation.

5.1.4.2 Failure mode

Following the testing phase, a thorough analysis was conducted to identify the types of failure modes observed in all four cases. In all cases, the presence of a visibly non-reflective surface in the overlap region strongly indicates the occurrence of delamination. However, two distinct behaviors were observed following the initiation of delamination in the upper substrate. In the vertical configuration for both woods the delamination process involved a larger number of fibers being pulled, accompanied by a fracture within the wood at a highly oblique angle (Figure 5.9(a),(b) and Figure 5.10(a),(b)). In the case of the inclined configuration, the failure plane exhibited a more vertical orientation and occurred suddenly, resulting in a flatter failure surface (Figure 5.9(c),(d) and Figure 5.10(c),(d)).

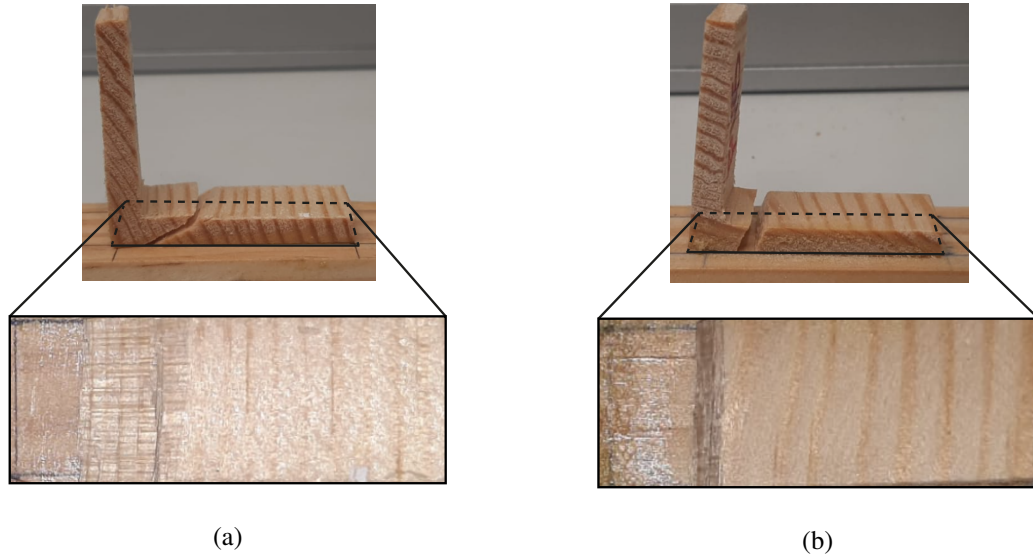


Figure 5.9: Failure mode observation for pine wood: (a) in the vertical configuration, (b) in the inclined configuration.

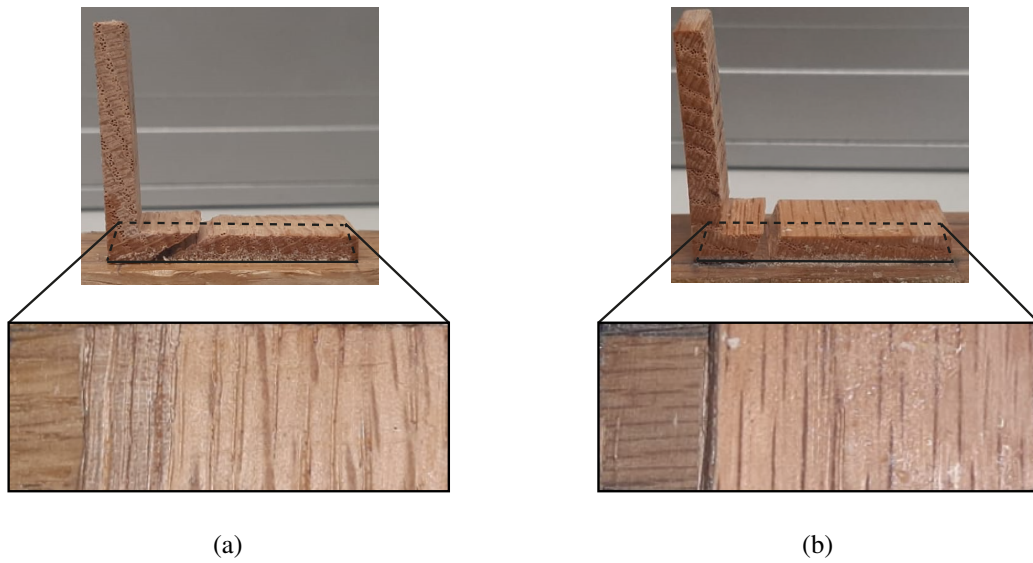


Figure 5.10: Failure mode observation for oak wood: (a) in the vertical configuration, (b) in the inclined configuration.

5.2 Numerical analysis

This section provides a comprehensive overview of the numerical details as well as the corresponding results obtained for all cases.

5.2.1 Numerical details

This section begins with a brief description of the general considerations taken into account, followed by the description of the boundary conditions and mesh employed.

5.2.1.1 General considerations

To predict the behavior of the L-joints, a finite element model was developed using the *Abaqus* 2022[®] software. The model aimed to simulate the real performance of the joints.

Four static numerical models were created to represent all configurations mentioned earlier (Figure 5.2), using both pine and oak wood with the bio-adhesive. To simulate the adhesive layer, cohesive elements were employed, and their behavior was defined using a triangular cohesive law. The properties of this law, such as strength and fracture characteristics, were determined through experimental research conducted by Corte-Real et al. [65], specifically for the novel bio-adhesive. For the wood substrates, a homogeneous anisotropic layer was used, and the corresponding elastic properties were obtained from Table 5.1 [13, 14, 61]. Additionally, a layer of cohesive elements was positioned at a distance of 0.15 mm from the interface of the upper and bottom substrates to account for potential delamination of the wood. The triangular cohesive law governing these elements was defined based on the strength and fracture properties of natural pine and oak wood, as determined through experimental analysis by Moura and Dourado [13], from the Forest Products Laboratory (US) [14] and Reiterer et al. [61]. A visual representation of the model is depicted in Figure 5.11.

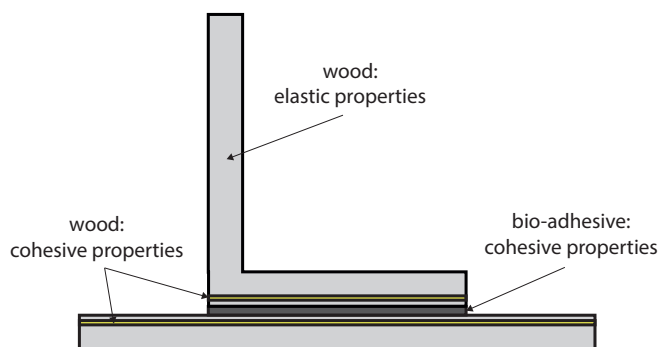


Figure 5.11: Cohesive elements placed in L-joint numerically tested.

5.2.1.2 Boundary conditions

In order to replicate the experimental testing conditions accurately, careful attention was given to the selection of appropriate boundary conditions. For the numerical simulation, the horizontal and vertical directions of the two upper edges of the bottom substrate were fixed to prevent any movement. The upper edge of the upper substrate was subjected to a vertical upward displacement,

while being constrained from any horizontal movement. A visual representation of this boundary condition configuration can be seen in Figure 5.12.

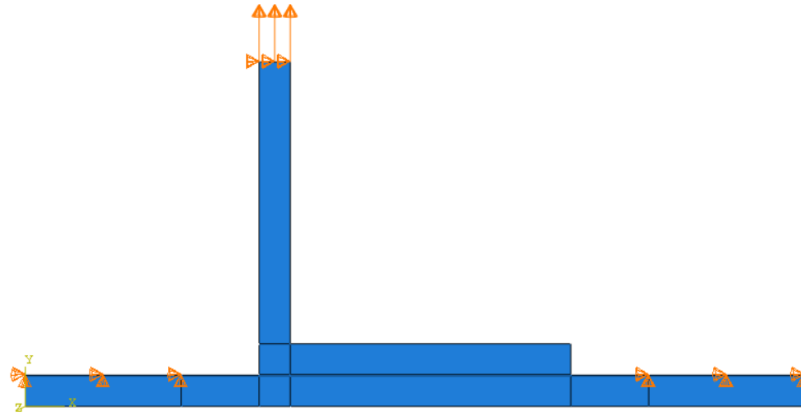


Figure 5.12: Boundary conditions applied to the L-joint.

5.2.1.3 Mesh

To ensure accurate simulations, a high mesh density was employed for all cases. The geometry was discretized using a total of 11740 four-node bilinear plane strain quadrilateral elements (CPE4) to capture the overall behavior. Additionally, 700 four-node two-dimensional cohesive elements (COH2D4) were incorporated to effectively model the cohesive layers. To accurately capture bending phenomena, the thickness of the substrates was discretized using more than 20 elements. Furthermore, a mesh refinement strategy was implemented specifically in the right-side region of the adhesive to ensure a more precise analysis of localized stress effects in that particular area. Figure 5.13 presents the final mesh configuration obtained.

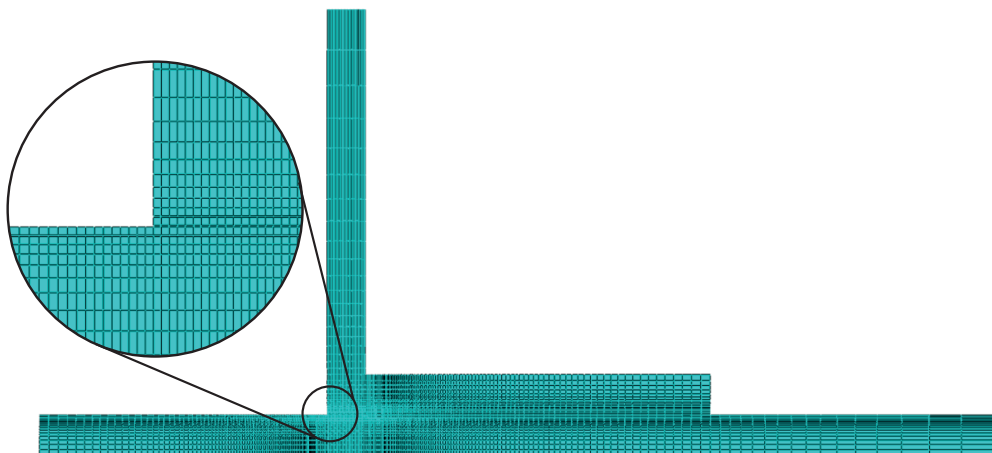


Figure 5.13: Mesh applied to the L-joint.

5.2.2 Numerical results

The models were subjected to a strength analysis, and the obtained results were compared with the experimental data for validation. Moreover, a thorough investigation was conducted to examine the failure paths, including analyzing the behavior of damage initiation and identifying the specific type of failure observed. This analysis aimed to provide insights into the structural response and failure mechanisms of the L-joints under the given loading conditions.

5.2.2.1 Strength analysis

The strength analysis for all configurations was conducted using the FE software. $P - \delta$ curves were obtained from the analysis and compared to the corresponding experimental data for both pine and oak wood.

Pine wood

The comparison between the numerical and experimental results specifically for pine wood is presented in Figure 5.14 for the vertical configuration and Figure 5.15 for the inclined configuration.

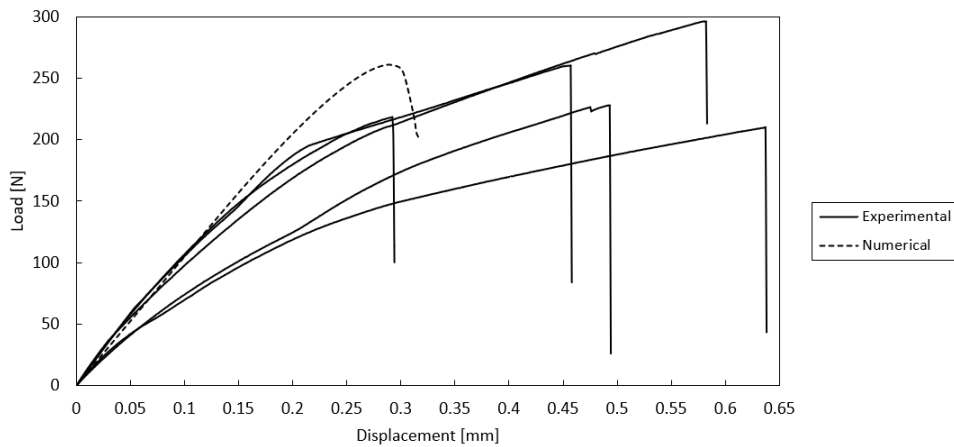


Figure 5.14: Comparison between experimental and numerical $P - \delta$ curves for the vertical configuration of the pine wood L-joint.

In the vertical configuration, the numerical analysis predicts a failure load of 261.2 N and a displacement of 0.28 mm. In comparison, the experimental failure load was observed to be 242.9 ± 32.4 N with a corresponding displacement of 0.49 ± 0.12 mm. These findings indicate that the numerical model accurately predicts the failure load of the joint. However, there is a slight discrepancy in the displacement values, where the numerical results fall slightly short of the experimental data. Up to approximately 0.2 mm of displacement, the slope between the numerical and experimental results aligns well. However, beyond that point, a noticeable deviation in behavior is observed between the numerical and experimental results. This deviation may be attributed to the presence of plastic behavior in the experimental testing, which allows for additional displacement

before failure. Unfortunately, this plastic behavior was not accounted for in the numerical model due to a lack of material data to accurately simulate it. Despite this limitation, the numerical results still demonstrate reasonable accuracy in predicting the overall behavior of the joint.

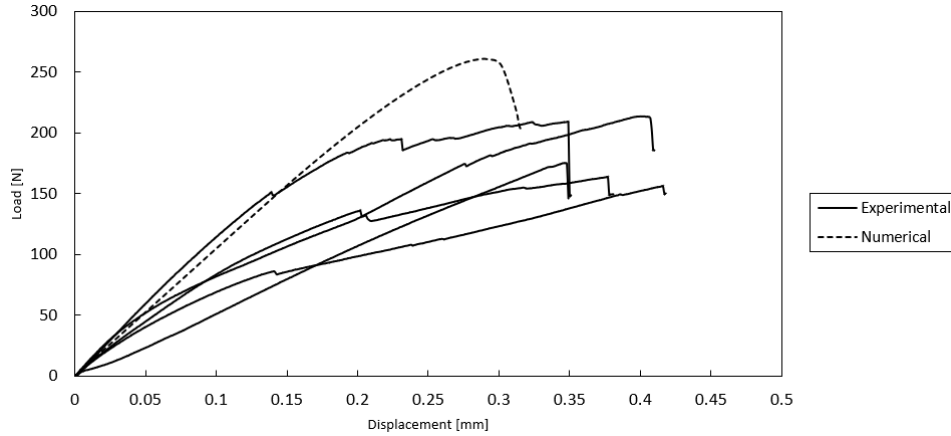


Figure 5.15: Comparison between experimental and numerical $P - \delta$ curves for the inclined configuration of the pine wood L-joint.

In the inclined configuration, the numerical analysis predicts a failure load of 260.8 N and a displacement of 0.29 mm. On the other hand, the experimental testing resulted in a load until failure of 183.2 ± 23.6 N, with a corresponding displacement of 0.38 ± 0.03 mm. This discrepancy can be attributed to the presence of microcracks in the experimental testing that are not accounted for in the finite element analysis (FEA). As a result, the FEA tends to predict higher failure loads compared to the experimental results.

The overall findings of the numerical analysis provide good predictions of the joint's strength. When comparing the $P - \delta$ curves, it is evident that the numerical results closely match the experimental data, indicating a reasonable agreement between the two approaches. Specially for the inclined configuration, once the plastic behavior of the material in the experimental study is not evident, the numerical results are very much closer. Nevertheless, for both cases the FEA accurately captures the general behavior of the joint, even though there may be some variation in the specific failure loads and displacements. Overall, the numerical analysis provides valuable insights into the strength characteristics of the joints.

When comparing the numerical analysis results for both joint configurations, it is evident that they exhibit similarities in terms of failure load, displacement, and the shape of the $P - \delta$ curve. However, in the experimental results, the two configurations exhibit distinct behaviors as described earlier in Section 5.1.4. Thus, it can be concluded that the experimental results play a crucial role in gaining a deep understanding of the behavior of the structure. The numerical analysis provides valuable insights and predictions, but it is the experimental data that provides the real-world validation and helps uncover the nuances and complexities of the system. Only combining the numerical and experimental approaches, a more comprehensive and accurate understanding of the structure's behavior can be achieved.

Oak wood

Figure 5.16 illustrates the comparison between the experimental and numerical results for the vertical configuration, while Figure 5.17 showcases the same comparison for the inclined configuration considering oak wood.

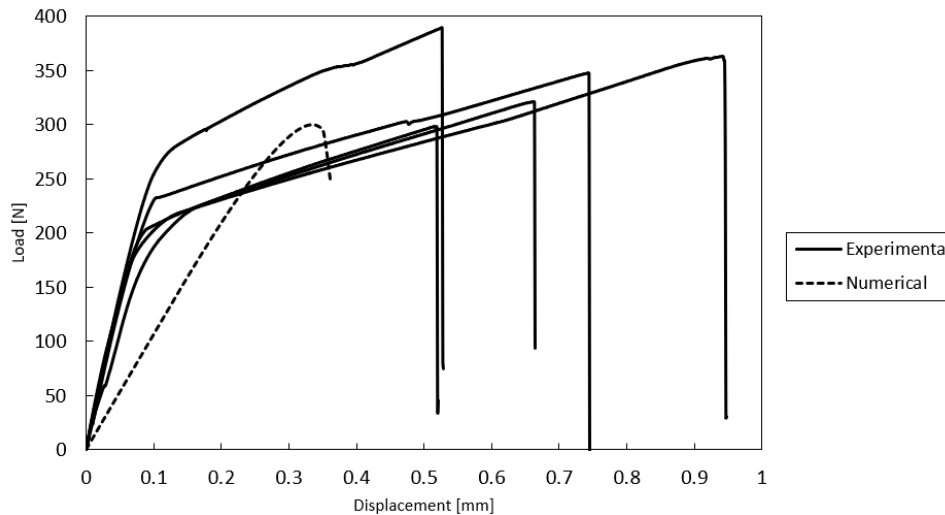


Figure 5.16: Comparison between experimental and numerical $P - \delta$ curves for the vertical configuration of the oak wood L-joint.

In the vertical configuration, the numerical analysis predicts a failure load of 299.9 N and a displacement of 0.33 mm. However, when compared to the experimental results, the observed failure load was 343.0 ± 33.0 N, accompanied by a displacement of 0.68 ± 0.16 mm. Therefore, although the numerical simulation managed to estimate a reasonably similar failure load, it falls short in accurately predicting both the displacement at failure and the material's behavior.

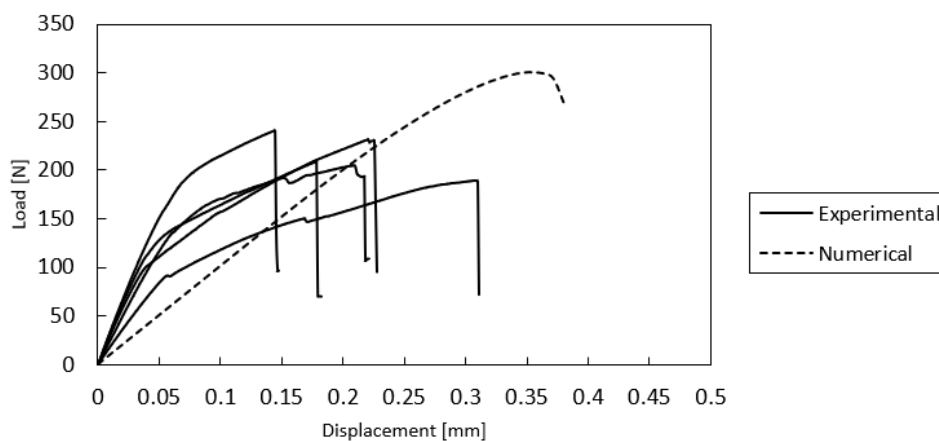


Figure 5.17: Comparison between experimental and numerical $P - \delta$ curves for the inclined configuration of the oak wood L-joint.

In the inclined configuration, the numerical analysis predicts a failure load of 300.2 N and a displacement of 0.35 mm. However, experimental testing revealed that the load until failure was observed to be 214.9 ± 18.4 N, with a corresponding displacement of 0.21 ± 0.05 mm. It is evident that the numerical simulation failed to accurately predict the failure load, displacement, and overall behavior of the joints.

Upon careful analysis of both cases, it becomes apparent that the FEA produced predictions that significantly deviated from the experimental results in terms of joint strength, displacement, and overall behavior, with the exception of the failure strength in the vertical configuration. This substantial disparity can be attributed to the inherent uncertainty surrounding the properties and conditions of the wood being studied, since the exact wood used in this study was not yet fully characterized and may be different from the one reported in literature. Consequently, the lack of certainty regarding the wood's properties presents a formidable challenge that exerted a substantial influence on the obtained results.

5.2.3 Failure mode

The numerical analysis revealed that delamination was the primary observed failure mode in the experiments. To gain a deeper understanding of this phenomenon, the state variable SDEG was utilized to track the evolution of damage. The SDEG value serves as an indicator of damage, with a value of zero indicating the absence of any damage. As damage progresses, the SDEG value increases linearly, and when it reaches one, the cohesive element fails and disappears. Figure 5.18 visually depicts the occurrence of delamination in the upper cohesive layer of the wood, specifically showcasing horizontal propagation.

Further analysis revealed the significance of considering the maximum principal stress criteria. According to the maximum principal stress failure predictor (MPSFP), fracture occurs when the maximum principal stress surpasses the local strength in a component experiencing a multiaxial stress system. Upon investigation, it was found that, at a given displacement, the maximum principal stress exceeded the local strength for the LR plane, which corresponds to 16 MPa for pine wood and around 20 MPa for oak wood. This indicates that the crack propagated in that specific region, perpendicular to the direction of the maximum principal stress. Figure 5.19 and Figure 5.20 show the direction of the maximum principal stress, retrieved directly from *Abaqus 2022*® software. Consequently, the observed failure path in the numerical analysis accurately corresponds to the failure observed in the experiments.

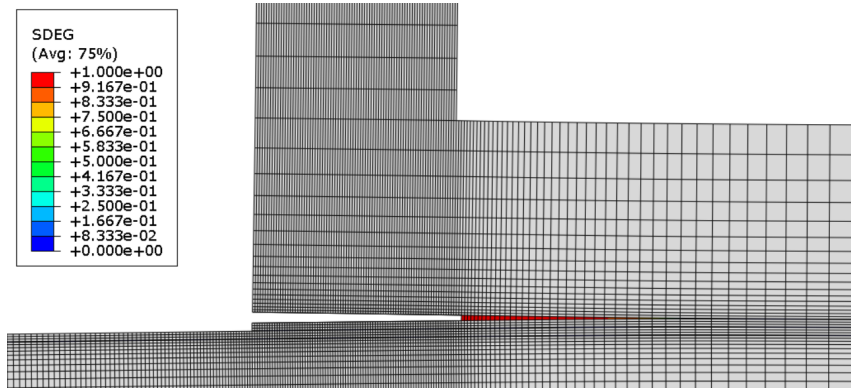


Figure 5.18: Damage analysis (SDEG) for L-joint.

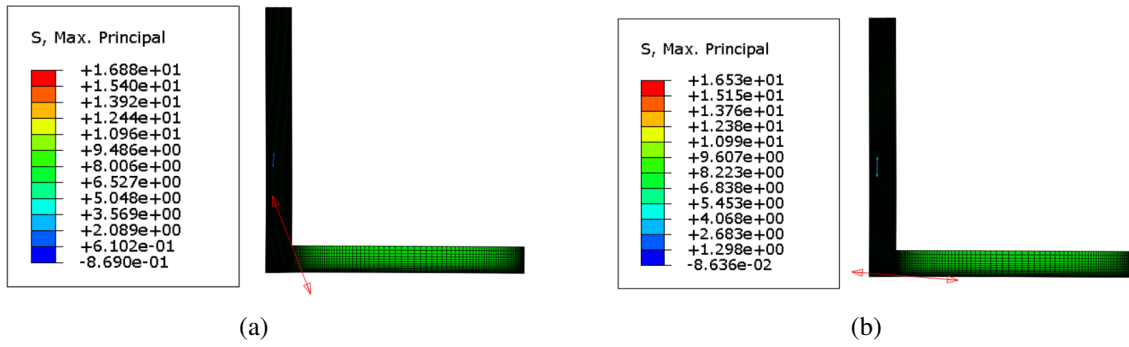


Figure 5.19: Maximum principal stress distribution in the FEA for pine wood with: (a) vertical configuration, (b) inclined configuration.

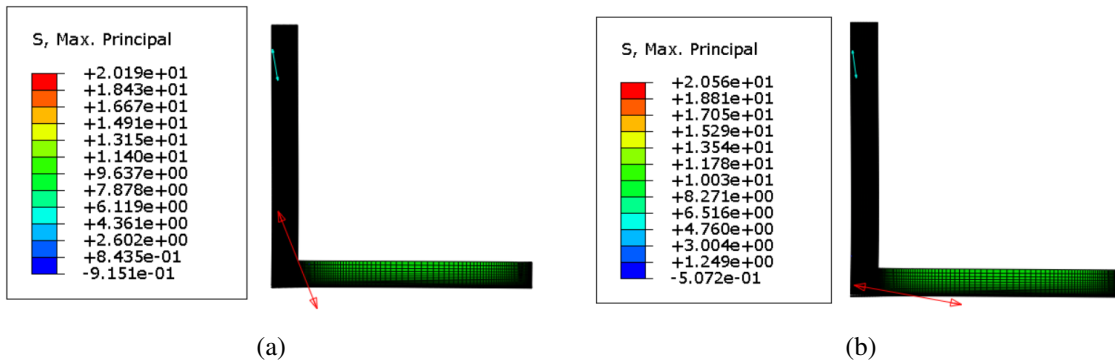


Figure 5.20: Maximum principal stress distribution in the FEA for oak wood with: (a) vertical configuration, (b) inclined configuration.

5.3 Discussion

This section aims to discuss the experimental and numerical findings for all case studies, focusing on the strength analysis and failure modes observed.

A comparison between the two wood types highlighted the superior behavior of oak wood, which is consistent with its classification as a hardwood. Specifically, for the vertical configuration, oak wood exhibited approximately 29% higher strength than pine and 28% greater displacement. For the inclined configuration, oak wood demonstrated approximately 15% higher strength but 81% lower displacement compared to pine. Therefore, these findings emphasize the significance of considering the chosen configuration when assessing wood performance.

There is noteworthy conformity in the behavior of both wood types when comparing the same joint configuration. The $P - \delta$ curves exhibited similar shapes, indicating consistent joint behavior. In the vertical configuration, an elastoplastic response was observed, while microcracks became apparent during the inclined configuration testing. However, it is important to note that the numerical results displayed similar curves when comparing different configurations for the same wood. This indicates that the numerical analysis, while providing valuable insights, was not entirely precise in predicting the exact joint behavior.

Among all the cases analyzed, it is evident that oak wood demonstrates the highest tensile strength, particularly in the vertical fiber orientation for the upper substrate. Pine wood with the same configuration exhibits the second-highest strength. However, the lowest strength is observed in the case of pine wood with the inclined fiber orientation. Considering displacement as a crucial factor, it is noteworthy that oak wood with vertical fiber orientation exhibits the most desirable behavior. Conversely, the same wood with the alternative configuration demonstrates less favorable displacement characteristics.

It is important to emphasize that there was a lack of agreement between the experimental and FEA analysis for oak wood. This disparity can be attributed to the limited characterization of oak wood, which introduces uncertainties regarding the specific wood properties and its conditions. Additionally, the properties of oak wood reported in the literature are not well-established, resulting in either a wide range of values or missing properties. Therefore, in future work, a careful characterization of the material should be conducted. Only then can the material properties used in the numerical simulation be adjusted. Consequently, the numerical simulation presented for oak wood should be disregarded until proper characterization is performed.

Regarding the failure mode, delamination consistently manifested in both the experimental and numerical results. Upon comparing all cases, distinct failure paths were observed, demonstrating a clear and consistent correlation with the joint configuration. As a result, both wood types exhibited similar behavior for the same configuration, showcasing satisfactory agreement between the experimental and numerical findings.

In conclusion, the results provide valuable insights into the behavior of pine and oak wood, particularly in terms of experimental testing. Similar behavior was observed when testing the same configurations for these wood types. It was evident that oak wood may offer better structural performance depending on the configuration employed. Hence, the configuration of the joint is a crucial aspect to consider as wood is an anisotropic material. In terms of pine wood, there is sufficient confidence to simulate it numerically, as it provides accurate strength predictions that align well with real-world applications, thanks to its extensive utilization and research. However,

for oak wood, further studies are required to reach the same level of confidence as pine wood in design applications.

Chapter 6

Conclusions

6.1 Conclusions

This chapter presents the conclusions drawn from the research conducted within the scope of this master's thesis. The study focused on analyzing and modeling eco-friendly adhesively bonded structures using analytical methods. Two specific analytical methods were selected and their suitability to predict the stress distributions in a wide range of adhesive joints configuration was verified through numerical analysis.

In addition to the analytical methods, a website was developed to implement the general sandwich-type model. This website allows users to design not only bio-based structures but also a wide range of materials, including isotropic, orthotropic, and arbitrary laminates. Another significant innovation of the website is its capability to easily study various joint configurations with a relatively high level of accuracy.

To further investigate the topic, adhesively bonded L-joints, commonly used in the automotive industry, were tested and numerically simulated. The study aimed to enhance the understanding of bio-materials, specifically wood, by selecting pine and oak wood along with a novel polyurethane-based bio-adhesive. The mechanical performance of these joints was compared, and it was observed that oak wood exhibited higher loading capacity and elongation capability compared to pine wood. Moreover, the vertical fiber orientation in the overlap region was found to provide a stronger configuration than other orientations. The numerical analysis demonstrated excellent agreement with the experimental results for pine wood, but this level of agreement was not observed for oak wood. Nevertheless, a high consistency was observed in the failure mode between the experimental and numerical outcomes.

Overall, this research provides valuable insights into the analytical modeling of eco-friendly adhesively bonded structures. The developed website and the analysis of L-joints contribute to advancing the knowledge and understanding of bio-materials and their mechanical behavior. The findings can be used to guide the design and optimization of adhesively bonded structures in various applications, particularly in the automotive industry.

6.2 Future work

Based on the progress made in the project, the next phase should prioritize the incorporation of nonlinear effects into the website that was developed. Specifically, these nonlinear effects should account for the plastic behavior of the adhesive. By considering these effects, the preliminary design of bonded joints can closely approximate real-life scenarios. This enhancement will provide users with more accurate and reliable results, allowing for better-informed design decisions.

Another crucial aspect to consider is the characterization of wood, which enables the determination of more accurate material properties. By conducting the characterization process, it becomes feasible to achieve a close correlation between experimental and numerical analyses. In the case of pine wood, it is recommended to focus on characterizing its plastic behavior. For oak wood, a comprehensive characterization process is suggested to capture its complete range of material properties.

Furthermore, it is highly recommended to expand the testing efforts by conducting further experiments on a larger number of L-joints. These joints should be subjected to various testing parameters, including:

- Diverse geometry sizes in order to investigate the impact of joint dimensions on its mechanical performance.
- Different loading modes to expand the knowledge of the behavior of the joint.
- Various loading rates to further evaluate the performance of the joints.
- Different fiber alignment to analyze its influence on the strength and durability of the joint.

By incorporating these variations into the testing procedure, more reliable and valid results can be obtained, leading to a well-defined envelope that characterizes the joint's performance across different conditions. This comprehensive approach will provide valuable insights for future design and optimization considerations, enabling engineers to make informed decisions regarding L-joint configurations and materials.

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Appendices

Appendix A

Software packages available on the market

Table A.1: Software packages available on the market [54].

Name	Supplier	Application	Features
Bolt	G.S. Springer Stanford University, USA	Design of pin-loaded holes in composites	• Prediction of failure strength and failure mode • Three types of bolted joints: joints with a single hole, joints with two identical holes in a row • Joints with two identical holes in tandem • Applicable to uniform tensile loads and symmetric laminates
BISEPSLOCO	AEA Technology, UK	Closed form computer code for predicting stresses and strains in adhesively-bonded single-lap joints	• Tensile/shear/bending moment loading • Adhesive peel and shear stress predictions • Allowance for plasticity in adhesive layer • Thermal stress analysis
BISEPSTUG	AEA Technology, UK	Closed form computer code for predicting stresses and strains in adhesively-bonded coaxial joints	• Stepped and profiled joints • Orthotropic adherends • Torsional and axial loading • Allowance for plasticity in adhesive layer • Thermal stress analysis
CoDA	National Physical Laboratory, UK	Preliminary design of composite beams and panels, and bolted joints	• Synthesis of composite material properties (lamina and laminates for a range of fibre formats) • Parametric analyses • Panel and beam design • Bonded and bolted double shear joints • Bearing, shear-out, pin shear and by-pass tensile failure prediction
DLR	DLR-Mitteilung, Germany	Preliminary design of composite joints	• Adhesively-bonded and bolted joints • Linear-elastic and linear-elastic/plastic behaviour

Name	Supplier	Application	Features
Feloco	AEA Technology, UK	Finite element module computer code for predicting stresses and strains in adhesively-bonded lap shear joints	• Tension and shear loading • Symmetric and asymmetric lap joints • Bearing, shear-out, pin shear and by-pass tensile failure prediction (washers and bolt tightening) • Stepped and profiled joints • Tensile/shear/bending moment/pressure loading • Linear and non-linear analysis • Peel, shear and longitudinal stress predictions in adhesive layer and adherends • Thermal stress analysis for adherend and adhesive
PAL	Permabond, UK	'Expert' system for adhesive selection	• Joined systems include: • Lap and butt joints, Sandwich structures, Bushes/gears/bearings/shafts/pipes/threaded fittings • Elastic analysis • Creep/fatigue effects on joint stiffness (graphical)
RETCALC	Loctite, UK	Interactive Windows-based general-purpose software	• Joint strength • Correction factors (temperature and fatigue)
ESDU	Engineering Science Data Unit, UK	Software for use in structural design	• ESDU 78042 shear stresses in the adhesives in bonded joints. Single-step double-lap joints loaded in tension • ESDU 79016 Inelastic shear stresses and strains in the adhesives bonding lap joints loaded in tension or shear (computer program) • ESDU 80011 elastic stresses in the adhesive in single step double-lap bonded joints

Name	Supplier	Application	Features
Joint designer (da Silva, Lima, & Teixeira, 2009)	Faculty of Engineering, University of Porto, Portugal	Closed-form computer code for predicting stresses and strength in adhesively-bonded joints	• ESDU 80039 elastic adhesive stresses in multi-step lap joints loaded in tension (computer program) • ESDU 81022 Guide to the use of Data Items in the design of bonded joints • Accessible to non-experts • Lap joints • Adhesive peel and shear stress predictions • Joint strength prediction • Allowance for plasticity in adhesive layer and adherends • Orthotropic adherends • Thermal stress analysis
JointCalc (Dragoni et al., 2010)	Henkel AG, Germany	Closed-form computer code for predicting stresses and strength in adhesively-bonded joints	• Accessible to non-experts • Single- and double-lap joints, single- and double-strap joints, peel joints and cylindrical joints • Adhesive peel and shear stress predictions • Joint strength prediction

Appendix B

Bigwood and Crocombe's model

B.1 Simplified peel analysis

To solve the peel stress distribution equation, the auxiliary equation method and the general form of the solution are employed. Two boundary conditions are required to establish the solution. Equations B.1 and B.2 are the derived results obtained by equating the applied load at both ends of the adherends with the derivatives of the peel stress equation.

$$\frac{d^2\sigma_y}{dx^2} = \frac{E_a}{t} \left(\frac{M_2}{D_2} - \frac{M_1}{D_1} \right) \quad (\text{B.1})$$

$$\frac{d^3\sigma_y}{dx^3} = \frac{E_a}{t} \left(\frac{V_2}{D_2} - \frac{V_1}{D_1} \right) \quad (\text{B.2})$$

Studying both ends of the overlap, four simultaneous equations defining the constants $A_1 - A_4$ of peel stress distribution equation. Their solutions are found following:

$$A_1 = \frac{[b_3 R_3 - 2b_2 \sinh(K_5 L) \sin(K_5 L) + b_1 R_6 + b_4 R_1]}{R_5} \quad (\text{B.3})$$

$$A_2 = \frac{b_2 R_2 - b_3 \sinh^2(K_5 L) - b_1 R_4 - b_4 \sinh(K_5 L) \sin(K_5 L)}{R_5} \quad (\text{B.4})$$

$$A_3 = \frac{b_2 R_2 - b_3 \sin^2(K_5 L) - b_1 R_4 - b_4 \sinh(K_5 L) \sin(K_5 L)}{R_5} \quad (\text{B.5})$$

$$A_4 = b_1 \quad (\text{B.6})$$

where $b_1 - b_4$ and $R_1 - R_6$ are constant and found following:

$$b_1 = \frac{E_a}{2K_5^2 t} \left(\frac{M_{21}}{D_2} - \frac{M_{11}}{D_1} \right) \quad (\text{B.7})$$

$$b_2 = \frac{E_a}{2K_5^2 t} \left(\frac{M_{22}}{D_2} - \frac{M_{12}}{D_1} \right) \quad (\text{B.8})$$

$$b_3 = \frac{E_a}{2K_5^3 t} \left(\frac{V_{21}}{D_2} - \frac{V_{11}}{D_1} \right) \quad (\text{B.9})$$

$$b_4 = \frac{E_a}{2K_5^3 t} \left(\frac{V_{22}}{D_2} - \frac{V_{12}}{D_1} \right) \quad (\text{B.10})$$

$$R_1 = \cosh(K_5 L) \sin(K_5 L) - \sinh(K_5 L) \cos(K_5 L) \quad (\text{B.11})$$

$$R_2 = \cosh(K_5 L) \sin(K_5 L) + \sinh(K_5 L) \cos(K_5 L) \quad (\text{B.12})$$

$$R_3 = \cosh(K_5 L) \sinh(K_5 L) - \cos(K_5 L) \sin(K_5 L) \quad (\text{B.13})$$

$$R_4 = \cosh(K_5 L) \sinh(K_5 L) + \cos(K_5 L) \sin(K_5 L) \quad (\text{B.14})$$

$$R_5 = \sinh(K_5 L) - \sin 2(K_5 L) \quad (\text{B.15})$$

$$R_6 = \cosh(K_5 L) - \cos(K_5 L) \quad (\text{B.16})$$

B.2 Simplified shear analysis

Similar to the previous case, two boundary conditions must be considered. These boundary conditions establish a relationship between the applied load at both ends of the adherends and the evaluation of the first derivative of shear stress at both ends of the overlap. This yields the following:

$$\frac{d\tau_{xy}}{dx} = \frac{G_a}{t} \left[\frac{(1-\mu_1^2)}{E_1 h_1} \left(T_1 - \frac{6M_1}{h_1} \right) - \frac{(1-\mu_2^2)}{E_2 h_2} \left(T_2 + \frac{6M_2}{h_2} \right) \right] = b_{1,2} \quad (\text{B.17})$$

where b_1 and b_2 are constant and represent the first derivative of shear stress at $x = 0$ and at $x = L$, respectively. Additionally, a third boundary condition is considered, which equates the integral of shear stress over the overlap length to the net tensile loading in the upper adherend, as follows:

$$\int_0^L \tau_{xy} dx = T_{12} - T_{11} = b_3 \quad (\text{B.18})$$

A system of three simultaneous equations are then produced for the constants $B_1 - B_3$, which are solved by:

$$B_1 = \left[\frac{b_2 - b_1 \cosh(K_6 L)}{K_6 \sinh(K_6 L)} \right] \quad (\text{B.19})$$

$$B_2 = \frac{b_1}{K_6} \quad (\text{B.20})$$

$$B_3 = \frac{b_3}{L} - \frac{(b_2 - b_1)}{K_6^2 L} \quad (\text{B.21})$$

Appendix C

***MATLAB* implementation: Bigwood and Crocombe's model**

C.1 Main code

```
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%% ✓
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%Ana Mafalda Couto
%FEUP - Faculty of Engineering of University of Porto

%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%% ✓
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
clc
clear
clear all
close force
format long
sym x ;

%% Set up the Import Options and import the data
opts = spreadsheetImportOptions("NumVariables", 4);

% Specify sheet and range
opts.Sheet = "Sheet1";
opts.DataRange = "A1:D6860";

% Specify column names and types
opts.VariableNames = ["x_peel", "y_peel", "x_shear", "y_shear"];
opts.VariableTypes = ["double", "double", "double", "double"];

% Import the data
tbl = readtable("D:\1.TESE\3. Numerical\L\no_curve\L_results.xlsx", opts, ✓
"UseExcel", false);

%% Convert to output type
x_peel = tbl.x_peel;
y_peel = tbl.y_peel;
x_shear = tbl.x_shear;
y_shear = tbl.y_shear;

%% Clear temporary variables
clear opts tbl

%Constants
E1 = 12e3; % Young's modulus of adherend 1 (MPa)
E2 = 12e3; % Young's modulus of adherend 2 (MPa)
Ea = 197.09; % Young's modulus of adhesive (MPa)
E = [E1 E2 Ea];

v1 = 0.47; % Poisson's ratio of adherend
v2 = 0.47; % Poisson's ratio of adherend
va = 0.33; % Poisson's ratio of adhesive
v = [v1 v2 va];

Ga=Ea/(2*(1+va)); % (MPa)
w = 15;
P0 = 750;
P0 = P0/w;
```

```

tp2 = 4; % Thickness of adherend 1 (mm)
tp1 = 4; % Thickness of adherend 2 (mm)
ta = 0.01; % Thickness of adhesive layer (mm)
t = [tp1 tp2 ta];

Lo = 30; % Overlap length of the joint (mm)
Lt = 100;
La = 30;
R = 0;

for i = 1:3
    D(i) = (E(i)*(t(i)^3))/(12*(1-(v(i)^2)));
end

T11 = 0;
V11 = 0;
M11 = 0;

T12 = 0;
V12 = 0;
M12 = 0;

T22 = 0;
V22 = -( (Lt-Lo)/2+(tp2/2) )*(P0)/(Lt)
M22 = abs(V22)*((Lt-Lo)/2)

T21 = 0;
V21 = (P0)-abs(V22)
M21 = abs(V21)*((Lt-Lo)/2)

%           1   2   3   4   5   6   7   8   9   10  11  12
loading = [T11,V11,M11,T12,V12,M12,T21,V21,M21,T22,V22,M22];

prompt = "Peel analysis (S22), insert '1'. Shear analysis (S12), insert '2'.";
x = input(prompt)

if x == 1 || x == 2
    if x == 1
        close force
        plot(x_peel,y_peel,'Color',[0.2 0.2 0.2],'LineWidth',2)
        hold on
        % Simplified peel analysis
        [A,K5,sig_V12,sig_M12,peel] = peel(loading,t,v,D,E,Lo,x_peel,y_peel)
        [A_a,K5_a,sig_V12_a,sig_M12_a,peel_a] = peel_a(loading_a,t,v,D,E,Lo,x_peel, y_peel)
        lgd = legend('FEA','LS+AA','FEA+AA','MBF+AA','fontsize',10);
        title('Peel stress distribution','fontsize',12);
        xlabel('{\it x} coordinate','fontsize',12);
        ylabel('Adhesive peel stress (Nmm^{-2})','fontsize',12);
        axis([0 30 -150 450]);
    else
        close force
    end
end

```

```
plot(x_shear,y_shear,'Color',[0.2 0.2 0.2],'LineWidth',2)
hold on
% Simplified shear analysis
[B,K6,tau_T12,tau_V12,tau_M12,shear] = shear(loading,t,v,D,E,Lo,Ga,x_shear, y_shear)
[B_a,K6_a,tau_T12_a,tau_V12_a,tau_M12_a,shear_a] = shear_a(loading_a,t,v,D, E,Lo,Ga,x_shear,y_shear)
axis([0 30 -150 450]);
xlabel('{\it x} coordinate', 'fontsize', 12);
ylabel('Adhesive shear stress (Nmm-2)', 'fontsize', 12);
title('Shear stress distribution', 'fontsize', 12);
lgd = legend('FEA', 'LS+AA', 'FEA+AA', 'MBF+AA', 'fontsize', 10);
end
else
disp('Insert a valid number.')
end
```


C.2 Peel stress distribution

```
function [A,K5,sig_V12,sig_M12,peel] = peel (loading,t,v,D,E,Lo,x_peel,y_peel)

beta1 = 12*E(3)*(1-v(1)^2)/(E(1)*(t(1)^3)*t(3));
beta2 = 12*E(3)*(1-v(2)^2)/(E(2)*(t(2)^3)*t(3));

K5 = ((beta1+beta2)/4)^(0.25);

b1 = E(3)/(2*(K5.^2)*t(3))*((loading(9)./D(2))-(loading(1)./D(1)));
b2 = E(3)/(2*(K5.^2)*t(3))*((loading(12)./D(2))-(loading(6)./D(1)));
b3 = E(3)/(2*(K5.^3)*t(3))*((loading(8)./D(2))-(loading(2)./D(1)));
b4 = E(3)/(2*(K5.^3)*t(3))*((loading(11)./D(2))-(loading(5)./D(1)));

R1 = cosh(K5*Lo)*sin(K5*Lo) - sinh(K5*Lo)*cos(K5*Lo);
R2 = cosh(K5*Lo)*sin(K5*Lo) + sinh(K5*Lo)*cos(K5*Lo);
R3 = cosh(K5*Lo)*sinh(K5*Lo) - cos(K5*Lo)*sin(K5*Lo);
R4 = cosh(K5*Lo)*sinh(K5*Lo) + cos(K5*Lo)*sin(K5*Lo);
R5 = (sinh(K5*Lo))^2 - (sin(K5*Lo))^2;
R6 = (cosh(K5*Lo))^2 - (cos(K5*Lo))^2;

A1 = (b3*R3 - 2*b2*sinh(K5*Lo)*sin(K5*Lo)+b1*R6+b4*R1)/R5;
A2 = (b2*R2 - b3*(sinh(K5*Lo))^2-b1*R4-b4*sinh(K5*Lo)*sin(K5*Lo))/R5;
A3 = (b2*R2-b3*(sin(K5*Lo))^2-b1*R4-b4*sinh(K5*Lo)*sin(K5*Lo))/R5;
A4 = b1;

sig_V12 = -(2)^(0.5)*beta1*loading(5)/((beta1+beta2)^(0.75));
sig_M12 = (-beta1*loading(6))/((beta1+beta2)^(0.5));
peel = sig_V12 + sig_M12;

R = [R1 R2 R3 R4 R5 R6];
A = [A1 A2 A3 A4];

%Plot
sig_y = @(x) (A(1)*cos(K5*x)*cosh(K5*x)+A(2)*cos(K5*x)*sinh(K5*x)+A(3)*sin(K5*x)
*cosh(K5*x)+A(4)*sin(K5*x)*sinh(K5*x));
fplot(@(x) (sig_y(x)), 'Color', [0.65 0.65 0.65], 'LineWidth', 2.5);
hold on

end
```

C.3 Shear stress distribution

```
function [B,K6,tau_T12,tau_V12,tau_M12,shear] = shear>Loading,t,v,D,E,Lo,Ga,↵
x_shear,y_shear)

alpha1 = Ga * (1-(v(1)^2))/(E(1)*t(1)*t(3));
alpha2 = Ga * ((1-(v(2)^2))/(E(2)*t(2)*t(3)));

K6 = 2*(alpha1+alpha2)^(0.5);

b1_shear = Ga/t(3)*((1-(v(1))^2)/(E(1)*t(1))*(loading(1)-6*loading(3)/t(1))-(1-v(2)↵
^2)/(E(2)*t(2))*(loading(7)+6*loading(9)/t(2)));
b2_shear = Ga/t(3)*((1-(v(1))^2)/(E(1)*t(1))*(loading(4)-6*loading(6)./t(1))-(1-v↵
(2))^2)/(E(2)*t(2))*(loading(10)+(6*loading(12)./t(2))));
b3_shear = loading(4)-loading(1);

B1 = (b2_shear-b1_shear*cosh(K6*Lo))/(K6*sinh(K6*Lo));
B2 = b1_shear./K6;
B3 = (b3_shear./Lo)-((b2_shear-b1_shear)/((K6.^2)*Lo));
B = [B1 B2 B3];

tau_T12 = -alpha1*loading(4)/(2*(alpha1+alpha2)^(0.5))
tau_V12 = 3*loading(5)/(4*t(1))
tau_M12 = 3*alpha1*loading(6)/(t(1)*(alpha1+alpha2)^(0.5))
shear = tau_T12+tau_V12+tau_M12

% Plot
tau_xy = @(x) (B(1)*cosh(K6*x) + B(2)*sinh(K6*x) + B(3));
fplot(@(x) (tau_xy(x)), 'Color',[0.65 0.65 0.65], 'LineWidth',2.5);
hold on

end
```

Appendix D

Paper 1

Analytical modeling of single lap joints, L-joints, and T-joints with bio-based materials for the automotive industry

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Abstract: Embracing sustainable practices can be greatly enhanced by exploring the potential of bio-based materials for adhesive joints. However, ensuring the successful implementation of these materials necessitates the provision of suitable tools and resources to facilitate effective design processes. The existing literature primarily focuses on methodologies for studying specific types of joint configurations. The approach of Bigwood and Crocombe stands out because it can be applied to a wider range of joint types, as long as end-loading values in the overlap edges are known. This work was focused on the automotive industry, where the most common joint geometries are single-lap joints, L-joints and T-joints. Thus, the problem of these joints under tensile loading was analyzed. An analytical study is performed using the methodology developed by Bigwood and Crocombe. The results in terms of peel and shear stress were compared with the ones obtained numerically to validate the results. This work aims to validate the accuracy of the results before creating a novel modular method that facilitates a fast and flexible design process for bonded vehicle structures. This approach would give users the ability to specify the overall dimensions of the structure and select the bio-based materials to be employed. The ultimate software employs a method analogous to that used by the *JointDesigner* web tool, which has been successfully developed to make it easier to design the most widely used joint geometry. The analytical method used proves its suitability for studying bio-based joints by accurately predicting critical stresses in different joint configurations, such as the SLJ, L-joint, and T-joint.

Keywords: SLJ, L-joint, T-joint, analytical modeling, finite element analysis, wooden bio-adhesive joint

1 Introduction

Adhesive joints have been constantly gaining popularity in several industries of the transportation sector, including the automotive, naval, aeronautical, and aerospace sec-

tors. In these sectors, environmental aspects are increasingly being considered and to decrease the environmental impact of the final structures, it has become important to use lightweight and sustainable structural components. Currently, adhesives and composite materials primarily rely mostly on petroleum-based sources. However, this dependency gives rise to some challenges. Firstly, there are health and safety concerns associated with potentially toxic components present in synthetic materials. Additionally, the extensive use of these petroleum-based materials contributes significantly to environmental pollution. The unique characteristics of natural materials necessitate special attention during the design phase of bonded structures because these materials differ from the ones currently employed. Thus, it becomes advantageous to equip designers with suitable modeling tools to address this specific requirement effectively.

Designing adhesive joints requires a comprehensive understanding of load transfer within the joint and the ability to predict the resulting stress distribution in the adhesive layer and adherends. Two commonly employed approaches for achieving this aim are analytical modeling and finite element analysis (FEA). FEA offers a more complete evaluation of stress distribution in adhesive bonds with various shapes. It enables detailed simulations and can accommodate complex geometries, providing valuable insights into the behavior of adhesive joints. However, it demands expertise and computational resources. As an alternative, analytical models exist in the literature. These models provide mathematical expressions that can compute the stress distribution in adhesive bonds. Analytical models offer simplicity, fast calculations, and provide valuable insights into stress distribution, making them suitable for initial design assessments and quick estimations [1].

Several different analytical models have been reported in the literature, starting from the pioneering ones proposed by Volkersen [2] and Goland and Reissner [3]. These models perform an elastic analysis of the bonded joint to study single-lap joints (SLJ). These models adopt the representation of the adherends as beams and the adhesive layer as smeared springs. While Volkersen's model only takes into account shear stresses, Goland and Reissner's model provides both shear and peel stress distributions. Some decades later, Hart-Smith [4] introduced an analysis that investigates the impact of multiple factors on the strength of adhesive bonded single-lap joints. Hence, a more realistic bending stress distribution across the end of the overlap region, resulting in an improved bending moment factor that better reflects real-world conditions. Additionally, stiffness imbalance between different adherends and the influence of laminated filamentary composite adherends were considered. This approach also allowed for the study of different joint configurations, such as double-lap joints, scarf, and stepped-lap joints. In a subsequent work, Renton and Vinson [5] introduced a flexible analysis approach that accounted for the equilibrium of the adhesive separately. This methodology enabled the modeling of a stress-free state at the end of the adhesive within the overlap region. Furthermore, the comprehensive analysis

was utilized to investigate the impact of different parameters on the stress distributions of both the adhesive and adherends in a single lap joint. Grant and Taig [6] evaluated different analytical techniques for determining adhesive stress distributions in bonded joints. They enhanced Volkersen’s shear-lag technique with the inclusion of the non-linear material properties of the adhesive. Additionally, a general analysis technique for stepped-lap joints was developed. Ojalvo and Eidinoff [7] extended the analysis of the single-lap joint by including both linear variations in shear stress across the adhesive layer and peel stress. Bigwood and Crocombe [8] proposed an extension of the Goland-Reissner model that can be applied to any joint geometry. The end-loading values need to be specified on each overlapping edge. Hence, this model enables analyzing different joint configurations using a sandwich-type analysis. The different configurations are simulated by imposing different end loading values on each side of upper and lower adherend.

Yang and Pang [9] have utilized the first-order shear deformation theory (FSDT) for SLJs. The objective was to take into account shear deformations, which are relevant in adherends made of fiber-reinforced plastics. Das Neves et al. [10] employed it for SLJs with mixed adhesives. Tsai, Morton, and Oplinger [11] used the FSDT for SLJ and double-lap joints. Yang and Pang’s analysis also accounted for the effect of bending-extension coupling in asymmetrically laminated adherends, requiring the solution of a system of 12th-order differential equations using Fourier series. Over the last years, several more sophisticated models have been proposed [12–18]. Furthermore, a comprehensive summary of available closed-form analytical models can be found in da Silva et al. [19].

Recently, more generalized analytical approaches have emerged. In 2014, Liu et al. [20] proposed an efficient analytical solution for determining adhesive stresses in both balanced and unbalanced adhesively bonded joints (e.g. balanced and unbalanced stiffened plate/joint, single-strap joint, and single-lap joint). It can handle mixed force loading and/or displacement boundary conditions, providing a comprehensive analysis for a wide range of scenarios. The model might be extended in order to predict the mechanical behavior of similar bonded structures, such as composite laminates, electronics packaging, and flexible electronics structures. In the same year, Weißgraeber et al. [21], introduced a versatile sandwich-type model, building upon Bigwood and Crocombe’s method. This model offers a broader scope, enabling analysis of diverse joint configurations and is suitable to study both isotropic and laminated adherends. The bending-extension coupling consideration sets this model apart from the original one. Hence, it allows for the investigation of asymmetrically laminated adherends. In 2022, Garcia Momm et al. [22] introduced two closed-form models based on continuum mechanics to evaluate the stress of bonded joints. These models were developed to analyze the stress in bonded joints composed of composite adherends and ductile adhesives under general loading conditions in non-pristine conditions. The foundation of these models relied on a model for linear elastic functionally graded adhesives. However, to account for adhesive plasticity, the

researchers extended the model using both a linear equivalent modulus method and a non-linear deformation theory.

Riccio and Sellitto et al. [23] in 2015 investigated the issue of skin-stringer debonding under a T-pull load condition. The analysis was conducted using the methodology established by Bigwood and Crocombe, demonstrating the applicability of their approach.

The present work focused on studying three typical joint geometries, commonly employed in automotive and other industries: SLJ, L-joints, and T-joints. The problem of loading these joints in tension was analyzed. A theoretical analysis was performed using the analytical model of Bigwood and Crocombe. Afterwards, the peel and shear stress distributions in the overlap region were compared against those obtained numerically, using the finite element method, to validate the analytical model. This work aims to validate a design process method for bonded vehicle structures, focusing on the method's speed and flexibility. The goal was to create a tool that allows users to specify the overall dimensions of the vehicle structure and choose from a range of materials, including bio-based ones for both the adherends and the adhesive. This tool was designed based on a method similar to that used by the *JointDesigner* web tool, which has been proven effective in simplifying the design of commonly used joint geometries. By implementing this method, the tool will empower users to rapidly and efficiently design bonded vehicle structures with the use of sustainable materials.

The methods implemented in this study are described in detail in Section 2. The obtained results are presented and discussed in Section 3, where they are also compared to the solutions obtained from the finite element analysis.

2 Methods

This section describes the methods used in this work. It starts by describing the analytical model employed for calculating the stress distributions in the overlap region of various joint configurations. Subsequently, it outlines the finite element method (FEM) developed to validate this analytical model.

2.1 Bigwood and Crocombe's analytical model

Bigwood and Crocombe's analytical method emerged as a response to the limitations observed in pre-existing analytical models, the majority of which were only applicable to a specific joint configuration. Therefore, using different computer implementations to determine the optimal joint configuration for a specific application was required. As a result, predicting the stresses in the overlap region with simplicity and satisfactory accuracy was not possible using the previous methods. Bigwood and Crocombe's analytical model was developed to address these shortcomings and provides a more versatile still reasonably

accurate approach to designing bonded joints. This model is suitable for analyzing several joint configurations, subjected to different loading conditions, as shown in Figure 1.

2.1.1 Basic information

Then, the model focuses on the overlap region of those joints, as shown in Figure 2. In the general case, the joint is subjected to axial forces, shear forces, and bending moments in the overlap edges. The analysis is thus specified to a basic ‘adherend-adhesive-adherend sandwich’ configuration (Figure 2). The analysis can be extended to incorporate material non-linearity [24].

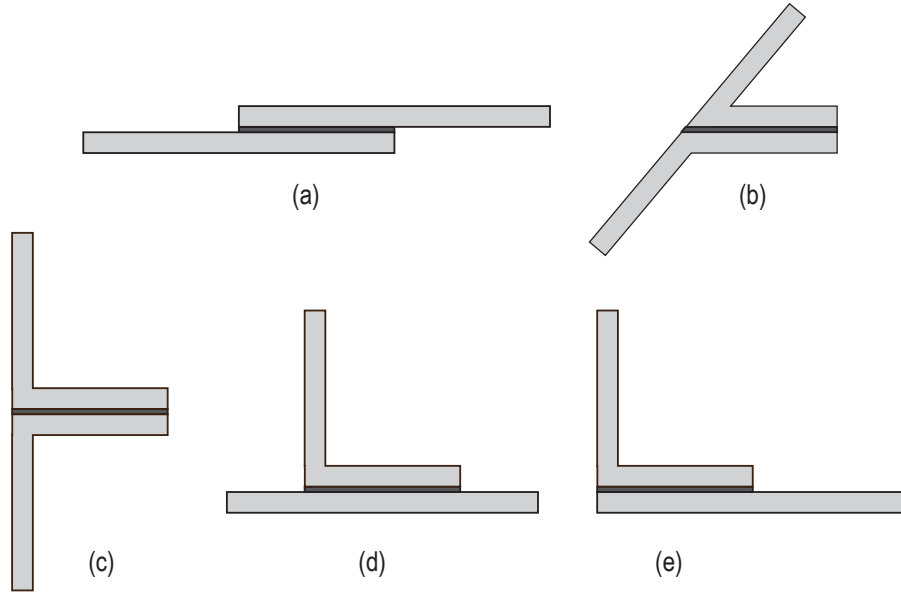


Figure 1: Adhesive joint configurations appropriate for study within Bigwood and Crocombe’s method: (a) single-lap joint, (b) inclined T-peel joint, (c) T-peel joint, (d)-(e) L-joint.

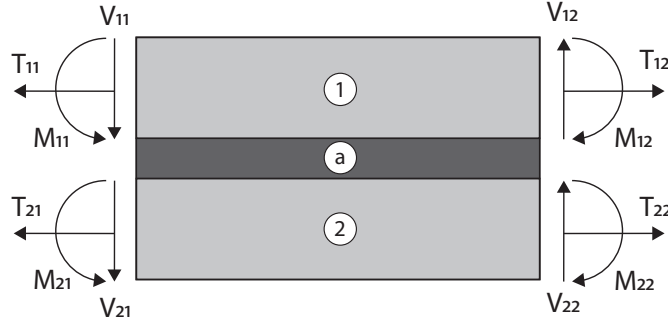


Figure 2: The general adherend-adhesive-adherend sandwich with general loading.

The analytical model presented considers a full elastic analysis. It is obtained by considering an infinitesimally small element of an adherend-adhesive-adherend sandwich beam, as depicted in Figure 3. The two boundaries of this element are loaded through axial forces, shear forces, and bending moments. This loading is transferred from the adherends to the adhesive layer, where peel and shear stresses are developed. Hence, equilibrium equations are derived to balance the applied loads with the adhesive stresses. To establish these equations, it is assumed that the longitudinal direct stress in the adhesive is insignificant compared to the corresponding stress in the adherends. Moreover, the adherends are considered isotropic and may have different thicknesses, as long as they remain constant. The adhesive is also considered isotropic, with a thickness much smaller than that of the adherends. The adherends are considered to be flat plates under bending, assuming a state of plane stress in the x - z plane, neglecting normal stresses along the thickness. Additionally, a plane strain assumption is made in the x - y plane.

Bigwood and Crocombe's model presents two different approaches: general and simplified. The general approach, on the one hand, assumes that the general equations to obtain the peel and shear stress distributions are coupled and may be used to study joints with isotropic, similar or dissimilar adherends. In other words, it requires lots of mathematical calculations but is more suitable for a wider range of applications and with more accuracy. On the other hand, the simplified approach employs a simpler approach although with some limitations, which are highlighted Section 3.2. The simplified approach follows the same elemental diagram (see Figure 3) for adherends in a state of plane strain condition. The simplified peel analysis assumes that the shear stress in the adhesive does not contribute to the flexural behavior of the adherends and because of it is disregarded. In turn, for the case of simplified shear analysis, the adhesive peel stress is neglected. There are mainly two advantages of employing the simplified analysis, which are its ease

of use and the possibility of including non-linear material considerations.

2.1.2 Stress distributions

For the present work, only the elastic analysis was employed, using a two-dimensional sandwich-type model of the overlap region in an adhesive joint. This approach allows for the study of dissimilar adherends under plane-strain in the x - y plane, considering general tensile, shear, and bending moment loading per unit width (Figure 3). Bigwood and Crocombe et al. [8] developed the general form for both simplified peel and shear analysis are given as follows:

$$\begin{aligned} \sigma_y = & A_1 \cos(K_5 x) \cosh(K_5 x) + A_2 \cos(K_5 x) \sinh(K_5 x) \\ & + A_3 \sin(K_5 x) \cosh(K_5 x) + A_4 \sin(K_5 x) \sinh(K_5 x) \end{aligned} \quad (1)$$

$$\tau_{xy} = B_1 \cosh(K_6 x) + B_2 \sinh(K_6 x) + B_3 \quad (2)$$

where $A_1 - A_4$ and K_5 are the elastic explicit constants for the full simplified peel stress analysis, while $B_1 - B_3$ and K_6 denote the elastic explicit constants for the full simplified shear stress analysis. These parameters were established by Bigwood and Crocombe [8].

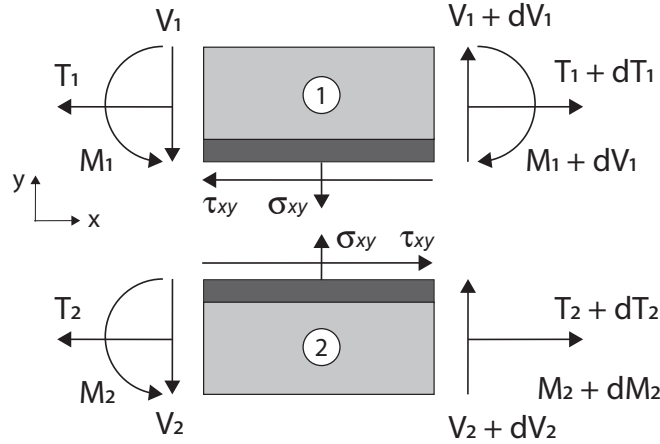


Figure 3: Free body cut of an element of the adherend-adhesive-adherend sandwich under general loading.

2.2 Application of the model to the SLJ, L-joint, and T-joint

As already mentioned, Bigwood and Crocombe's model will be applied for the analysis of three of the most characteristic joint configurations: SLJ (baseline), T-joint, and L-joint. The aim is to check if this model is capable of capturing the stress distribution in

the latter two configurations so that it can be used for the ‘preliminary’ design of modular joints. The three configurations are illustrated in Figure 4.

2.2.1 Boundary conditions

Employing an analytical model requires previous knowledge of the internal forces and moments in the element boundaries. To estimate them, two different approaches were used. The simplest one is to just transfer the external loading to the element boundaries following rigid-body statics, as previously employed by Liu et al. [20] and Weißgraeber et al. [21]. This, however, does not consider the effect of altered load lines due to large bending deformations of the adherends, which, in turn, can lead to nonlinear effects. The second approach comprises of conducting a complete simulation to extract the desired data of the joint using a finite element software. This approach offers a more comprehensive understanding of the joint’s behavior. However, it comes with certain drawbacks since it demands substantial computational resources and it is time-consuming. Moreover, its critical limitation is that the procedure cannot be automated. Nevertheless, this approach may be used as a valuable means to validate the results obtained from alternative methods (e.g. linear statics).

In the case of axially loaded SLJs, several approaches have been proposed that use moment factors to capture this nonlinear effect [3, 4, 25–27]. In the present study, one of the analysis of the SLJ incorporates the utilization of bending moment factors, as described by Zhao et al. [26]. These factors are applied in the following manner:

$$k_1 = \frac{\zeta_2}{\zeta_1 + \zeta_1 \zeta_2 L_0 + \zeta_2} \quad (3)$$

$$k_2 = \frac{\zeta_1}{\zeta_1 + \zeta_1 \zeta_2 L_0 + \zeta_2} \quad (4)$$

where $\zeta_i = \sqrt{P_0/D_i}$ for $i = 1, 2$.

The bending moment factors are then used in the formulation of the internal forces and moments in the overlap region, as follows:

$$M_1 = k_1 \frac{(h_1 + h_2)}{2} P \quad (5)$$

$$M_2 = k_2 \frac{(h_1 + h_2)}{2} P \quad (6)$$

$$Q = \frac{(1 - k_1 - k_2)(h_1 + h_2)P}{L_0} \quad (7)$$

These equations are applied in the boundary conditions of the SLJ. For a general case where the load is applied in the right-side of the upper adherend while the bottom left-side is fixed (see Figure 4(a)), the boundary conditions become:

$$N_1|_{x=L_O} = P_0, \quad Q_1|_{x=L_O} = Q_0, \quad M_1|_{x=L_O} = -M_1 \quad (8)$$

$$N_2|_{x=0} = P_0, \quad Q_2|_{x=0} = Q_0, \quad M_2|_{x=0} = M_2 \quad (9)$$

while the remaining boundary conditions are free, hence equal to zero.

The joint configuration and boundary conditions for the SLJ, L-joint, and T-joint of the present study are illustrated in Figure 4. All joints were considered to be double-supported in the left-side while being simple-supported in the right-side. Furthermore, all the joint configurations were specifically investigated under tension. Table 1 displays the general boundary conditions utilized.

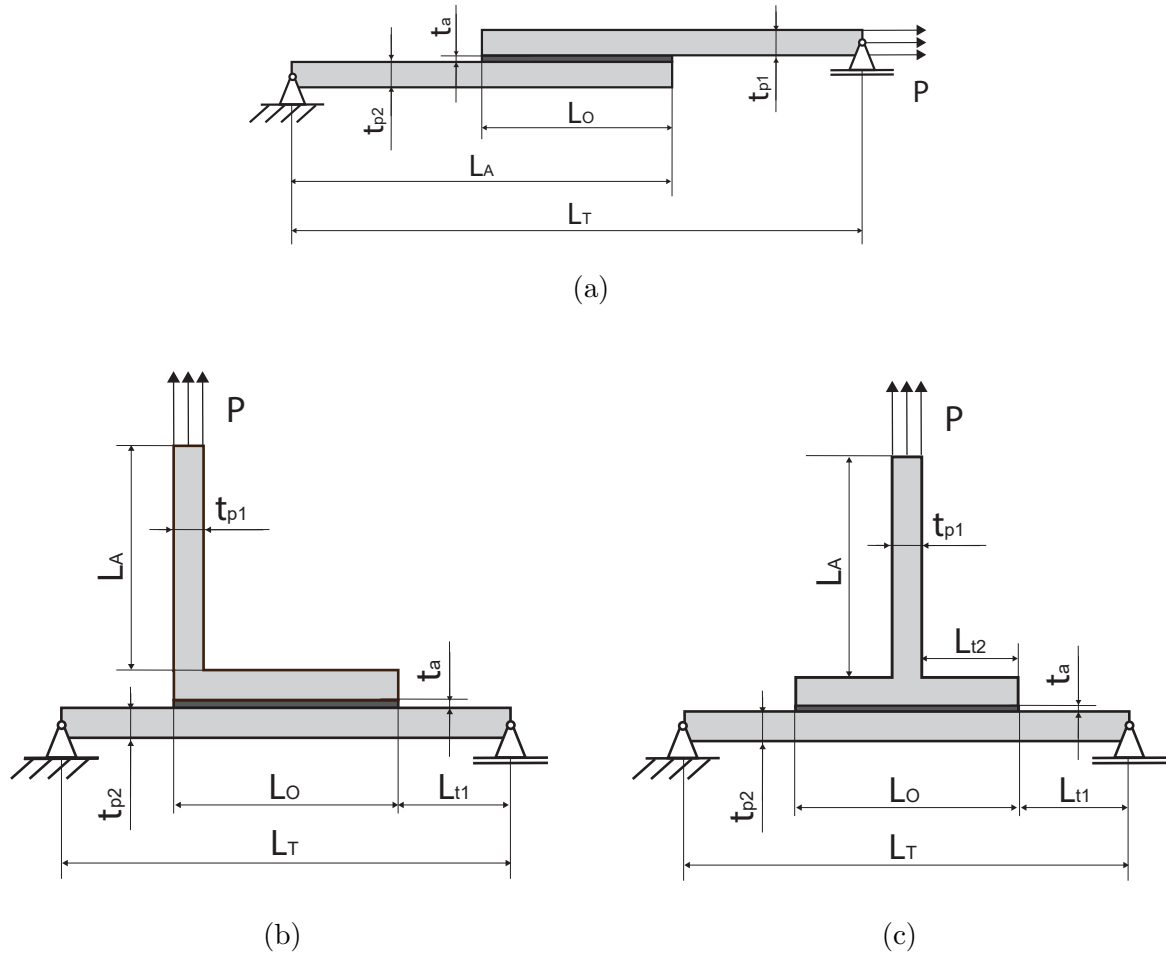


Figure 4: Geometries and boundary conditions: (a) SLJ, (b) L-joint, (c) T-Joint.

Table 1: Boundary conditions for the SLJ, L-joint and T-joint considering the linear statics approach.

		SLJ			L-joint			T-joint		
		T	V	M	T	V	M	T	V	M
Adherend 1	$x = 0$	0	0	0	0	0	0	0	0	0
	$x = L_O$	T_{12}	V_{12}	M_{12}	0	0	0	0	0	0
Adherend 2	$x = 0$	T_{21}	V_{21}	M_{21}	0	V_{21}	M_{21}	0	V_{21}	M_{21}
	$x = L_O$	0	0	0	0	V_{22}	M_{22}	0	V_{22}	M_{22}

2.2.2 Material orthotropy

The initial development of the analytical model assumed isotropic adherends. However, this study specifically concentrates on bio-based materials, with a special emphasis on wood as the adherend. Wood, unlike isotropic materials, exhibits orthotropic behavior. Consequently, in order to adapt the analytical model to this study, adjustments were made to the flexural rigidity to ensure its applicability. To achieve this, Equation 10, as presented in [28], was employed. This equation accounts for the orthotropic nature of the material and was incorporated to better represent the material behavior.

$$D_i = \frac{E_{1,i}h_i^3}{12(1 - \nu_{12,i}\nu_{21,i})} \quad (10)$$

where $i = 1, 2$ and represents the upper or bottom adherends, respectively.

2.3 Finite element model

A two-dimensional finite element model was developed to simulate the overall joints in order to validate the analytical results obtained. The Abaqus 2022[®] software was used to carry out the analysis.

Section 2.2.1 previously exposed the joint configuration and boundary conditions for the SLJ, L-joint, and T-joint (Figure 4). To provide more specific information, Table 2 provides the relevant geometric parameters and describes the loading case applied. Furthermore, Table 3 outlines the material properties used in the present study. Regarding the material orientation, the fibers were simulated as being horizontally oriented. It is important to note that the bio-adhesive utilized in the study cures at zero thickness, which led to its simulation as an extremely thin adhesive layer. The adhesive layer is represented by four elements throughout its thickness. The final mesh was built with high density

to ensure convergence of the peel and shear stresses. The mesh elements selected were 4-node isoparametric plane strain finite elements (CP4R in Abaqus). Figure 5 present the mesh details.

Table 2: Geometric parameters [mm] and loading case [N/mm^2].

	L_T	L_O	L_A	L_{t1}	L_{t2}	t_{p1}	t_{p2}	t_a	w	P
SLJ	175	25	100	-	-	4	4	0.01	25	62.5
L-Joint	100	30	30	35	-	4	4	0.01	15	12.5
T-Joint	100	30	30	35	13	4	4	0.01	15	12.5

Table 3: Material properties.

	Young's modulus [MPa]	Poisson's ratio [-]	Shear modulus [MPa]
Adherend	$E_1 = 12000$	$\nu_{12} = 0.47$	$G_{12} = 1120$
	$E_2 = 1900$	$\nu_{13} = 0.47$	$G_{13} = 1040$
	$E_3 = 1100$	$\nu_{23} = 0.57$	$G_{23} = 290$
Adhesive	$E_a = 197.09$	$\nu_a = 0.33$	$G_a = 74.1$

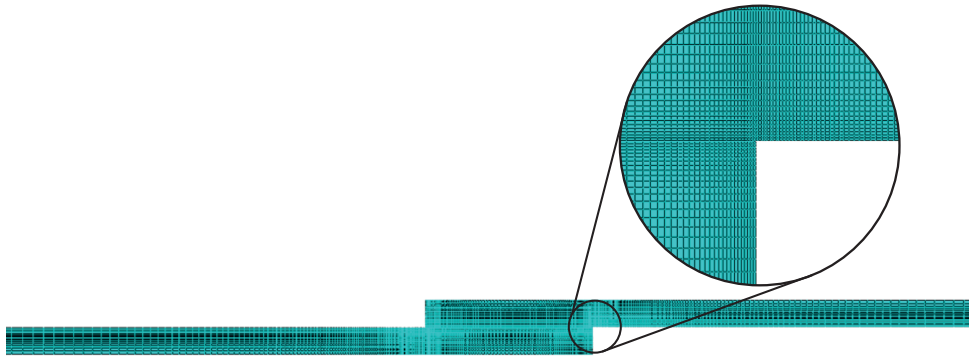
3 Results and discussion

This section compares the stress distributions obtained from Bigwood and Crocombe's analytical method with the respective ones predicted from the FEA. The stress distribution diagrams are presented for the SLJ, L-joint, and T-joint. Next, a discussion of the limitations of the analytical method and an outlook are provided.

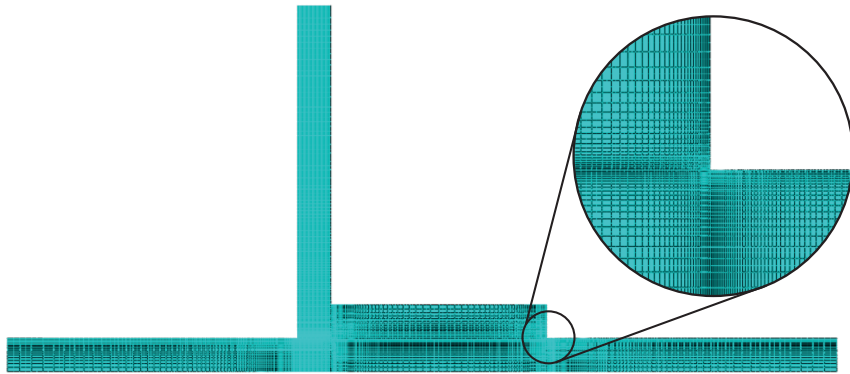
3.1 Comparison of analytical and numerical results

3.1.1 Analytical model's constants

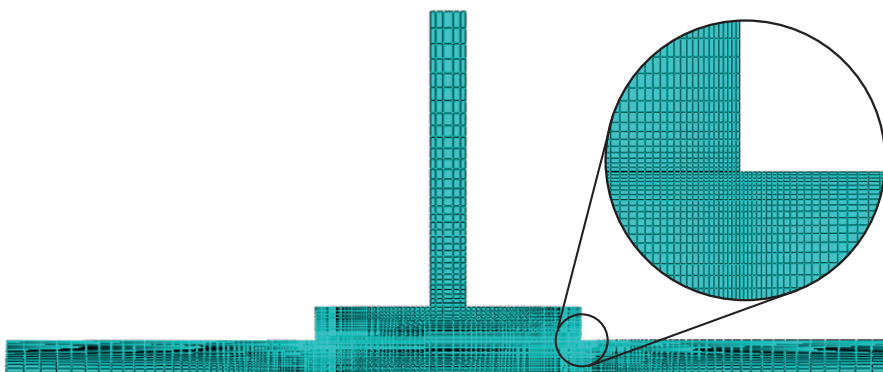
The results of the elastic explicit constants obtained for both peel and shear analysis, for all study cases, are presented in Tables 4 and 5. For the sake of simplicity, the linear statics approach will be referred to as LS, and the bending moment factor approach as BMF, from this point onward.



(a)



(b)



(c)

Figure 5: Mesh details: (a) SLJ, (b) L-joint, (c) T-Joint.

Table 4: Elastic analyses explicit constants for the simplified peel analysis.

		A_1	A_2	A_3	A_4	K_5
SLJ	LS	152.2	-152.2	-148.8	148.8	0.5885
	FEA	87.84	-87.84	-85.84	85.84	0.5885
	BMF	112.1	-112.1	-102.5	102.5	0.5885
L-Joint	LS	400.4	-400.4	-381.9	381.9	0.5885
	FEA	357.6	-357.6	-379.0	379.0	0.5885
T-Joint	LS	317.8	-317.8	-303.1	303.1	0.5885
	FEA	310.6	-310.6	-302.2	302.2	0.5885

Table 5: Elastic analyses explicit constants for the simplified shear analysis.

		B_1	B_2	B_3	K_6
SLJ	LS	109.7	-109.7	1.055	0.9809
	FEA	74.87	-74.87	3.625	0.9809
	BMF	85.08	-85.08	3.061	0.9809
L-Joint	LS	202.8	-202.8	-2.844	0.9809
	FEA	196.9	-196.9	-2.981	0.9809
T-Joint	LS	160.9	-160.9	0	0.9809
	FEA	156.9	-156.9	0	0.9809

3.1.2 Single-lap joint (SLJ)

The symmetric SLJ was thoroughly simulated using finite element software, allowing for the direct retrieval of the stress distribution within the adhesive. Subsequently, the analytical analysis were conducted. As detailed in Section 2.2.1, three different approaches were suggested as viable options to obtain a estimation of the internal forces required as input for the analytical analysis. Consequently, all three methods were employed, and their respective outcomes are presented in Table 6.

Table 6: Comparison between internal forces obtained from different sources: LS (linear statics), FEA (finite element analysis) and application of the BMF (bending moment factors).

Internal forces	LS	FEA	BMF
T_{11}	0	7.077	0
V_{11}	0	3.270	0
M_{11}	0	-9.894	0
T_{12}	250	250.1	250
V_{12}	5.729	7.400	16.33
M_{12}	-429.6	-253.3	-295.9
T_{21}	250	250.2	250
V_{21}	5.729	6.681	16.33
M_{21}	429.6	254.9	295.9
T_{22}	0	6.956	0
V_{22}	0	3.311	0
M_{22}	0	9.750	0

Figure 6 presents the stress distribution in the overlap region of a symmetric Single Lap Joint (SLJ), utilizing both Finite Element Analysis (FEA) and analytical analysis. Three different analytical approaches were employed to determine the internal forces at the overlap edges, as described earlier. The first analytical analysis involved utilizing the internal forces obtained from linear statics modeling. In this method, the internal forces were calculated using simplified assumptions and equations derived from basic principles of mechanics. The second analytical analysis corresponds to implementing the internal forces calculated from FEA results. By utilizing the more accurate and detailed information obtained from FEA simulations, a more precise estimation of the internal forces within the joint can be achieved. The third analysis involves applying the bending moment factors.

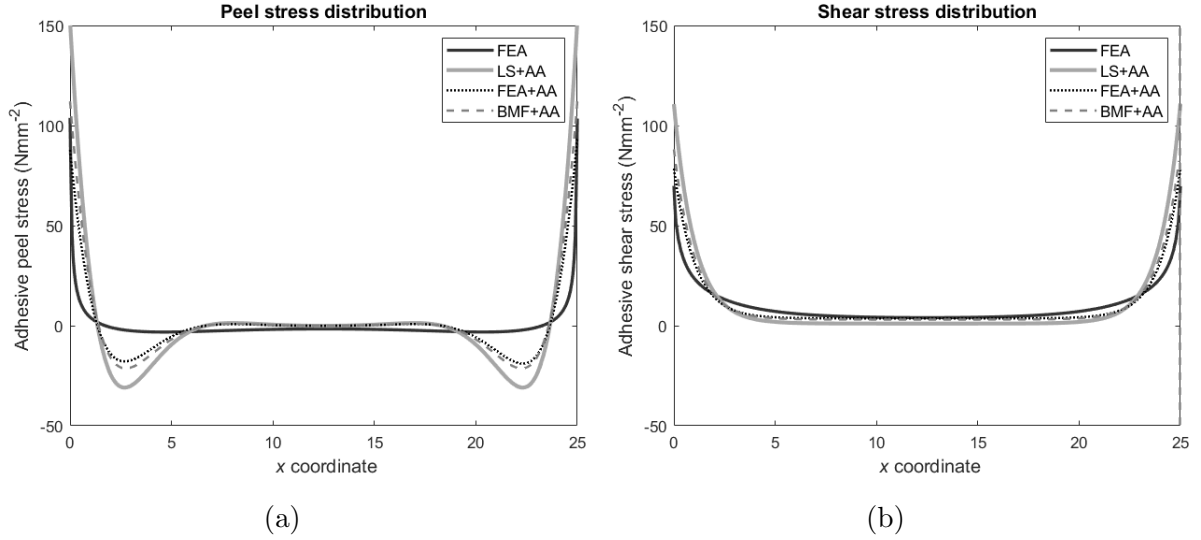


Figure 6: Stress distribution in a symmetric single-lap joint with pine wood adhered and bio-adhesive: (a) Peel stress, (b) Shear stress.

Noticeable differences are observed among the various implemented approaches. It is evident that there is a lack of agreement between the FEA and the analytical models, particularly in the peel stress distribution plots (Figure 6(a)).

This discrepancy can possibly stem from the adhesive utilized in the study, which cures only when used in zero thickness layers, resulting in an extremely thin adhesive layer. To further investigate and validate this observation, an additional study was conducted. The adhesive thickness study (Figure 7) was carried out by systematically increasing the adhesive thickness while keeping all other parameters constant. As a result, this scenario has been identified as the underlying cause of the observed differences.

Furthermore, it is important to highlight that the usage of linear statics modeling results in significantly higher stress values compared to the other methods, as anticipated based on the findings presented in Table 6. In contrast, both the FEA-based approach and the implementation of bending moment factors yield more accurate and realistic stress distributions. The remarkable accuracy achieved by employing FEA and bending moment factors in analyzing the stress distribution highlights their greater effectiveness at capturing the behavior of the joint.

Moreover, it is important to highlight that the most critical stresses, which will govern the failure of the joint, registered at the boundaries of the overlap region ($x = 0$ and $x = 25$), exhibit a good agreement across all studies, both in terms of peel and shear stress analysis, with the exception of the analysis derived from the linear static approach. However, since the primary objective of this study is to develop a pre-joint design tool, this discrepancy does not pose a significant concern. This is because, in the critical regions, the stress values are very similar, indicating a high level of consistency that is suitable for the tool's development.

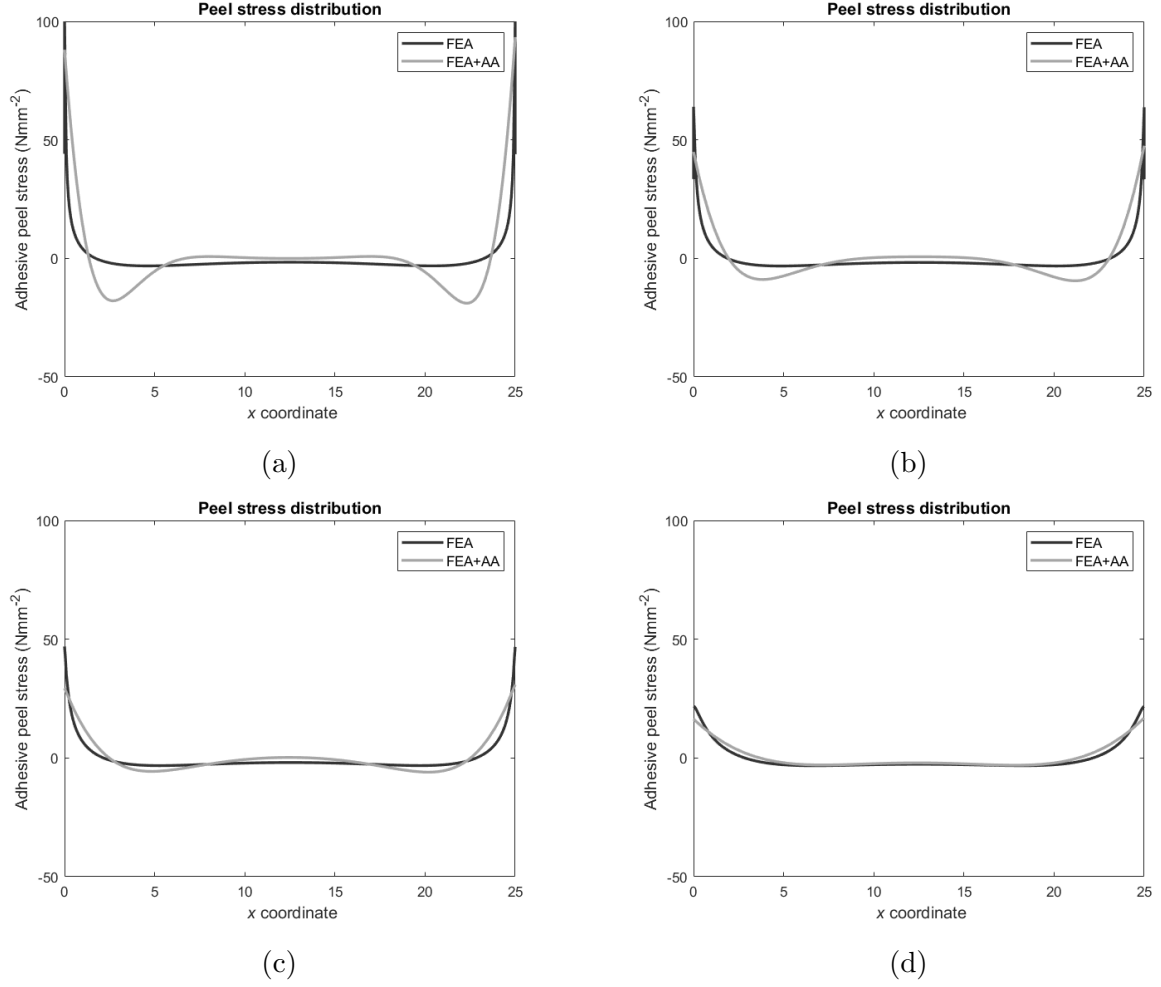


Figure 7: Peel stress distribution in a symmetric single-lap joint with pine wood adhered and bio-adhesive with thickness: (a) 0.01 mm, (b) 0.04 mm, (c) 0.1 mm, (d) 0.4 mm.

3.1.3 L-joint

An L-joint subjected to opening loading is now studied. Table 7 provides a comparison between the internal forces derived from the finite element software (Abaqus) and linear statics modeling. It is important to note that, for this particular joint geometry, the previously mentioned bending moment factors approach is not applicable.

Table 7: Comparison between internal forces retrieved from Abaqus and linear statics modeling for L-joint.

Internal forces	FEA	LS	Variation [%]
T_{11}	16.77	0	-
V_{11}	9.06	0	-
M_{11}	-23.38	0	-
T_{12}	10.19	0	-
V_{12}	-5.586	0	-
M_{12}	-14.05	0	-
T_{21}	-8.807	0	-
V_{21}	27.21	31.5	15.8
M_{21}	1111	1102.5	-0.77
T_{22}	-5.279	0	-
V_{22}	-15.93	-18.5	-16.1
M_{22}	653.1	647.5	-0.86

The stress distribution in the overlap region, as calculated from the FEA and the analytical model, is compared in Figure 8. In the case of this specific geometry, both analytical analyses demonstrate excellent agreement, as anticipated, since the internal forces obtained in both cases are quite similar. However, there is a noticeable discrepancy in the peel stress distribution between the FEA and analytical analysis. It is even more pronounced compared to the previous case (SLJ study). Nevertheless, in the most critical region (overlap edges), the values remain relatively similar. This consistency indicates that the methods are validated for the intended purpose. Regarding the shear stress distribution, a high degree of agreement is observed throughout the entire overlap region. Furthermore, both plots exhibit an asymmetric distribution of shear and peel stress, characterized by localized peaks at the ends of the overlapping region.

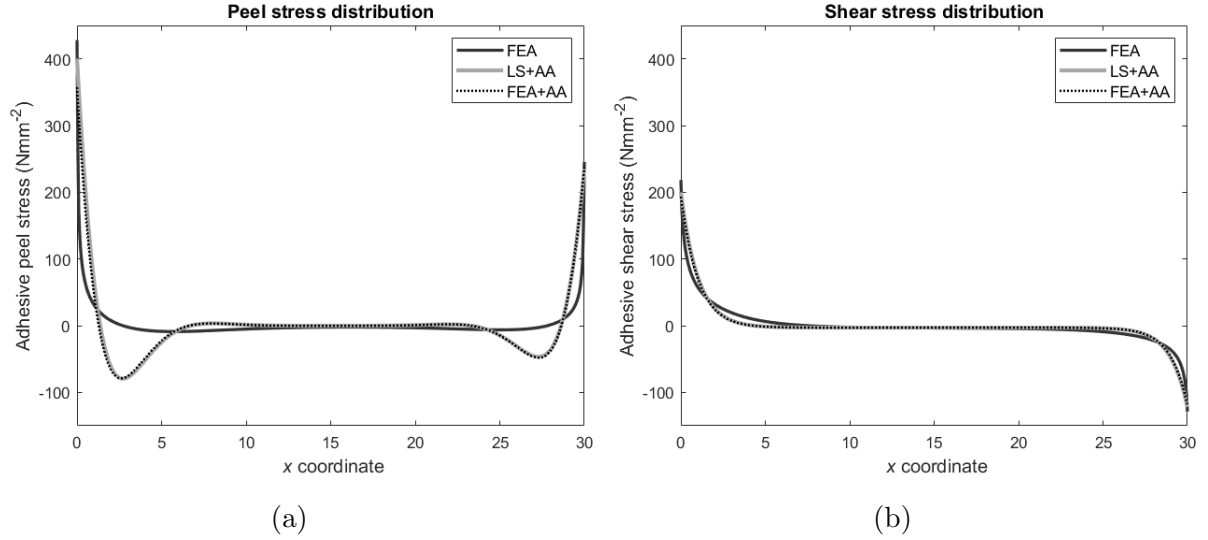


Figure 8: Stress distribution in a L-joint with pine wood adhered and bio-adhesive:(a) Peel stress, (b) Shear stress.

3.1.4 T-joint

A T-joint under a tensile loading has also been studied. Table 8 displays the comparison between the internal forces obtained. The adhesive stress distributions are presented in Figure 9. The plots are very similar to the ones presented above for the L-joint. However, the peel stress distribution symmetric in this case. Also, it is noted that the local peaks at the end of the overlap region reduced significantly in relation to the ones found for the L-joint.

Table 8: Comparison between internal forces retrieved from Abaqus and linear statics modeling for T-joint.

Internal forces	FEA	LS	Variation [%]
T_{11}	13.73	0	-
V_{11}	7.409	0	-
M_{11}	-18.60	0	-
T_{12}	13.48	0	-
V_{12}	-7.561	0	-
M_{12}	-18.31	0	-
T_{21}	-7.620	0	-
V_{21}	21.71	25	15.3
M_{21}	886.2	875	-1.26
T_{22}	-7.759	0	-
V_{22}	-21.52	-25	-16.2
M_{22}	886.3	875	-1.27

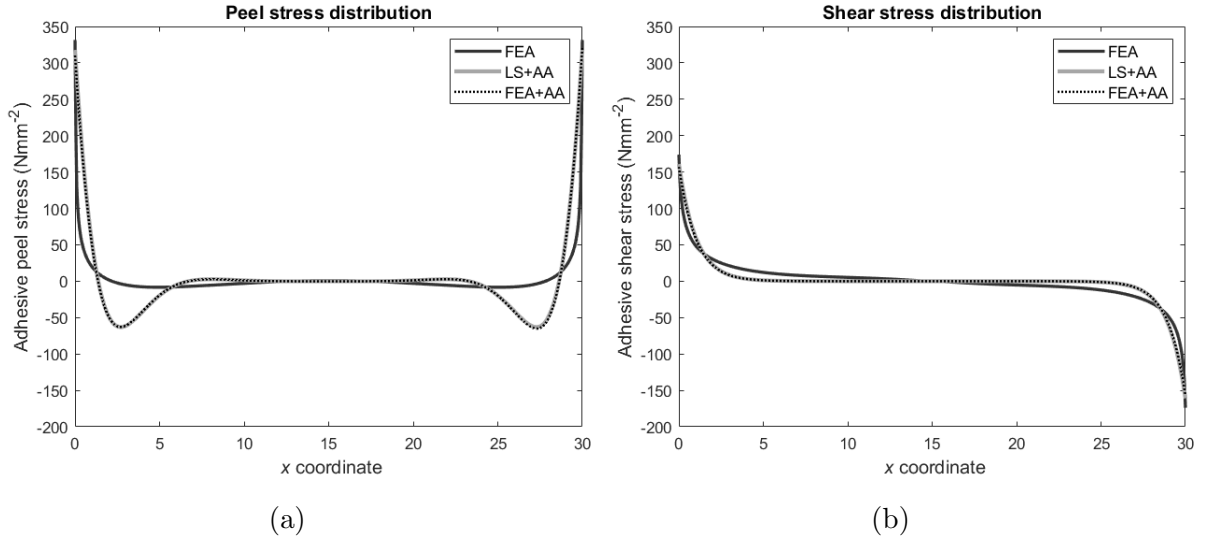


Figure 9: Stress distribution in a T-joint with pine wood adhered and bio-adhesive: (a) Peel stress, (b) Shear stress.

3.2 Discussion

The aim of this study was to validate the results of the analytical model and develop a joint design tool. In order to achieve this, it was essential to verify the accuracy and

reliability of the generated results, which will enable more effective decision-making during the design process.

One important limitation of this study concerns the employment of Bigwood and Crocombe’s simplified design analysis. The reason is the inherently reduced accuracy resulting from the simplifying assumptions made during its derivation. Since it relies on simplifications, the analysis is susceptible to limitations and inaccuracies. The first problem occurs when the hyperbolic and trigonometric arguments (e.g. K_5L_O, K_6L_O) are large. The second is related to the disregard of the coupling effect of the peel and shear stresses for the case of dissimilar adherends.

It is crucial to emphasize that the analytical model relies on previous knowledge of the internal forces and moments acting on the element boundaries. In practical applications, the available information is limited to global loads and moments. Consequently, the analytical approach necessitates additional calculations and approximations to determine the internal forces and moments within the region of interest. Three different approaches were utilized for the SLJ analysis. It was observed that incorporating the bending moment factors resulted in excellent agreement between the stress distribution and critical stresses obtained from the analytical study performed through the finite element software calculations for the internal forces in the overlap edges. In contrast, the linear statics approach did not perform as well for this specific geometry. Therefore, it can be concluded that the bending moment factors can be successfully incorporated for the SLJ analysis. In the case of L-joint and T-joint configurations, where only the linear statics approach is applicable, it has demonstrated favorable results. However, it is important to acknowledge that the linear statics approach simplifies the analysis and disregards several influential factors, which can result in inaccuracies and discrepancies when compared to real-world scenarios. Despite its limitations, the linear statics approach still produced satisfactory results, exhibiting good agreement for the most critical stresses in these geometries.

Another source of discrepancy between the results of the analytical model and the finite element analysis is the small adhesive thickness, resulting in inaccurate calculations. However, as the adhesive thickness is increased, there is a noticeable improvement in the agreement between the plots, particularly in terms of the peel stress distribution. Additionally, the analysis conducted in this study was based on the assumption of elastic behavior of the materials. To broaden the range of applications and make this study more applicable to real-world situations, it is possible to include Crocombe’s nonlinear analysis. This would further improve the accuracy and reliability of the results.

Despite the simplifications adopted, the effectiveness of this analytical model has been demonstrated, making it a suitable tool for obtaining reasonable accurate results for bio-based joints, while also offering the advantage of computational efficiency. By employing this analytical model, engineers and researchers can easily analyze and evaluate the performance of bio-based joints without the need for time-consuming and computationally

expensive simulations.

4 Conclusion

The present study utilized an analytical model to investigate the stress distributions in various bio-based joints with different configurations, including SLJs, L-joints, and T-joints. Certain simplifications were made in the analytical model to facilitate the analysis, enabling a more straightforward calculation process and offering initial insights into the stress distribution within the joints. However, it is important to acknowledge that these simplifications may introduce limitations to the accuracy of the results.

In this study, the stress distributions obtained from the analytical model deviated from the initially expected values. Nevertheless, a comparison between the critical stress obtained from the analytical model and finite element analysis (FEA) revealed a high degree of similarity in the calculations. Considering the advantages of simplicity and efficiency, the analytical model can be effectively employed in practical applications, providing valuable insights into the stress distribution of joints during the initial design stage.

It is important to note, however, that for more precise results and a comprehensive understanding of joint behavior, additional validation through experimental testing or more advanced numerical methods may be necessary. Nonetheless, the findings presented herein offer valuable information for the design and optimization of joint configurations, particularly when utilizing wooden substrates with bio-based adhesives.

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Appendix E

***Abaqus* plugin**

```

1  from abaqus import *
2  from abaqusConstants import *
3  from caeModules import *
4  from regionToolset import *
5  import visualization
6  import xyPlot
7  import displayGroupOdbToolset as dgo
8  from driverUtils import executeOnCaeStartup
9  import os
10 from abaqus import mdb, session
11 from odbAccess import *
12 import part
13 import assembly
14 from textRepr import *
15 import numpy as np
16
17 def createLJ(L_T,L_A,L_O,t_P1,t_P2,t_A,w,Load,smallerSize,biggerSize,partName):
18     s = mdb.models['Model-1'].ConstrainedSketch(name='__profile__',
19         sheetSize=200.0)
20     g, v, d, c = s.geometry, s.vertices, s.dimensions, s.constraints
21     s.setPrimaryObject(option=STANDALONE)
22     s.Spot(point=(0.0, 0.0))
23     s.FixedConstraint(entity=v[0])
24     s.rectangle(point1=(0.0, 0.0), point2=(L_T, (t_P1+t_A+t_P2+L_A)))
25     p = mdb.models['Model-1'].Part(name='Part-1', dimensionality=TWO_D_PLANAR,
26         type=DEFORMABLE_BODY)
27     p = mdb.models['Model-1'].parts['Part-1']
28     p.BaseShell(sketch=s)
29     s.unsetPrimaryObject()
30     p = mdb.models['Model-1'].parts['Part-1']
31     session.viewports['Viewport: 1'].setValues(displayedObject=p)
32     del mdb.models['Model-1'].sketches['__profile__']
33     p = mdb.models['Model-1'].parts['Part-1']
34     f, e, d1 = p.faces, p.edges, p.datums
35     s1 = mdb.models['Model-1'].ConstrainedSketch(name='__profile__',
36         sheetSize=218.51, gridSpacing=5.46)
37     g, v, d, c = s1.geometry, s1.vertices, s1.dimensions, s1.constraints
38     s1.setPrimaryObject(option=SUPERIMPOSE)
39     p = mdb.models['Model-1'].parts['Part-1']
40     p.projectReferencesOntoSketch(sketch=s1, filter=COPLANAR_EDGES)
41     s1.Line(point1=((L_T/2)-(L_O/2)), 0), point2=((L_T/2)-(L_O/2)),
42         (t_P1+t_A+t_P2+L_A)))
43     p = mdb.models['Model-1'].parts['Part-1']
44     f = p.faces
45     AllFaces = p.faces.getByBoundingBox(xMin=0.0, yMin=0.0, zMin=0.0,
46         xMax=L_T, yMax=(t_P1+t_A+t_P2+L_A), zMax=0.0)
47     e1, d2 = p.edges, p.datums
48     p.PartitionFaceBySketch(faces=AllFaces, sketch=s1)
49     s1.unsetPrimaryObject()
50     del mdb.models['Model-1'].sketches['__profile__']
51     p = mdb.models['Model-1'].parts['Part-1']
52     f, e, d1 = p.faces, p.edges, p.datums
53     s2 = mdb.models['Model-1'].ConstrainedSketch(name='__profile__',
54         sheetSize=218.51, gridSpacing=5.46)
55     g, v, d, c = s2.geometry, s2.vertices, s2.dimensions, s2.constraints
56     s2.setPrimaryObject(option=SUPERIMPOSE)
57     p = mdb.models['Model-1'].parts['Part-1']
58     p.projectReferencesOntoSketch(sketch=s2, filter=COPLANAR_EDGES)
59     s2.Line(point1=((L_T/2)-(L_O/2)+t_P1), (0.0)), point2=((L_T/2)-(L_O/2)+t_P1),
60         (t_P1+t_A+t_P2+L_A)))
61     p = mdb.models['Model-1'].parts['Part-1']
62     f = p.faces
63     e1, d2 = p.edges, p.datums
64     p.PartitionFaceBySketch(faces=AllFaces, sketch=s2)
65     s2.unsetPrimaryObject()
66     del mdb.models['Model-1'].sketches['__profile__']
67     p = mdb.models['Model-1'].parts['Part-1']
68     f, e, d1 = p.faces, p.edges, p.datums
69     s3 = mdb.models['Model-1'].ConstrainedSketch(name='__profile__',

```

```

68     sheetSize=218.51, gridSpacing=5.46)
69     g, v, d, c = s3.geometry, s3.vertices, s3.dimensions, s3.constraints
70     s3.setPrimaryObject(option=SUPERIMPOSE)
71     p = mdb.models['Model-1'].parts['Part-1']
72     p.projectReferencesOntoSketch(sketch=s3, filter=COPLANAR_EDGES)
73     s3.Line(point1=((L_T/2)-(L_O/2)), (t_P1+t_A+t_P2)), point2=((L_T/2)+(L_O/2)),
74     (t_P1+t_A+t_P2)))
75     p = mdb.models['Model-1'].parts['Part-1']
76     f = p.faces
77     e1, d2 = p.edges, p.datums
78     p.PartitionFaceBySketch(faces=AllFaces, sketch=s3)
79     s3.unsetPrimaryObject()
80     del mdb.models['Model-1'].sketches['__profile__']
81     p = mdb.models['Model-1'].parts['Part-1']
82     f, e, d1 = p.faces, p.edges, p.datums
83     s4 = mdb.models['Model-1'].ConstrainedSketch(name='__profile__',
84     sheetSize=218.51, gridSpacing=5.46)
85     g, v, d, c = s4.geometry, s4.vertices, s4.dimensions, s4.constraints
86     s4.setPrimaryObject(option=SUPERIMPOSE)
87     p = mdb.models['Model-1'].parts['Part-1']
88     p.projectReferencesOntoSketch(sketch=s4, filter=COPLANAR_EDGES)
89     s4.Line(point1=((L_T/2)+(L_O/2)), 0), point2=((L_T/2)+(L_O/2)), (t_P1+t_A+t_P2)))
90     p = mdb.models['Model-1'].parts['Part-1']
91     f = p.faces
92     e1, d2 = p.edges, p.datums
93     p.PartitionFaceBySketch(faces=AllFaces, sketch=s4)
94     s4.unsetPrimaryObject()
95     del mdb.models['Model-1'].sketches['__profile__']
96     p = mdb.models['Model-1'].parts['Part-1']
97     f, e, d1 = p.faces, p.edges, p.datums
98     s5 = mdb.models['Model-1'].ConstrainedSketch(name='__profile__',
99     sheetSize=218.51, gridSpacing=5.46)
100     g, v, d, c = s5.geometry, s5.vertices, s5.dimensions, s5.constraints
101     s5.setPrimaryObject(option=SUPERIMPOSE)
102     p = mdb.models['Model-1'].parts['Part-1']
103     p.projectReferencesOntoSketch(sketch=s5, filter=COPLANAR_EDGES)
104     s5.Line(point1=(0, t_P2+t_A), point2=(L_T, t_P2+t_A)) #
105     p = mdb.models['Model-1'].parts['Part-1']
106     f = p.faces
107     AllFaces = p.faces.getByBoundingBox(xMin=0.0, yMin=0.0, zMin=0.0,
108     xMax=L_T, yMax=(t_P1+L_A+t_P2+t_A), zMax=0.0)
109     e1, d2 = p.edges, p.datums
110     p.PartitionFaceBySketch(faces=AllFaces, sketch=s5)
111     s5.unsetPrimaryObject()
112     del mdb.models['Model-1'].sketches['__profile__']
113     p = mdb.models['Model-1'].parts['Part-1']
114     f, e, d1 = p.faces, p.edges, p.datums
115     s6 = mdb.models['Model-1'].ConstrainedSketch(name='__profile__',
116     sheetSize=218.51, gridSpacing=5.46)
117     g, v, d, c = s6.geometry, s6.vertices, s6.dimensions, s6.constraints
118     s6.setPrimaryObject(option=SUPERIMPOSE)
119     p = mdb.models['Model-1'].parts['Part-1']
120     p.projectReferencesOntoSketch(sketch=s6, filter=COPLANAR_EDGES)
121     s6.Line(point1=(0, t_P2), point2=(L_T, t_P2)) #
122     p = mdb.models['Model-1'].parts['Part-1']
123     f = p.faces
124     AllFaces = p.faces.getByBoundingBox(xMin=0.0, yMin=0.0, zMin=0.0,
125     xMax=L_T, yMax=(t_P1+L_A+t_P2+t_A), zMax=0.0)
126     e1, d2 = p.edges, p.datums
127     p.PartitionFaceBySketch(faces=AllFaces, sketch=s6)
128     s6.unsetPrimaryObject()
129     del mdb.models['Model-1'].sketches['__profile__']
130     p = mdb.models['Model-1'].parts['Part-1']
131     f, e, d1 = p.faces, p.edges, p.datums
132     s7 = mdb.models['Model-1'].ConstrainedSketch(name='__profile__',
133     sheetSize=218.51, gridSpacing=5.46)
134     g, v, d, c = s7.geometry, s7.vertices, s7.dimensions, s7.constraints
135     s7.setPrimaryObject(option=SUPERIMPOSE)
136     p = mdb.models['Model-1'].parts['Part-1']

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136 p.projectReferencesOntoSketch(sketch=s7, filter=COPLANAR_EDGES)
137 s7.Line(point1=(L_T/2-L_O/2, t_P2+t_A+t_P1), point2=(L_T/2-L_O/2+t_P1,
138 t_P2+t_A+t_P1)) #
139 p = mdb.models['Model-1'].parts['Part-1']
140 f = p.faces
141 AllFaces = p.faces.getByBoundingBox(xMin=L_T/2-L_O/2, yMin=t_P2+t_A, zMin=0.0,
142                                     xMax=L_T/2-L_O/2+t_P1,
143                                     yMax=(t_P1+L_A+t_P2+t_A), zMax=0.0)
144
145 e1, d2 = p.edges, p.datums
146 p.PartitionFaceBySketch(faces=AllFaces, sketch=s7)
147 s7.unsetPrimaryObject()
148
149 del mdb.models['Model-1'].sketches['__profile__']
150 p = mdb.models['Model-1'].parts['Part-1']
151 f1 = p.faces
152 removeFaces = p.faces.findAt( ((0, t_P1+t_A/2, 0.0),),
153                                ((0, t_P1+t_A+t_P2, 0.0),),
154                                ((L_T/2+L_O/2, t_P1+t_A/2, 0.0),),
155                                ((L_T/2+L_O/2, t_P1+t_A+t_P2, 0.0),),)
156
157 p.RemoveFaces(faceList = removeFaces, deleteCells=False)
158
159 #####
160 #Material
161 mdb.models['Model-1'].Material(name='Wood')
162 mdb.models['Model-1'].materials['Wood'].Elastic(type=ENGINEERING_CONSTANTS,
163 table=((12000.0, 1910.0, 1010.0, 0.47, 0.51, 0.31, 1120.0, 1040.0, 290.0),
164 ))
165 mdb.models['Model-1'].Material(name='Adhesive')
166 mdb.models['Model-1'].materials['Adhesive'].Elastic(table=((197.09, 0.33), ))
167 #####
168 #Section
169 mdb.models['Model-1'].HomogeneousSolidSection(name='Wood',
170 material='Wood', thickness=w)
171 mdb.models['Model-1'].HomogeneousSolidSection(name='Adhesive',
172 material='Adhesive', thickness=w)
173 #####
174 # Partitions
175 p.PartitionEdgeByPoint(edge=e.findAt(coordinates=(0, t_P2/2, 0)),
176 point=p.InterestingPoint(edge=e.findAt(coordinates=(0, t_P2/2, 0)),
177 rule=MIDDLE))
178 p.PartitionEdgeByPoint(edge=e.findAt(coordinates=(L_T/2-L_O/2, t_P2/2, 0)),
179 point=p.InterestingPoint(edge=e.findAt(coordinates=(L_T/2-L_O/2, t_P2/2, 0)),
180 rule=MIDDLE))
181 p.PartitionEdgeByPoint(edge=e.findAt(coordinates=(L_T/2-L_O/2+t_P1, t_P2/2, 0)),
182 point=p.InterestingPoint(edge=e.findAt(coordinates=(L_T/2-L_O/2+t_P1, t_P2/2, 0)),
183 rule=MIDDLE))
184 p.PartitionEdgeByPoint(edge=e.findAt(coordinates=(L_T/2+L_O/2, t_P2/2, 0)),
185 point=p.InterestingPoint(edge=e.findAt(coordinates=(L_T/2+L_O/2, t_P2/2, 0)),
186 rule=MIDDLE))
187 p.PartitionEdgeByPoint(edge=e.findAt(coordinates=(L_T, t_P2/2, 0)),
188 point=p.InterestingPoint(edge=e.findAt(coordinates=(L_T, t_P2/2, 0)),
189 rule=MIDDLE))
190 p.PartitionEdgeByPoint(edge=e.findAt(coordinates=(L_T/2-L_O/2+t_P1/2,
191 t_P1+t_A+t_P2+L_A, 0)),
192 point=p.InterestingPoint(edge=e.findAt(coordinates=(L_T/2-L_O/2+t_P1/2,
193 t_P1+t_A+t_P2+L_A, 0)),
194 rule=MIDDLE))
195 #####
196 #Sets
197 edg_m_6 = p.edges.getByBoundingBox(xMin=L_T/2+L_O/2, yMin=t_P2+t_A, zMin=0.0,
198                                     xMax=L_T/2+L_O/2, yMax=t_P1+t_A+t_P2, zMax=0.0)
199
200 edg_m_7 = p.edges.getByBoundingBox(xMin=L_T/2-L_O/2+t_P1, yMin=t_P2+t_A, zMin=0.0,
201                                     xMax=L_T/2-L_O/2+t_P1, yMax=t_P1+t_P2+t_A,
202                                     zMax=0.0)
203
204 edg_m_8 = p.edges.getByBoundingBox(xMin=L_T/2-L_O/2, yMin=t_P2+t_A, zMin=0.0,
205                                     xMax=L_T/2-L_O/2, yMax=t_P1+t_P2+t_A, zMax=0.0)

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200   edg_m_9 = p.edges.getByBoundingBox(xMin=L_T/2-L_O/2, yMin=0, zMin=0.0,
201                                     xMax=L_T/2-L_O/2+t_P1, yMax=0, zMax=0.0)
202   edg_m_10 = p.edges.getByBoundingBox(xMin=L_T/2-L_O/2, yMin=t_P2, zMin=0.0,
203                                       xMax=L_T/2-L_O/2+t_P1, yMax=t_P2, zMax=0.0)
204   edg_m_11 = p.edges.getByBoundingBox(xMin=L_T/2-L_O/2, yMin=t_P2+t_A, zMin=0.0,
205                                       xMax=L_T/2-L_O/2+t_P1, yMax=t_P2+t_A, zMax=0.0)
206   edg_m_12 = p.edges.getByBoundingBox(xMin=L_T/2-L_O/2, yMin=t_P2+t_A+t_P1, zMin=0.0,
207                                       xMax=L_T/2-L_O/2+t_P1, yMax=t_P2+t_A+t_P1,
208                                       zMax=0.0)
209   edg_m_14 = p.edges.getByBoundingBox(xMin=0.0, yMin=0.0, zMin=0.0,
210                                       xMax=L_T/2-L_O/2, yMax=0.0, zMax=0.0)
211   edg_m_15 = p.edges.getByBoundingBox(xMin=0.0, yMin=t_P2, zMin=0.0,
212                                       xMax=L_T/2-L_O/2, yMax=t_P2, zMax=0.0)
213   edg_m_16 = p.edges.getByBoundingBox(xMin=L_T/2+L_O/2, yMin=0.0, zMin=0.0,
214                                       xMax=L_T, yMax=0.0, zMax=0.0)
215   edg_m_17 = p.edges.getByBoundingBox(xMin=L_T/2+L_O/2, yMin=t_P1, zMin=0.0,
216                                       xMax=L_T, yMax=t_P2, zMax=0.0)
217   edg_m_18 = p.edges.getByBoundingBox(xMin=L_T/2-L_O/2, yMin=t_P2, zMin=0.0,
218                                       xMax=L_T/2-L_O/2, yMax=t_P2+t_A, zMax=0.0)
219   edg_m_19 = p.edges.getByBoundingBox(xMin=L_T/2-L_O/2+t_P1, yMin=t_P2, zMin=0.0,
220                                       xMax=L_T/2-L_O/2+t_P1, yMax=t_P2+t_A, zMax=0.0)
221   edg_m_20 = p.edges.getByBoundingBox(xMin=L_T/2+L_O/2, yMin=t_P2, zMin=0.0,
222                                       xMax=L_T/2+L_O/2, yMax=t_P2+t_A, zMax=0.0)
223   edg_m_21 = p.edges.getByBoundingBox(xMin=0, yMin=t_P2/2, zMin=0.0,
224                                       xMax=0, yMax=t_P2, zMax=0.0)
225   edg_m_22 = p.edges.getByBoundingBox(xMin=L_T/2-L_O/2, yMin=t_P2/2, zMin=0.0,
226                                       xMax=L_T/2-L_O/2, yMax=t_P2, zMax=0.0)
227   edg_m_23 = p.edges.getByBoundingBox(xMin=L_T/2-L_O/2+t_P1, yMin=t_P2/2, zMin=0.0,
228                                       xMax=L_T/2-L_O/2+t_P1, yMax=t_P2, zMax=0.0)
229   edg_m_24 = p.edges.getByBoundingBox(xMin=L_T/2+L_O/2, yMin=t_P2/2, zMin=0.0,
230                                       xMax=L_T/2+L_O/2, yMax=t_P2, zMax=0.0)
231   edg_m_25 = p.edges.getByBoundingBox(xMin=L_T, yMin=t_P2/2, zMin=0.0,
232                                       xMax=L_T, yMax=t_P2, zMax=0.0)
233   edg_m_26 = p.edges.getByBoundingBox(xMin=0, yMin=0, zMin=0.0,
234                                       xMax=0, yMax=t_P2/2, zMax=0.0)
235   edg_m_27 = p.edges.getByBoundingBox(xMin=L_T/2-L_O/2, yMin=0, zMin=0.0,
236                                       xMax=L_T/2-L_O/2, yMax=t_P2/2, zMax=0.0)
237   edg_m_28 = p.edges.getByBoundingBox(xMin=L_T/2-L_O/2+t_P1, yMin=0, zMin=0.0,
238                                       xMax=L_T/2-L_O/2+t_P1, yMax=t_P2/2, zMax=0.0)
239   edg_m_29 = p.edges.getByBoundingBox(xMin=L_T/2+L_O/2, yMin=0, zMin=0.0,
240                                       xMax=L_T/2+L_O/2, yMax=t_P2/2, zMax=0.0)
241   edg_m_30 = p.edges.getByBoundingBox(xMin=L_T, yMin=0, zMin=0.0,
242                                       xMax=L_T, yMax=t_P2/2, zMax=0.0)
243   face_plate = p.faces.getByBoundingBox(xMin=0.0, yMin=0.0, zMin=0.0,
244                                         xMax=L_T, yMax=t_P2, zMax=0.0)
245   face_adhesive = p.faces.getByBoundingBox(xMin=L_T/2-L_O/2, yMin=t_P2, zMin=0.0,
246                                             xMax=L_T/2+L_O/2, yMax=t_P2+t_A,
247                                             zMax=0.0)
248   face_L = p.faces.getByBoundingBox(xMin=L_T/2-L_O/2, yMin=t_P2+t_A, zMin=0.0,
249                                     xMax=L_T/2+L_O/2, yMax=t_P1+t_A+t_P2+L_A, zMax=0.0)
250   face_substrate1 = p.faces.getByBoundingBox(xMin=L_T/2-L_O/2, yMin=t_P2+t_A, zMin=0.0,
251                                               xMax=L_T/2+L_O/2, yMax=t_P1+t_A+t_P2,
252                                               zMax=0.0)
253   face_substrate2 = p.faces.getByBoundingBox(xMin=L_T/2-L_O/2, yMin=0.0, zMin=0.0,
254                                               xMax=L_T/2+L_O/2, yMax=t_P2, zMax=0.0)
255   p.Set(faces=face_plate+face_L, name= 'Wood')
256   p.Set(faces=face_L, name= 'face_L')
257   p.Set(faces=face_plate, name= 'face_plate')
258   p.Set(faces=face_adhesive, name= 'Adhesive')
259   p.Set(faces=face_substrate1, name= 'Substrate1')
260   p.Set(faces=face_substrate2, name= 'Substrate2')
261   p.Set(edges=edg_m_6, name= 'ed_6')
262   p.Set(edges=edg_m_7, name= 'ed_7')
263   p.Set(edges=edg_m_8, name= 'ed_8')
264   p.Set(edges=edg_m_9, name= 'ed_9')
265   p.Set(edges=edg_m_10, name= 'ed_10')
266   p.Set(edges=edg_m_11, name= 'ed_11')
267   p.Set(edges=edg_m_12, name= 'ed_12')
268   p.Set(edges=edg_m_14, name= 'ed_14')

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266 p.Set(edges=edg_m_15, name= 'ed_15')
267 p.Set(edges=edg_m_16, name= 'ed_16')
268 p.Set(edges=edg_m_17, name= 'ed_17')
269 p.Set(edges=edg_m_18, name= 'ed_18')
270 p.Set(edges=edg_m_19, name= 'ed_19')
271 p.Set(edges=edg_m_20, name= 'ed_20')
272 p.Set(edges=edg_m_21, name= 'ed_21')
273 p.Set(edges=edg_m_22, name= 'ed_22')
274 p.Set(edges=edg_m_23, name= 'ed_23')
275 p.Set(edges=edg_m_24, name= 'ed_24')
276 p.Set(edges=edg_m_25, name= 'ed_25')
277 p.Set(edges=edg_m_26, name= 'ed_26')
278 p.Set(edges=edg_m_27, name= 'ed_27')
279 p.Set(edges=edg_m_28, name= 'ed_28')
280 p.Set(edges=edg_m_29, name= 'ed_29')
281 p.Set(edges=edg_m_30, name= 'ed_30')
282 p = mdb.models['Model-1'].parts['Part-1']
283 v = p.vertices
284 verts = v.getByBoundingBox(xMin=0, yMin=t_P2/2, zMin=0.0,
285                             xMax=0, yMax=t_P2/2, zMax=0.0)
286 p.Set(vertices=verts, name='fixed_1')
287 p = mdb.models['Model-1'].parts['Part-1']
288 v = p.vertices
289 verts = v.getByBoundingBox(xMin=L_T, yMin=t_P2/2, zMin=0.0,
290                             xMax=L_T, yMax=t_P2/2, zMax=0.0)
291 p.Set(vertices=verts, name='fixed_2')
292 p = mdb.models['Model-1'].parts['Part-1']
293 v = p.vertices
294 verts = v.getByBoundingBox(xMin=L_T/2-L_O/2+t_P1/2, yMin=t_P1+t_A+t_P2+L_A, zMin=0.0,
295                             xMax=L_T/2-L_O/2+t_P1/2, yMax=t_P1+t_A+t_P2+L_A,
296                             zMax=0.0)
296 p.Set(vertices=verts, name='load')
297 #####
298 # Section Assignment
299 p = mdb.models['Model-1'].parts['Part-1']
300 region = p.sets['Wood']
301 p = mdb.models['Model-1'].parts['Part-1']
302 p.SectionAssignment(region=region, sectionName='Wood', offset=0.0,
303 offsetType=MIDDLE_SURFACE, offsetField='',
304 thicknessAssignment=FROM_SECTION)
305 p = mdb.models['Model-1'].parts['Part-1']
306 region = p.sets['Adhesive']
307 p = mdb.models['Model-1'].parts['Part-1']
308 p.SectionAssignment(region=region, sectionName='Adhesive', offset=0.0,
309 offsetType=MIDDLE_SURFACE, offsetField='',
310 thicknessAssignment=FROM_SECTION)
311 #####
312 # Material Orientation
313 session.viewports['Viewport: 1'].partDisplay.setValues(sectionAssignments=ON,
314 engineeringFeatures=ON, mesh=OFF)
315 session.viewports['Viewport: 1'].partDisplay.meshOptions.setValues(
316 meshTechnique=OFF)
317 session.viewports['Viewport: 1'].view.setValues(nearPlane=142.118,
318 farPlane=155.523, width=78.7497, height=35.6198, viewOffsetX=0.72707,
319 viewOffsetY=-8.72021)
320 p = mdb.models['Model-1'].parts['Part-1']
321 region = p.sets['face_L']
322 orientation=None
323 mdb.models['Model-1'].parts['Part-1'].MaterialOrientation(region=region,
324 orientationType=GLOBAL, axis=AXIS_3, additionalRotationType=ROTATION_NONE,
325 localCsys=None, fieldName='', stackDirection=STACK_3)
326 p = mdb.models['Model-1'].parts['Part-1']
327 region = p.sets['face_plate']
328 orientation=None
329 mdb.models['Model-1'].parts['Part-1'].MaterialOrientation(region=region,
330 orientationType=GLOBAL, axis=AXIS_3, additionalRotationType=ROTATION_NONE,
331 localCsys=None, fieldName='', stackDirection=STACK_3)
332 #####
333 # Assembly

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334 a = mdb.models['Model-1'].rootAssembly
335 session.viewports['Viewport: 1'].setValues(displayedObject=a)
336 session.viewports['Viewport: 1'].assemblyDisplay.setValues(loads=ON, bcs=ON,
337 predefinedFields=ON, connectors=ON, optimizationTasks=OFF,
338 geometricRestrictions=OFF, stopConditions=OFF)
339 session.viewports['Viewport: 1'].assemblyDisplay.setValues(loads=OFF, bcs=OFF,
340 predefinedFields=OFF, connectors=OFF)
341 a = mdb.models['Model-1'].rootAssembly
342 a.DatumCsysByDefault(CARTESIAN)
343 p = mdb.models['Model-1'].parts['Part-1']
344 a.Instance(name='Part-1', part=p, dependent=ON)
345 #####
346 # Step
347 session.viewports['Viewport: 1'].partDisplay.setValues(sectionAssignments=OFF,
348 engineeringFeatures=OFF)
349 session.viewports['Viewport: 1'].partDisplay.geometryOptions.setValues(
350 referenceRepresentation=ON)
351 p = mdb.models['Model-1'].parts['Part-1']
352 session.viewports['Viewport: 1'].setValues(displayedObject=p)
353 a = mdb.models['Model-1'].rootAssembly
354 session.viewports['Viewport: 1'].setValues(displayedObject=a)
355 session.viewports['Viewport: 1'].assemblyDisplay.setValues(
356 adaptiveMeshConstraints=ON)
357 mdb.models['Model-1'].StaticStep(name='Step-1', previous='Initial',
358 maxNumInc=1000000, initialInc=0.01, minInc=1e-20, maxInc=0.1)
359 session.viewports['Viewport: 1'].assemblyDisplay.setValues(step='Step-1')
360 #####
361 # Boundary Conditions
362 session.viewports['Viewport: 1'].assemblyDisplay.setValues(loads=ON, bcs=ON,
363 predefinedFields=ON, connectors=ON, adaptiveMeshConstraints=OFF)
364 a = mdb.models['Model-1'].rootAssembly
365 region = a.instances['Part-1'].sets['fixed_1']
366 mdb.models['Model-1'].DisplacementBC(name='BC-1', createStepName='Step-1',
367 region=region, u1=0.0, u2=0.0, ur3=UNSET, amplitude=UNSET, fixed=OFF,
368 distributionType=UNIFORM, fieldName='', localCsys=None)
369 a = mdb.models['Model-1'].rootAssembly
370 region = a.instances['Part-1'].sets['fixed_2']
371 mdb.models['Model-1'].DisplacementBC(name='BC-2', createStepName='Step-1',
372 region=region, u1=UNSET, u2=0.0, ur3=UNSET, amplitude=UNSET, fixed=OFF,
373 distributionType=UNIFORM, fieldName='', localCsys=None)
374 a = mdb.models['Model-1'].rootAssembly
375 region = a.instances['Part-1'].sets['load']
376 mdb.models['Model-1'].ConcentratedForce(name='Load', createStepName='Step-1',
377 region=region, cf2=750.0, distributionType=UNIFORM, field='',
378 localCsys=None)
379
380 #####
381 # Mesh
382 p = mdb.models['Model-1'].parts['Part-1']
383 pickedFaces = p.sets['Wood'].faces
384 p.setMeshControls(regions=pickedFaces, elemShape=QUAD, technique=STRUCTURED)
385 elemType1 = mesh.ElemType(elemCode=CPE4R, elemLibrary=STANDARD,
386 secondOrderAccuracy=OFF, hourglassControl=DEFAULT,
387 distortionControl=DEFAULT)
388 elemType2 = mesh.ElemType(elemCode=CPE3, elemLibrary=STANDARD)
389 pickedRegions = p.sets['Wood']
390 p.setElementType(regions=pickedRegions, elemTypes=(elemType1, elemType2))
391 elemType1 = mesh.ElemType(elemCode=CPE8, elemLibrary=STANDARD)
392 elemType2 = mesh.ElemType(elemCode=CPE6M, elemLibrary=STANDARD)
393 pickedFaces = p.sets['Adhesive'].faces
394 p.setMeshControls(regions=pickedFaces, elemShape=QUAD, technique=STRUCTURED)
395 elemType1 = mesh.ElemType(elemCode=CPE8, elemLibrary=STANDARD)
396 elemType2 = mesh.ElemType(elemCode=CPE6M, elemLibrary=STANDARD)
397 pickedRegions = p.sets['Adhesive']
398 p.setElementType(regions=pickedRegions, elemTypes=(elemType1, elemType2))
399 overlap_edges = p.edges.getByBoundingBox(xMin=L_T/2-L_O/2+t_P1, yMin=-t_A/2,
400 zMin=0.0,
xMax=L_T/2+L_O/2+t_P1,
yMax=t_P1+t_A+t_P2+t_A/2, zMax=0.0)

```

```

401 overlap_edges_H = []
402 for e in p.edges:
403     if e in overlap_edges and abs(e.getSize() - L_O) < 1e-6:
404         overlap_edges_H.append(e)
405 p.seedEdgeByBias(biasMethod=DOUBLE, endEdges=overlap_edges, minSize=smallerSize,
406                 maxSize=biggerSize, constraint=FINER)
407 longer_edges = p.edges.getByBoundingBox(xMin=L_T/2-L_O/2, yMin=t_P1+t_A+t_P2,
408                                         zMin=0.0,
409                                         xMax=L_T/2-L_O/2+t_P1,
410                                         yMax=t_P1+t_A+t_P2+L_A, zMax=0.0)
411
412 longer_edges_V = []
413 for e in p.edges:
414     if e in longer_edges and abs(e.getSize() - L_A) < 1e-6:
415         longer_edges_V.append(e)
416 p.seedEdgeBySize(edges=longer_edges, size=biggerSize, deviationFactor=0.1,
417                 constraint=FINER)
418 edg = edg_m_6 + edg_m_7 + edg_m_8
419 p.seedEdgeByBias(biasMethod=SINGLE, end1Edges=edg, minSize=smallerSize,
420                 maxSize=biggerSize, constraint=FINER)
421 edg = edg_m_9 + edg_m_10 + edg_m_11 + edg_m_12
422 p.seedEdgeBySize(edges=edg, size=smallerSize, deviationFactor=0.1, constraint=FINER)
423 edg2 = edg_m_21 + edg_m_22 + edg_m_23 + edg_m_24
424 edg1 = edg_m_25
425 p.seedEdgeByBias(biasMethod=SINGLE, end2Edges=edg2, minSize=smallerSize,
426                 maxSize=biggerSize*2/3, constraint=FINER)
427 p.seedEdgeByBias(biasMethod=SINGLE, end1Edges=edg1, minSize=smallerSize,
428                 maxSize=biggerSize*2/3, constraint=FINER)
429 edg = edg_m_9 + edg_m_10 + edg_m_11 + edg_m_12
430 p.seedEdgeBySize(edges=edg, size=smallerSize, deviationFactor=0.1, constraint=FINER)
431 edg2 = edg_m_26 + edg_m_27 + edg_m_28 + edg_m_29
432 edg1 = edg_m_30
433 p.seedEdgeByBias(biasMethod=SINGLE, end2Edges=edg2, minSize=biggerSize*2/3,
434                 maxSize=biggerSize, constraint=FINER)
435 p.seedEdgeByBias(biasMethod=SINGLE, end2Edges=edg1, minSize=biggerSize*2/3,
436                 maxSize=biggerSize, constraint=FINER)
437 edg = edg_m_18 + edg_m_19 + edg_m_20
438 p.seedEdgeBySize(edges=edg, size=t_A/4, deviationFactor=0.1, constraint=FINER)
439 opposite_direction = edg_m_15 + edg_m_16
440 default_direction = edg_m_14 + edg_m_17
441 p.seedEdgeByBias(biasMethod=SINGLE, end1Edges=default_direction, minSize=smallerSize,
442                 maxSize=biggerSize, constraint=FINER)
443 p.seedEdgeByBias(biasMethod=SINGLE, end2Edges=opposite_direction,
444                 minSize=smallerSize,
445                 maxSize=biggerSize, constraint=FINER)
446 p.generateMesh()
447
448 #####
449 #Job
450 session.viewports['Viewport: 1'].view.setValues(nearPlane=17.5252,
451 farPlane=23.267, width=13.6698, height=6.20229, viewOffsetX=1.28924,
452 viewOffsetY=0.521433)
453 a1 = mdb.models['Model-1'].rootAssembly
454 a1.regenerate()
455 a = mdb.models['Model-1'].rootAssembly
456 session.viewports['Viewport: 1'].setValues(displayedObject=a)
457 session.viewports['Viewport: 1'].assemblyDisplay.setValues(mesh=OFF)
458 session.viewports['Viewport: 1'].assemblyDisplay.meshOptions.setValues(
459     meshTechnique=OFF)
460 mdb.Job(name='Job-1', model='Model-1', description='', type=ANALYSIS,
461         atTime=None, waitMinutes=0, waitHours=0, queue=None, memory=90,
462         memoryUnits=PERCENTAGE, getMemoryFromAnalysis=True,
463         explicitPrecision=SINGLE, nodalOutputPrecision=SINGLE, echoPrint=OFF,
464         modelPrint=OFF, contactPrint=OFF, historyPrint=OFF, userSubroutine='',
465         scratch='', resultsFormat=ODB, numThreadsPerMpiProcess=1,
466         multiprocessingMode=DEFAULT, numCpus=6, numDomains=6, numGPUs=6)
467 mdb.jobs['Job-1'].submit(consistencyChecking=OFF)
468 mdb.jobs['Job-1'].waitForCompletion(6000) # Wait for the job to complete
469 o3 = session.openOdb(name='Job-1.odb')
470 session.viewports['Viewport: 1'].setValues(displayedObject=o3)

```

```

467 session.viewports['Viewport: 1'].makeCurrent()
468 a = mdb.models['Model-1'].rootAssembly
469 session.viewports['Viewport: 1'].setValues(displayedObject=a)
470 session.viewports['Viewport: 1'].assemblyDisplay.setValues(loads=OFF, bcs=OFF,
471 predefinedFields=OFF, connectors=OFF)
472 session.viewports['Viewport: 1'].setValues(
473 displayedObject=session.odbs['D:/1.TESE/8. Abaqus_plugins/Job-1.odb'])
474 o3 = session.openOdb(name='D:/1.TESE/8. Abaqus_plugins/Job-1.odb')
475 session.viewports['Viewport: 1'].setValues(displayedObject=o3)
476 session.viewports['Viewport: 1'].makeCurrent()
477
478 #####
479 #Output
480 session.Path(name='Path-1', type=POINT_LIST, expression=((L_T/2-L_O/2, t_P2+t_A/2,
481 0.0), (
482 L_T/2+L_O/2, t_P2+t_A/2, 0.0)))
483 o3 = session.openOdb(name='D:/1.TESE/8. Abaqus_plugins/Job-1.odb')
484 session.viewports['Viewport: 1'].setValues(displayedObject=o3)
485 session.viewports['Viewport: 1'].makeCurrent()
486 session.viewports['Viewport: 1'].view.setValues(nearPlane=249.051,
487 farPlane=356.12, width=0.220152, height=0.0783733, viewOffsetX=-12.2646,
488 viewOffsetY=-12.3351)
489 session.viewports['Viewport: 1'].odbDisplay.setPrimaryVariable(
490 variableLabel='S', outputPosition=INTEGRATION_POINT, refinement=(COMPONENT,
491 'S22'))
492 pth = session.paths['Path-1']
493 session.viewports['Viewport: 1'].odbDisplay.setFrame(step=0, frame=-1)
494 session.viewports['Viewport: 1'].odbDisplay.setFrame(step=0, frame=-1)
495 session.XYDataFromPath(name='S22', path=pth, includeIntersections=True,
496 projectOntoMesh=False, pathStyle=PATH_POINTS, numIntervals=10,
497 projectionTolerance=0, shape=UNDEFORMED, labelType=TRUE_DISTANCE,
498 removeDuplicateXYPairs=True, includeAllElements=False)
499 session.viewports['Viewport: 1'].odbDisplay.setPrimaryVariable(
500 variableLabel='S', outputPosition=INTEGRATION_POINT, refinement=(COMPONENT,
501 'S12'))
502 pth = session.paths['Path-1']
503 session.viewports['Viewport: 1'].odbDisplay.setFrame(step=0, frame=-1)
504 session.viewports['Viewport: 1'].odbDisplay.setFrame(step=0, frame=-1)
505 session.XYDataFromPath(name='S12', path=pth, includeIntersections=True,
506 projectOntoMesh=False, pathStyle=PATH_POINTS, numIntervals=10,
507 projectionTolerance=0, shape=UNDEFORMED, labelType=TRUE_DISTANCE,
508 removeDuplicateXYPairs=True, includeAllElements=False)
509
510 #Excel export
511 sys.path.insert(9,
512
513     r'd:/SIMULIA/EstProducts/2022/win_b64/code/python2.7/lib/abaqus_plugins/excelUtil
514     ities')
515 import abq_ExcelUtilities.excelUtilities
516 abq_ExcelUtilities.excelUtilities.XYtoExcel(xyDataNames='S22', trueName='')
517 abq_ExcelUtilities.excelUtilities.XYtoExcel(xyDataNames='S12', trueName='')
518
519 #Side 11
520 leaf = dgo.LeafFromElementSets(elementSets=("PART-1.SUBSTRATE1", ))
521 session.viewports['Viewport: 1'].odbDisplay.displayGroup.replace(leaf=leaf)
522 session.viewports['Viewport: 1'].odbDisplay.setValues(viewCutNames=('X-Plane',
523 ), viewCut=ON)
524 session.viewports['Viewport: 1'].odbDisplay.viewCuts['X-Plane'].setValues(
525 showModelBelowCut=False)
526 session.viewports['Viewport: 1'].odbDisplay.viewCuts['X-Plane'].setValues(
527 showModelAboveCut=True)
528 session.viewports['Viewport: 1'].odbDisplay.viewCuts['X-Plane'].setValues(
529 showModelOnCut=False)
530 session.viewports['Viewport: 1'].odbDisplay.viewCuts['X-Plane'].setValues(
531 showModelOnCut=True)
532 session.viewports['Viewport: 1'].odbDisplay.viewCuts['X-Plane'].setValues(
533 showFreeBodyCut=True)
534 session.viewports['Viewport: 1'].odbDisplay.freeBodyOptions.setValues(
535 constantLengthArrow=ON, vectorDisplay=VECTOR_COMPONENT)

```

```

533 session.viewports['Viewport: 1'].odbDisplay.viewCuts['X-Plane'].setValues(
534     position=L_T/2-L_O/2)
535 odbName=session.viewports[session.currentViewportName].odbDisplay.name
536 session.odbData[odbName].setValues(activeFrames= (('Step-1', (-1, )), ))
537 odb = session.odbs['D:/1.TESE/8. Abaqus_plugins/Job-1.odb']
538 xyList = xyPlot.XYDataFromFreeBody(odb=odb, force=ON, moment=ON,
539     heatFlowRate=OFF, resultant=OFF, comp1=ON, comp2=ON, comp3=ON)
540 xyp = session.XYPlot('XYPlot-1')
541 chartName = xyp.charts.keys()[0]
542 chart = xyp.charts[chartName]
543 curveList = session.curveSet(xyData=xyList)
544 chart.setValues( curvesToPlot=curveList)
545 session.charts[chartName].autoColor(lines=True, symbols=True)
546 session.viewports['Viewport: 1'].setValues(displayedObject=xyp)
547 x0 = session.xyDataObjects['_X-Plane force component 1']
548 x1 = session.xyDataObjects['_X-Plane force component 2']
549 x2 = session.xyDataObjects['_X-Plane moment component 3']
550 session.writeXYReport(fileName='D:/1.TESE/8. Abaqus_plugins/InternalForces'+partName,
551     xyData=(x0, x1, x2))
552
553 #Side 12
554 odb = session.odbs['D:/1.TESE/8. Abaqus_plugins/Job-1.odb']
555 session.viewports['Viewport: 1'].setValues(displayedObject=odb)
556 session.viewports['Viewport: 1'].odbDisplay.display.setValues(plotState=(
557     UNDEFORMED, ))
558 leaf = dgo.LeafFromElementSets(elementSets= ("PART-1.SUBSTRATE1", ))
559 session.viewports['Viewport: 1'].odbDisplay.displayGroup.replace(leaf=leaf)
560 session.viewports['Viewport: 1'].odbDisplay.viewCuts['X-Plane'].setValues(
561     showModelBelowCut=True)
562 session.viewports['Viewport: 1'].odbDisplay.viewCuts['X-Plane'].setValues(
563     showModelOnCut=True)
564 session.viewports['Viewport: 1'].odbDisplay.viewCuts['X-Plane'].setValues(
565     showModelAboveCut=False)
566 session.viewports['Viewport: 1'].odbDisplay.viewCuts['X-Plane'].setValues(
567     position=L_T/2+L_O/2-0.0001)
568 odbName=session.viewports[session.currentViewportName].odbDisplay.name
569 session.odbData[odbName].setValues(activeFrames= (('Step-1', (-1, )), ))
570 odb = session.odbs['D:/1.TESE/8. Abaqus_plugins/Job-1.odb']
571 xyList = xyPlot.XYDataFromFreeBody(odb=odb, force=ON, moment=ON,
572     heatFlowRate=OFF, resultant=OFF, comp1=ON, comp2=ON, comp3=ON)
573 xyp = session.xyPlots['XYPlot-1']
574 chartName = xyp.charts.keys()[0]
575 chart = xyp.charts[chartName]
576 curveList = session.curveSet(xyData=xyList)
577 chart.setValues( curvesToPlot=curveList)
578 session.charts[chartName].autoColor(lines=True, symbols=True)
579 session.viewports['Viewport: 1'].setValues(displayedObject=xyp)
580 x0 = session.xyDataObjects['_X-Plane force component 1_1']
581 x1 = session.xyDataObjects['_X-Plane force component 2_1']
582 x2 = session.xyDataObjects['_X-Plane moment component 3_1']
583 session.writeXYReport(fileName='D:/1.TESE/8. Abaqus_plugins/InternalForces'+partName,
584     xyData=(x0, x1, x2))
585
586 #Side 21
587 odb = session.odbs['D:/1.TESE/8. Abaqus_plugins/Job-1.odb']
588 session.viewports['Viewport: 1'].setValues(displayedObject=odb)
589 session.viewports['Viewport: 1'].odbDisplay.display.setValues(plotState=(
590     UNDEFORMED, ))
591 leaf = dgo.LeafFromElementSets(elementSets= ("PART-1.SUBSTRATE2", ))
592 session.viewports['Viewport: 1'].odbDisplay.displayGroup.replace(leaf=leaf)
593 session.viewports['Viewport: 1'].odbDisplay.viewCuts['X-Plane'].setValues(
594     showModelBelowCut=False)
595 session.viewports['Viewport: 1'].odbDisplay.viewCuts['X-Plane'].setValues(
596     showModelOnCut=True)
597 session.viewports['Viewport: 1'].odbDisplay.viewCuts['X-Plane'].setValues(
598     showModelAboveCut=True)
599 session.viewports['Viewport: 1'].odbDisplay.viewCuts['X-Plane'].setValues(
600     position=L_T/2-L_O/2)
601 odbName=session.viewports[session.currentViewportName].odbDisplay.name

```

```

602 session.odbData[odbName].setValues(activeFrames= (('Step-1', (-1, )), ))
603 odb = session.odbs['D:/1.TESE/8. Abaqus_plugins/Job-1.odb']
604 xyList = xyPlot.XYDataFromFreeBody(odb=odb, force=ON, moment=ON,
605     heatFlowRate=OFF, resultant=OFF, comp1=ON, comp2=ON, comp3=ON)
606 xyp = session.xyPlots['XYPlot-1']
607 chartName = xyp.charts.keys()[0]
608 chart = xyp.charts[chartName]
609 curveList = session.curveSet(xyData=xyList)
610 chart.setValues(curvesToPlot=curveList)
611 session.charts[chartName].autoColor(lines=True, symbols=True)
612 session.viewports['Viewport: 1'].setValues(displayedObject=xyp)
613 x0 = session.xyDataObjects['_X-Plane force component 1']
614 x1 = session.xyDataObjects['_X-Plane force component 2']
615 x2 = session.xyDataObjects['_X-Plane moment component 3']
616 session.writeXYReport(fileName='D:/1.TESE/8. Abaqus_plugins/InternalForces'+partName,
617     xyData=(x0, x1, x2))
618
619 #Side 22
620 odb = session.odbs['D:/1.TESE/8. Abaqus_plugins/Job-1.odb']
621 session.viewports['Viewport: 1'].setValues(displayedObject=odb)
622 session.viewports['Viewport: 1'].odbDisplay.display.setValues(plotState=(
623     UNDEFORMED, ))
624 leaf = dgo.LeafFromElementSets(elementSets=("PART-1.SUBSTRATE2", ))
625 session.viewports['Viewport: 1'].odbDisplay.displayGroup.replace(leaf=leaf)
626 session.viewports['Viewport: 1'].odbDisplay.viewCuts['X-Plane'].setValues(
627     showModelBelowCut=True)
628 session.viewports['Viewport: 1'].odbDisplay.viewCuts['X-Plane'].setValues(
629     showModelOnCut=True)
630 session.viewports['Viewport: 1'].odbDisplay.viewCuts['X-Plane'].setValues(
631     showModelAboveCut=False)
632 session.viewports['Viewport: 1'].odbDisplay.viewCuts['X-Plane'].setValues(
633     position=L_T/2+L_O/2-0.0001)
634 odbName=session.viewports[session.currentViewportName].odbDisplay.name
635 session.odbData[odbName].setValues(activeFrames= (('Step-1', (-1, )), ))
636 odb = session.odbs['D:/1.TESE/8. Abaqus_plugins/Job-1.odb']
637 xyList = xyPlot.XYDataFromFreeBody(odb=odb, force=ON, moment=ON,
638     heatFlowRate=OFF, resultant=OFF, comp1=ON, comp2=ON, comp3=ON)
639 xyp = session.xyPlots['XYPlot-1']
640 chartName = xyp.charts.keys()[0]
641 chart = xyp.charts[chartName]
642 curveList = session.curveSet(xyData=xyList)
643 chart.setValues(curvesToPlot=curveList)
644 session.charts[chartName].autoColor(lines=True, symbols=True)
645 session.viewports['Viewport: 1'].setValues(displayedObject=xyp)
646 x0 = session.xyDataObjects['_X-Plane force component 1_1']
647 x1 = session.xyDataObjects['_X-Plane force component 2_1']
648 x2 = session.xyDataObjects['_X-Plane moment component 3_1']
649 session.writeXYReport(fileName='D:/1.TESE/8. Abaqus_plugins/InternalForces'+partName,
650     xyData=(x0, x1, x2))
651
652

```


Appendix F

***Python* implementation: CLPT**

```

1 import numpy as np
2
3 class Laminate:
4     """
5     In-plane stiffness properties of a laminate as per the Classical Laminate Theory.
6
7     Attributes:
8         `E11`: Longitudinal stiffness
9         `E22`: Transverse stiffness
10        `G12`: Shear stiffness
11        `v12`: In-plane Poisson's ratio
12        `orientations`: Orientation of each ply
13        `thicknesses` (optional): Thickness of each ply. By default, all plies are
14        considered to have the same thickness.
15
16    Methods:
17        `clt(self)`: Computes the global stiffness matrix of the laminate.
18    """
19    def __init__(self, orientations, E11, E22, G12, v12, v21=None, thicknesses=None):
20        self.orientations = orientations
21        self.E11 = E11
22        self.E22 = E22
23        self.G12 = G12
24        self.v12 = v12
25        self.v21 = v21
26
27        self.v21 = v21 if v21 != None else v12*E22/E11
28        self.thicknesses = thicknesses if thicknesses != None else [1]*len(orientations)
29
30    def clt(self):
31        E11, E22, G12 = self.E11, self.E22, self.G12
32        v12, v21 = self.v12, self.v21
33        # On-axis plane stress stiffness of UD ply
34        v = 1 - v12*v21
35        Q = [
36            [E11/v,      v21*E11/v, 0 ],
37            [v12*E22/v, E22/v,      0 ],
38            [0,          0,          G12]]
39
40        A = np.zeros([3, 3])
41        thetas = np.array(self.orientations)*np.pi/180 # rad
42
43        for theta, t in zip(thetas, self.thicknesses):
44            m, n = np.cos(theta), np.sin(theta)
45            T = [
46                [ m**2,   n**2,   m*n           ],
47                [ n**2,   m**2,   -m*n           ],
48                [ -2*m*n,  2*m*n,   m**2 - n**2 ] ]
49            A += np.transpose(T) @ Q @ T*t
50
51        C = np.linalg.inv(A/sum(self.thicknesses))
52
53        Ex = 1/C[0,0]
54        Ey = 1/C[1,1]
55        Gxy = 1/C[2,2]
56        vxy = -C[0,1]/C[0,0]
57        vyx = -C[0,1]/C[1,1]
58
59        return Ex, Ey, Gxy, vxy, vyx
60
61 # Create an instance of the Laminate class
62 laminate = Laminate(
63     orientations=(0,90), # deg
64     E11=135e3,
65     E22=10e3,
66     G12=5e3,
67     v12=0.27,
68     v21=0.27
69 )

```

Appendix G

Web implementation

G.1 General sandwich-type model

```

1  import numpy as np
2  import matplotlib.pyplot as plt
3  from numpy import real, imag
4  import sys
5  import math
6
7  P = sys.argv[1]
8  w = sys.argv[2]
9  Lt = sys.argv[3]
10 Lo = sys.argv[4]
11 t1 = sys.argv[5]
12 t2 = sys.argv[6]
13 ta = sys.argv[7]
14 Ea = sys.argv[8]
15 va = sys.argv[9]
16 materialDecision1 = sys.argv[10]
17 materialDecision2 = sys.argv[11]
18 A11_1 = sys.argv[12]
19 B11_1 = sys.argv[13]
20 D11_1 = sys.argv[14]
21 A55_1 = sys.argv[15]
22 k1 = sys.argv[16]
23 A11_2 = sys.argv[17]
24 B11_2 = sys.argv[18]
25 D11_2 = sys.argv[19]
26 A55_2 = sys.argv[20]
27 k2 = sys.argv[21]
28
29 S22 = sys.argv[22]
30 S12 = sys.argv[23]
31
32 P = float(P)
33 w = float(w)
34 Lt = float(Lt)
35 Lo = float(Lo)
36 t1 = float(t1)
37 t2 = float(t2)
38 ta = float(ta)
39 Ea = float(Ea)
40 nua = float(va)
41 A11_1 = float(A11_1)
42 A11_2 = float(A11_2)
43 B11_1 = float(B11_1)
44 B11_2 = float(B11_2)
45 D11_1 = float(D11_1)
46 D11_2 = float(D11_2)
47 A55_1 = float(A55_1)
48 A55_2 = float(A55_2)
49 k1 = float(k1)
50 k2 = float(k2)
51 S22 = float(S22)
52 S12 = float(S12)
53
54 P = P/w
55
56 T11=0
57 V11=0
58 M11=0
59
60 T12=0
61 V12=0
62 M12=0
63
64 T22 = 0
65 V22 = ((Lt-Lo)/2+(t2/2)) * (P) / (Lt)
66 M22 = -abs(V22) * ((Lt-Lo)/2)
67
68 T21 = 0
69 V21 = -((P)-abs(V22))

```

```

70 M21 = -abs(V21)*((Lt-Lo)/2)
71
72 n = 100
73
74 D1 = A11_1 * D11_1 - B11_1 ** 2
75 D2 = A11_2 * D11_2 - B11_2 ** 2
76
77 Ga = Ea / (2 * (1 + nua))
78 Eas = Ea * (1 - nua) / ((1 + nua) * (1 - 2 * nua))
79
80 a1 = (
81     Ga / ta
82     * (
83         1 / D1 * (D11_1 - (t1 + ta) * B11_1 + (0.5 * (t1 + ta)) ** 2 * A11_1)
84         + 1 / D2 * (D11_2 + (t2 + ta) * B11_2 + (0.5 * (t2 + ta)) ** 2 * A11_2)
85     )
86 )
87 a2 = Ga / ta * (1 / D1 * (B11_1 - (0.5 * (t1 + ta)) * A11_1) + 1 / D2 * (B11_2 + (0.5 *
88     (t2 + ta)) * A11_2))
89 a3 = Ga / 2 * (1 / (A55_2 * k2) - 1 / (A55_1 * k1))
89 b1 = Eas / ta * (1 / (A55_1 * k1) + 1 / (A55_2 * k2))
90 b2 = Eas / ta * (A11_1 / D1 + A11_2 / D2)
91 b3 = Eas / ta * (
92     1 / D1 * (B11_1 - (0.5 * (t1 + ta)) * A11_1) + 1 / D2 * (B11_2 + (0.5 * (t2 + ta)) *
93     A11_2)
94 )
95 A = np.array([[0, 1, 0, 0, 0, 0, 0],
96               [0, 0, 1, 0, 0, 0, 0],
97               [0, a1, 0, -a2, 0, a3, 0],
98               [0, 0, 0, 0, 1, 0, 0],
99               [0, 0, 0, 0, 0, 1, 0],
100              [0, 0, 0, 0, 0, 0, 1],
101              [0, b3, 0, -b2, 0, b1, 0]])
102
103 Eval,Evect = np.linalg.eig(A)
104 EvalDiag = np.diag(Eval)
105 evZ = Evect[:, np.isclose(EvalDiag.diagonal(), 0)].real
106
107 condition = np.logical_and(np.imag(EvalDiag) == 0, np.real(EvalDiag) != 0)
108 ewR = np.real(np.reshape(EvalDiag[np.where(condition)], (-1, 1)))
109 evR = np.real(Evect[:, np.where(condition)])
110
111 imag_EvalDiag = np.imag(EvalDiag)
112 condition = imag_EvalDiag > 0
113 ewC = np.reshape(EvalDiag[np.where(condition)], (-1, 1))
114 evC = Evect[:, np.where(condition)]
115
116 if evC.size > 0:
117     evC = evC[:, 0, :]
118     evC[:, 1] = -evC[:, 1]
119
120 sR = np.zeros(np.shape(ewR))
121 sC = np.zeros(np.shape(ewC))
122 tau0_int = np.zeros((7, len(evR[0])))
123 sR[ewR > 0], sR[ewR < 0], sC[np.real(ewC) > 0], sC[np.real(ewC) < 0] = -Lo/2, Lo/2,
124     -Lo/2, Lo/2
125
126 evR = evR[:, 0, :]
127
128 sR = np.zeros(ewR.shape)
129 sC = np.zeros(ewC.shape)
130 sR[ewR > 0] = -Lo / 2
131 sR[ewR < 0] = Lo / 2
132 sC[np.real(ewC) > 0] = -Lo / 2
133 sC[np.real(ewC) < 0] = Lo / 2
134
135 tau0_int = []

```

```

136 tau0_int.append(evZ[0, 0] * Lo)
137
138 line2 = evR[0, :] / ewR.T * (np.exp(ewR * Lo / 2) - np.exp(ewR * (-Lo / 2))).T *
139 np.exp(ewR.T * sR.T)
140 tau0_int.append(line2.flatten())
141
142 line3 = np.exp(np.real(ewC).T * sC.T) * (1 / (np.real(ewC.T) ** 2 + np.imag(ewC.T) **
143 2)) * np.exp(-np.real(ewC).T * Lo / 2) * ((np.exp(np.real(ewC).T * Lo) - 1) *
144 np.cos(np.imag(ewC.T) * Lo / 2) * (np.real(ewC.T) * np.real(evC[0, :]) + np.imag(ewC.T) *
145 np.imag(evC[0, :])) + (1 + np.exp(np.real(ewC).T * Lo)) * np.sin(np.imag(ewC.T) *
146 Lo / 2) * (np.imag(ewC.T) * np.real(evC[0, :]) - np.real(ewC.T) *
147 np.imag(evC[0, :])))
148 tau0_int.append(line3.flatten())
149
150 line4 = np.exp(np.real(ewC).T * sC.T) * 1 / (np.real(ewC.T) ** 2 + np.imag(ewC.T) ** 2)
151 * np.exp(-np.real(ewC).T * Lo / 2) * (
152 (np.exp(np.real(ewC).T * Lo) - 1) * np.cos(np.imag(ewC.T) * Lo / 2) *
153 (np.real(ewC.T) * np.imag(evC[0, :]) - np.imag(ewC.T) * np.real(evC[0, :])) +
154 (1 + np.exp(np.real(ewC).T * Lo)) * np.sin(np.imag(ewC.T) * Lo / 2) *
155 (np.imag(ewC.T) * np.imag(evC[0, :]) + np.real(ewC.T) * np.real(evC[0, :])))
156 tau0_int.append(line4.flatten())
157
158 tau0_int = np.concatenate([np.atleast_1d(arr) for arr in tau0_int])
159 dt = np.arange(-Lo / 2, Lo / 2 + Lo / (n - 1), Lo / (n - 1))
160 Dres = np.zeros((7, 7 * len(dt)))
161
162 for k in range(len(dt)):
163     tmp = np.zeros((7, 7))
164     sR = np.zeros_like(ewR)
165     sC = np.zeros_like(ewC)
166
167     indices_R_pos = ewR > 0
168     indices_R_neg = ewR < 0
169     indices_C_real_pos = np.real(ewC) > 0
170     indices_C_real_neg = np.real(ewC) < 0
171
172     sR[indices_R_pos] = dt[k] - Lo/2
173     sR[indices_R_neg] = dt[k] + Lo/2
174     sC[indices_C_real_pos] = dt[k] - Lo/2
175     sC[indices_C_real_neg] = dt[k] + Lo/2
176
177     tmp[:, 0] = evZ[:, 0]
178
179     for j in range(len(ewR)):
180         tmp[:, j+1] = evR[:, j] * np.exp(ewR[j] * sR[j])
181
182     for j in range(len(ewC)):
183         tmp[:, len(ewR) + j + 1] = np.real(
184             np.exp(np.real(ewC[j]) * sC[j])
185             * (np.real(evC[:, j]) * np.cos(np.imag(ewC[j]) * dt[k]) - np.imag(evC[:, j])
186               * np.sin(np.imag(ewC[j]) * dt[k]))
187         )
188         tmp[:, len(ewR) + len(ewC) + j + 1] = np.real(
189             np.exp(np.real(ewC[j]) * sC[j])
190             * (np.imag(evC[:, j]) * np.cos(np.imag(ewC[j]) * dt[k]) + np.real(evC[:, j])
191               * np.sin(np.imag(ewC[j]) * dt[k]))
192         )
193
194     Dres[:, k*7 : (k+1)*7] = tmp
195
196 r_r = Dres[:, -7:]
197 r_l = Dres[:, :7]
198
199 LHS = (tau0_int,
200        r_r[1, :] - Ga/2 * (1/(A55_2*k2) - 1/(A55_1*k1)) * r_r[3, :],
201        r_l[1, :] - Ga/2 * (1/(A55_2*k2) - 1/(A55_1*k1)) * r_l[3, :],
202        r_r[5, :] - Eas/ta * (1/(A55_2*k2) + 1/(A55_1*k1)) * r_r[3, :],
203        r_l[5, :] - Eas/ta * (1/(A55_2*k2) + 1/(A55_1*k1)) * r_l[3, :],

```

```

195     r_r[6, :] - Eas/ta * (1/(A55_2*k2) + 1/(A55_1*k1)) * r_r[4, :] - Eas/ta * (1/D1 *
196     (B11_1 - (0.5*(t1+ta))*A11_1) + 1/D2 * (B11_2 + (0.5*(t2+ta))*A11_2)) * r_r[0, :],
197     r_l[6, :] - Eas/ta * (1/(A55_2*k2) + 1/(A55_1*k1)) * r_l[4, :] - Eas/ta * (1/D1 *
198     (B11_1 - (0.5*(t1+ta))*A11_1) + 1/D2 * (B11_2 + (0.5*(t2+ta))*A11_2)) * r_l[0, :]
199 )
200 RHS = np.array(
201     [
202         T11 - T12,
203         Ga / ta
204         * (
205             1 / D2
206             * ((D11_2 + (0.5 * (t2 + ta)) * B11_2) * T22 - (B11_2 + (0.5 * (t2 + ta)) *
207             A11_2) * M22)
208             + 1 / D1
209             * ((-D11_1 + (0.5 * (t1 + ta)) * B11_1) * T12 - (-B11_1 + (0.5 * (t1 + ta))
210             * A11_1) * M12)
211         ),
212         Ga / ta
213         * (
214             1 / D2
215             * ((D11_2 + (0.5 * (t2 + ta)) * B11_2) * T21 - (B11_2 + (0.5 * (t2 + ta)) *
216             A11_2) * M21)
217             + 1 / D1
218             * ((-D11_1 + (0.5 * (t1 + ta)) * B11_1) * T11 - (-B11_1 + (0.5 * (t1 + ta))
219             * A11_1) * M11)
220         ),
221         Eas / ta * (1 / D1 * (-B11_1 * T12 + A11_1 * M12) - 1 / D2 * (-B11_2 * T22 +
222         A11_2 * M22)),
223         Eas / ta * (1 / D1 * (-B11_1 * T11 + A11_1 * M11) - 1 / D2 * (-B11_2 * T21 +
224         A11_2 * M21)),
225         Eas / ta * (A11_1 / D1 * V12 - A11_2 / D2 * V22),
226         Eas / ta * (A11_1 / D1 * V11 - A11_2 / D2 * V21),
227     ]
228 )
229 C = np.linalg.solve(LHS, RHS)
230
231 tau0 = np.zeros(len(dt))
232 sigma0 = np.zeros(len(dt))
233 for k in range(len(dt)):
234     stress = np.dot(Dres[:, k * 7 : (k + 1) * 7], C)
235     tau0[k] = -stress[0]
236     sigma0[k] = stress[3]
237
238 # Plot sigma0
239 plt.figure()
240 plt.plot(dt, sigma0, color=[0.8500, 0.3250, 0.0980], linewidth=1.5)
241
242 if S22 != 0:
243     plt.axhline(y=S22, color='red', linestyle='--', linewidth=1.5)
244     plt.legend(['Peel Stress', 'Maximum Failure Stress Criteria'], fontsize=10)
245 else:
246     plt.legend(['Peel Stress'], fontsize=10)
247
248 plt.xlabel('x-coordinate of overlap [mm]')
249 plt.ylabel('Stresses [MPa]')
250 plt.axis([-Lo/2, Lo/2, min(min(tau0), min(sigma0)), max(max(tau0), max(sigma0))])
251 plt.grid(True)
252 plt.savefig(r"D:\wamp64\www\My_model\L-joint\S22.png")
253 plt.close()
254
255 # Plot tau0
256 plt.figure()
257 plt.plot(dt, tau0, color=[0, 0.4470, 0.7410], linewidth=1.5)
258
259 if S12 != 0:
260     plt.axhline(y=S12, color='red', linestyle='--', linewidth=1.5)
261     plt.legend(['Shear Stress', 'Maximum Failure Stress Criteria'], fontsize=10)

```

```
256 else:
257     plt.legend(['Shear Stress'], fontsize=10)
258
259 plt.xlabel('x-coordinate of overlap [mm]')
260 plt.ylabel('Stresses [MPa]')
261 plt.axis([-Lo/2, Lo/2, min(min(tau0), min(sigma0)), max(max(tau0), max(sigma0))])
262 plt.grid(True)
263 plt.savefig(r"D:\wamp64\www\My_model\L-joint\S12.png")
264 plt.close()
```


G.2 Arbitrary laminate material properties

```

1  import numpy as np
2  import matplotlib.pyplot as plt
3  from numpy import real, imag
4  import sys
5
6  E1 = float(sys.argv[1])
7  E2 = float(sys.argv[2])
8  E3 = float(sys.argv[3])
9  v12 = float(sys.argv[4])
10 v13 = float(sys.argv[5])
11 v23 = float(sys.argv[6])
12 G12 = float(sys.argv[7])
13 G13 = float(sys.argv[8])
14 G23 = float(sys.argv[9])
15 n = int(sys.argv[10])
16 ang_str = sys.argv[11]
17 ang_list = ang_str.split()
18 ang = np.array(ang_list, dtype=float)
19 t = float(sys.argv[12])
20 ta = float(sys.argv[13])
21 seq = sys.argv[14]
22
23 S11 = 1/E1
24 S12 = -v12/E1
25 S13 = -v13/E1
26 S22 = 1/E2
27 S23 = -v23/E2
28 S33 = 1/E3
29 S44 = 1/G23
30 S55 = 1/G13
31 S66 = 1/G12
32
33 S = np.array([ [S11, S12, S13, 0, 0, 0],
34                [S12, S22, S23, 0, 0, 0],
35                [S13, S23, S33, 0, 0, 0],
36                [0, 0, 0, S44, 0, 0],
37                [0, 0, 0, 0, S55, 0],
38                [0, 0, 0, 0, 0, S66] ])
39
40 Q11 = S22 / (S11 * S22 - S12 ** 2)
41 Q12 = abs(S12) / (S11 * S22 - S12 ** 2)
42 Q22 = S11 / (S11 * S22 - S12 ** 2)
43 Q44 = G23
44 Q55 = G13
45 Q66 = G12
46
47 Q = np.array([ [Q11, Q12, 0, 0, 0, 0],
48                [Q12, Q22, 0, 0, 0, 0],
49                [0, 0, 0, 0, 0, 0],
50                [0, 0, 0, Q44, 0, 0],
51                [0, 0, 0, 0, Q55, 0],
52                [0, 0, 0, 0, 0, Q66] ])
53
54 ang = ang * np.pi / 180
55
56 if seq == "asym" and n == 2:
57     ang = np.flip(ang)
58
59 A = np.zeros((6, 6))
60 D = np.zeros((6, 6))
61 B = np.zeros((6, 6))
62
63 p = len(ang)
64
65 z = np.zeros(p + 1)
66 z[0] = -t / 2
67 z[p] = t / 2
68
69 for i in range(1, p):

```

```

70     z[i] = -t/2 + t/p*(i)
71     for i in range(p):
72         T = np.array([[np.cos(ang[i]) ** 2, np.sin(ang[i]) ** 2, 0, 0, 0, -np.sin(2 *
73             ang[i])],
74                        [np.sin(ang[i]) ** 2, np.cos(ang[i]) ** 2, 0, 0, 0, np.sin(2 *
75                        ang[i])],
76                        [0, 0, 1, 0, 0, 0],
77                        [0, 0, 0, np.cos(ang[i]), np.sin(ang[i]), 0],
78                        [0, 0, 0, -np.sin(ang[i]), np.cos(ang[i]), 0],
79                        [np.sin(ang[i]) * np.cos(ang[i]), -np.sin(ang[i]) * np.cos(ang[i]), 0,
80                        0, 0,
81                        np.cos(ang[i]) ** 2 - np.sin(ang[i]) ** 2]])
82     Q_1 = np.matmul(np.matmul(T, Q), np.transpose(T)) # Perform the matrix calculation
83     A = A + Q_1 * (z[i + 1] - z[i])
84     D = D + Q_1 * ((z[i + 1]) ** 3 - (z[i]) ** 3)
85     B = B + Q_1 * ((z[i + 1]) ** 2 - (z[i]) ** 2)
86     matrix_A = A
87     matrix_D = D / 3
88     matrix_B = B / 2
89
90     with open('D:/wamp64/www/My_model/L-joint/results.txt', 'w') as file:
91         file.write(str(matrix_A[0,0]) + '\n')
92         file.write(str(matrix_B[0,0]) + '\n')
93         file.write(str(matrix_D[0,0]) + '\n')
94         file.write(str(matrix_A[4,4]) + '\n')

```


Appendix H

Paper 2

Experimental and numerical study on eco-friendly adhesive joints for the automotive industry

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Abstract: In the mobility industry, there is an increasing recognition of the environmental consequences related to adhesive joints. As a result, efforts are being made to explore and implement environmentally friendly materials. Some examples of these materials include wooden substrates and bio-adhesives.

This study focus on adhesive L-joint, which is a commonly employed configuration in the automotive sector for creating bonded structural components with significant bending rigidity. The investigation utilized pine wooden substrates and bio-based adhesives to examine the behavior of these joints. Two distinct configurations were studied, differing solely in the fiber orientation of the wood. The research combined experimental testing and finite element modeling to analyze the strength of the joints and determine their failure mode when subjected to tensile loading conditions. The findings indicate that the configuration of the joint plays a crucial role in its overall performance, with the vertical configuration demonstrating higher strength. Additionally, a high degree of agreement was observed between the experimental and numerical analyses.

Keywords: automotive industry, adhesive bonding, L-joint, finite element analysis, pine wood, biomaterials

1 Introduction

Adhesive bonding joint has emerged in many industries, including automotive, aeronautical, aerospace, and electronics. Compared to conventional joining methods, adhesive joints offer numerous significant benefits. A great advantage is that they ensure a more uniform stress distribution along the bonded area, resulting in enhanced load transmission, good damping properties and fatigue resistance. Additionally, they also contribute to substantial reductions in weight and cost for the final structure [1].

Wood, a naturally porous material with anisotropic properties, has been extensively utilized since ancient times due to its numerous advantages. These include its renewable nature, ability to be shaped, remarkable durability, high mechanical strength, wide

availability, affordable price, excellent thermal and acoustic insulation capabilities, and appealing appearance. Furthermore, wood exhibits a lightweight composition and displays resilience in diverse environmental conditions [2,3]. In terms of mechanical strength, when considered relative to density, wood may be competitive to high-strength steel and aluminum [4].

Wood’s significance has grown substantially due to its eco-friendly characteristics and renewable nature, leading to a minimal carbon footprint [3]. Its utilization contributes to the development of lightweight vehicles, resulting in a noteworthy reduction of approximately 15-20% in CO_2 emissions without compromising performance [4]. To ensure the long-term health and diversity of forests, forestry and harvesting practices prioritize the adoption of fast-growing timber species. However, these species often exhibit low density and weak mechanical properties, which limits their application in advanced engineering structures and other industries [5–7]. Therefore, their mechanical properties must be carefully evaluated.

Wood has played a significant role in the automotive industry, often used alongside steel [8]. However, as industrialization progressed and the global steel industry flourished, metals gradually replaced wood in vehicle bodies. Despite this transition, there are still companies that incorporate natural materials, such as Morgan Motor Company and Ford. Morgan Motor Company has utilized an ash wood-aluminum chassis, while Ford actively researches and develops natural fiber-based compounds, particularly for interior panels [9]. Academic research has also shown interest in exploring the possibility of using wood materials. For instance, the University of Applied Sciences Merseburg and Technische Universität Chemnitz collaborated to create a lightweight concept car named NIOS, weighing only 160 kg. Various components of this urban car, powered by fuel cells, were crafted using wood [10]. Another noteworthy project by Technische Universität Chemnitz, in collaboration with Fritz Beck KG, Volkswagen AG, and Sachs Engineering GmbH, is the HAMMER study. This study investigated the utilization of metal and fiber-reinforced laminated wood in automotive applications, and it concluded that introducing wood into the automotive industry is technically viable [11–13]. Furthermore, FH Aachen University of Applied Sciences conducted the ‘HoKa’ study, which focused on examining the static and dynamic mechanical properties of wood. The study confirmed the outstanding specific mechanical properties and technical potential of wood [14].

Numerical simulations have become an indispensable component of product design and development in various industries, promoting substantial cost and time savings while improving product quality and production processes [15]. Despite the potential benefits they offer, engineering techniques utilizing numerical simulations are not widely employed throughout the entire forest wood chain. Nowadays, the mathematical description of wood material involves three main approaches: homogeneous volume elements that simplify wood properties and structure [16–18], simulating the wood cell wall and structure [19],

and employing multiscale modeling techniques [20–23]. Numerical methods have found extensive application in civil engineering, particularly in studying deformations under static loads, material failure, and crack propagation using finite element methods (FEM) [24–26]. Nevertheless, material failure of wood has not been extensively studied using numerical methods [27–34]. In the case of adhesive joints, the use of cohesive zone elements provides a reasonable balance between calculation time and prediction accuracy.

Numerical models require prior knowledge of material properties. In the context of adhesively joints with bio-materials, the material properties of the substrates and adhesives must be carefully evaluated. Therefore, an experimental investigation must be conducted to characterize these materials. Once the properties are successfully obtained, the numerical model can simulate the overall behavior of the joint. Additional experimental studies within the overall joint must be performed, for validation of the model. Moura et al. [2], conducted a study that presented various practical applications with pine wood. These applications encompassed the repair of beams under tensile loading, repair of beams under bending loading, reinforcement of wood structures, steel-wood-steel connections, and wood-wood joints. The researchers utilized typical wood connections and structural details to emphasize the importance of incorporating non-linear fracture mechanics concepts, particularly cohesive zone modeling, in this field. The study achieved a comprehensive qualitative and quantitative representation of the mechanical behavior and failure modes observed in these applications. Hence, its work establishes the proposed procedures as fundamental tools for the efficient design of wood structures. In a separate study, Corte-real et al. [35] in 2022, investigated single lap joints (SLJs) using different types of wooden substrates. These substrates included natural wood, wood/cork composites, and densified pine wood bonded with a novel polyurethane-based bio-adhesive. It was concluded that the densification process was successful, although it presented an additional challenge due to the resulting surface. Furthermore, increasing the overlap had a positive impact on the energy absorption of the joint, and the addition of cork resulted in a more consistent failure mode and higher strain to failure.

Pine wood is of particular interest due to its economic viability and abundance availability. While previous studies have explored the use of pine wood, there is limited research specifically focusing on its application in L-joint configurations, which are of great importance to the automotive industry. Extensive research is required to enhance the understanding of the behavior of adhesively bonded wooden L-joints. This study aims to contribute to this field by investigating the potential of pine wood for L-joint configurations. Therefore, this study specifically examines adhesively bonded L-joints under tension loading conditions. To achieve this objective, an experimental study was conducted, considering two different configurations. Additionally, a numerical study was performed to complement the experimental findings. The main goal of this work is to validate the design process method for bonded vehicle structures, with a strong emphasis

on the reliability and reproducibility of the method.

The experimental work of this study is described in detail in Section 2. The numerical analysis is exposed in Section 3. Finally, Section 4 presents the conclusions of the study.

2 Experimental testing

In this section, a careful description of the materials used, including their properties, as well as the geometry and dimensions of the joint are provided. Additionally, the testing conditions of all experiments are revealed in order to ensure transparency and reproducibility of the results. Finally, the section presents the conclusive findings obtained from the experiments.

2.1 Materials

The materials chosen for the investigation of the L-joint configuration comprised bio-based materials. Wood materials were selected for the substrate, while a bio-based adhesive was chosen for bonding purposes.

2.1.1 Wood substrate

For the experimental procedures conducted, maritime pine timber (*Pinus pinaster* Ait.) was utilized. Pine wood is a softwood commonly employed in practical structural applications due to its cost-effectiveness, strength, and long-lasting nature.

The elastic and strength properties of *Pinus pinaster* Ait. were determined through experimental testing conducted by Moura and Dourado et al. [2]. Table 1 provides the nominal elastic values of Young’s modulus (E_i), Poisson’s ratios (ν_i), and shear modulus (G_i) for *Pinus pinaster* Ait. Table 2 presents the nominal strength properties in different directions, denoted as L (longitudinal), R (radial), and T (transverse). Moreover, Table 3 displays the cohesive parameters for this particular wood species.

Table 1: Nominal elastic properties of *Pinus pinaster* Ait. [2].

E_L [GPa]	E_R [GPa]	E_T [GPa]	ν_{LT}	ν_{LR}	ν_{TR}	G_{LR} [GPa]	G_{LT} [GPa]	G_{TR} [GPa]
12.0	1.91	1.01	0.51	0.47	0.31	1.12	1.04	0.29

Table 2: Nominal strength properties of *Pinus pinaster* Ait. [2].

σ_{uL} [MPa]	σ_{uR} [MPa]	σ_{uT} [MPa]	σ_{uLR} [MPa]	σ_{uLT} [MPa]	σ_{uRT} [MPa]
97.5	7.90	4.20	16.0	16.0	4.50

Table 3: Cohesive parameters of *Pinus pinaster* Ait. in the RL fracture system [36–38].

G_{Ic} [N/mm]	G_{IIc} [N/mm]	$\sigma_{1,I}$ [MPa]	$\sigma_{1,II}$ [MPa]
0.264	0.94	5.34	9.27

2.1.2 Adhesive

The adhesive employed in this study is a polyurethane-based bio-adhesive, with 70% of its composition derived from natural sources. It is essential to mention that this bio-adhesive is currently in the prototype stage and has not been commercially released. The adhesive was developed by Professor João Bordado from Instituto Superior Técnico and was specifically selected due to its excellent compatibility and adhesion to wood substrates. Unlike conventional adhesives, this bio-adhesive demonstrates distinctive characteristics as it necessitates porous wood substrates for effective curing. Once cured, it forms a very thin adhesive layer with zero thickness.

The experimental characterization of the adhesive properties was conducted by Corte-Real et al. [35]. In terms of mechanical characterization, testing using a modified thick adherend shear test (TAST) specimen revealed a minimum shear strength value of 12.4 MPa. The author mentions tests utilizing modified double-cantilever beam (DCB) and end-notched flexure (ENF) specimens provided G_{Ic} and G_{IIc} values of 0.16 N/mm. In an additional study involving butt-joints, the tensile strength was measured to be 16.5 MPa.

2.2 Joint manufacturing

In the context of joint manufacturing, a critical parameter to consider is the orientation of wood rings. Depending on the substrate, different types of orientations were taken into consideration. Specifically, for the bottom substrate, the preferred orientation of the rings is illustrated in Figure 1. However, it is important to acknowledge that achieving perfect symmetry of the wood rings in relation to a centered vertical axis was not always attainable. Regarding the upper substrate, two distinct configurations were

studied and are presented in Figure 2. Consequently, two distinct L-joint configurations were manufactured.

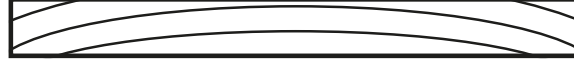


Figure 1: Illustration of the cross-section dimensions of the bottom substrate.

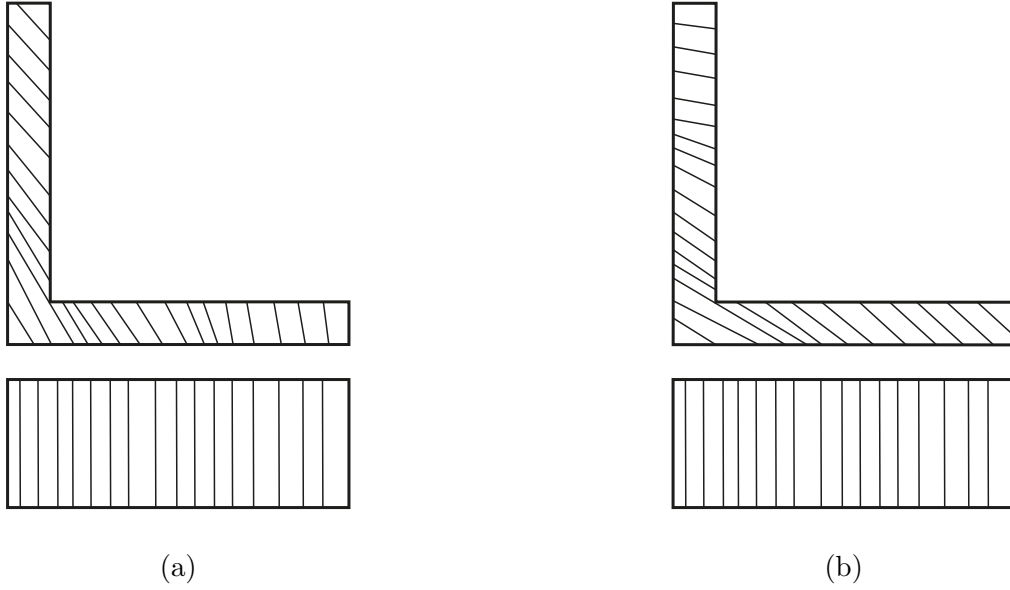


Figure 2: Illustration of the cross-section dimensions of the upper substrate: (a) vertical orientation, (b) inclined orientation.

The manufacturing process involved preparing the substrates by cleaning them. A thin layer of adhesive was then applied to both substrates, in the overlap region. The substrates were aligned and pressed together. To ensure a strong bond, pressure was applied to the overlap region using a plate and clamps. The assembled substrates were then placed in an oven at a temperature of 100°C for 8 hours, allowing the adhesive to cure. Finally, the joints were retrieved and any excess adhesive on the edges was cleaned when necessary. The final assembly and the overall dimensions of the joints are presented in Figure 3.

2.3 Testing conditions

The L-joints manufactured for this work were tested via the application of a tensile load on the upper substrate and were conducted by the quasi-static test machine. The traction tests were performed using a universal testing machine *INSTRON 3367*® with

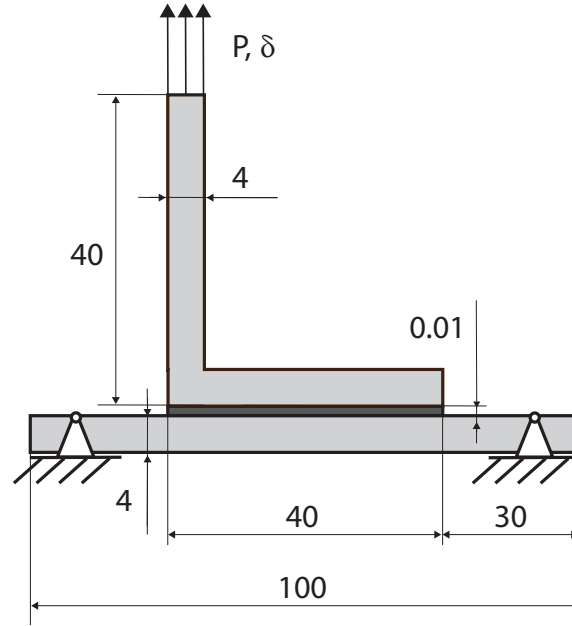


Figure 3: Dimensions of the L-joint tested.

a load cell capacity of 30 kN and both the load and displacement were recorded by the machine at a constant rate of 1 mm/min.

Figure 4 shows the set-up utilized to conduct the testing procedures. It should be noted that at least five specimens were tested for each condition. The tests were conducted in laboratory conditions of temperature and humidity.



Figure 4: Experimental setup for testing the L-joints.

2.4 Experimental results

A comprehensive strength analysis was conducted for pine wood, considering the two types of geometries presented earlier. The observed differences among the studied cases were examined in detail. Furthermore, this section provides a study of the failure paths, including an analysis of the damage initiation behavior and the type of failure observed.

2.4.1 Strength analysis

A strength analysis was conducted considering both configurations. The $P - \delta$ curves were derived from experimental procedures, directly extracted from the testing machine. It is important to highlight that, in order to eliminate the influence of machine component measurements, the displacement values for the experimental curves were adjusted using a correction factor of 0.6. Figure 5 and Figure 6 present the results for the vertical and inclined configurations, respectively.

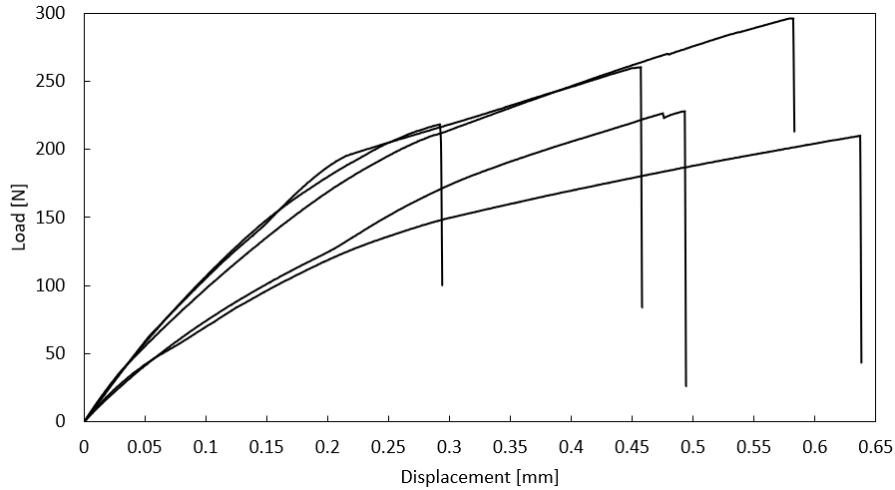


Figure 5: P - δ curve for pine wood with vertical configuration.

In the case of pine wood with a vertical fiber orientation of the upper substrate (Figure 5), the joint experienced failure at a load of 242.9 ± 32.4 N, with a corresponding displacement of 0.49 ± 0.12 mm. On the other hand, for the alternative geometry (Figure 6), the joint sustained a load of 183.2 ± 23.6 N before failure, resulting in a displacement of 0.38 ± 0.03 mm.

A comparison between these two cases reveals that the first case exhibited approximately 25% higher strength than the second case. Additionally, in terms of displacement, it demonstrated around 22% more capacity for displacement-to-failure.

Furthermore, the behavior observed in the two cases differs significantly. In the vertical configuration, there appears to be a change in the material's behavior, as indicated by a distinct change in the slope of the curve. Initially, the material exhibits an elastic

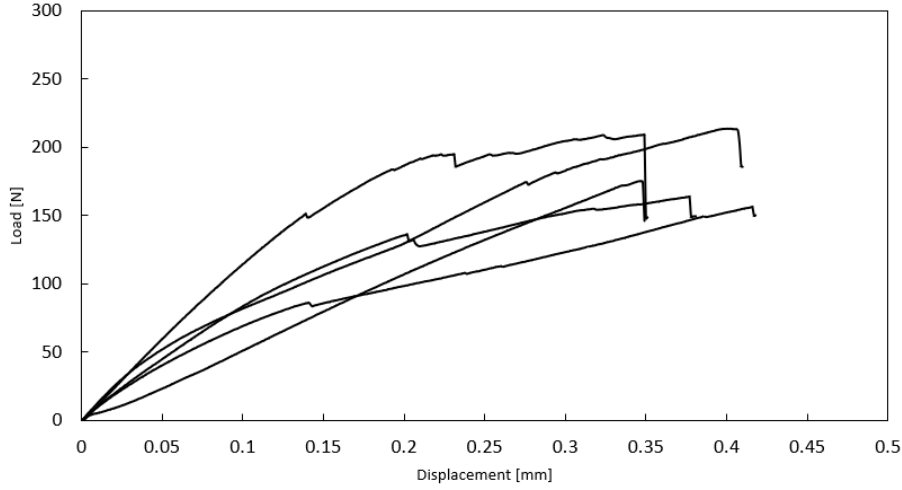


Figure 6: P- δ curve for pine wood with inclined configuration.

behavior, but then transitions to a plastic behavior, allowing for more displacement before failure. In the inclined configuration, there are small fractures in the wood that were not observed in the vertical configuration. These small fractures suggest the initiation of localized damage in the inclined configuration, contributing to its distinctive behavior. Therefore, it can be concluded that the two configurations not only exhibit differences in strength and displacement capacity but also display distinct behaviors during testing.

Moreover, it is crucial to emphasize that the vertical configuration demonstrates superior characteristics including higher energy absorption, greater strength, and increased displacement capability prior to failure. The presence of vertical fiber orientations in the overlap region results in the loading being applied along the strongest direction, thereby enhancing the stiffness and strength of the joint. The vertical alignment of the fibers maximizes their load-bearing capacity, leading to improved overall performance of the joint.

2.4.2 Failure mode

Following the testing phase, a detailed analysis was conducted to identify the type of failure modes observed for both cases. Two types of behavior were registered. In both cases a visibly non-reflective surface in the overlap region strongly suggests the occurrence of delamination. In the case of the vertical configuration, the delamination process involved a greater number of fibers being pulled, accompanied by a fracture occurring within the wood at an very oblique angle (Figure 7(a)). Conversely, for the inclined configuration, the plane of failure exhibited a more vertical orientation and was sudden, resulting in a flatter failure surface (Figure 7(b)).

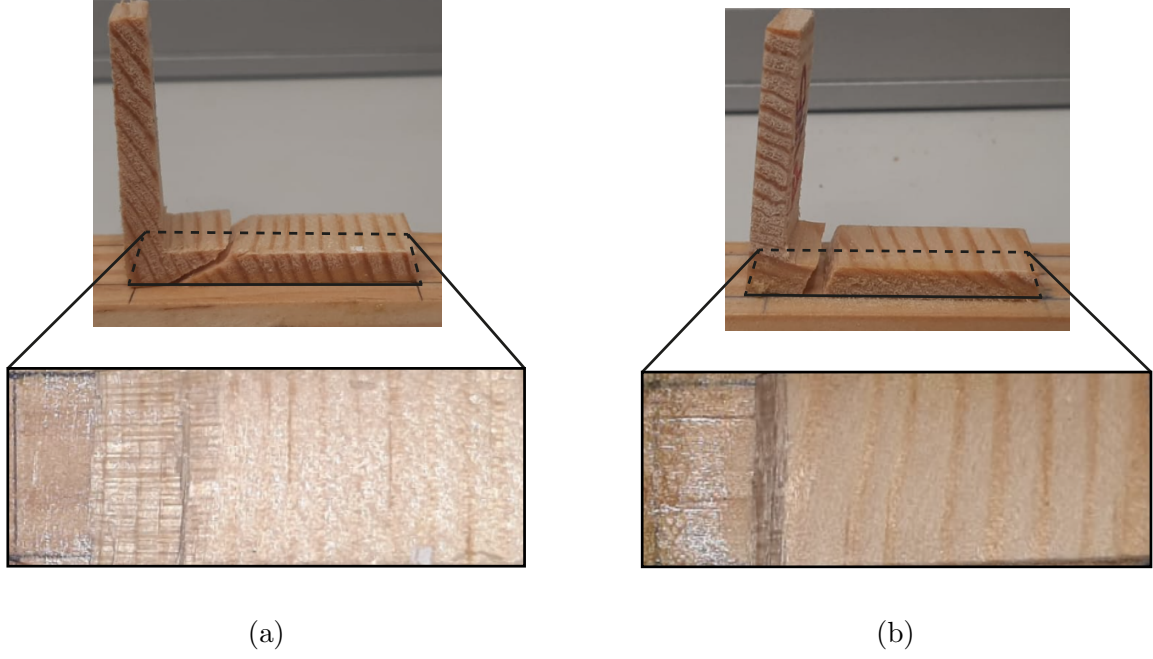


Figure 7: Failure mode observation for: (a) vertical configuration, (b) inclined configuration.

3 Numerical analysis

This section provides a comprehensive overview of the numerical methodology employed as well as the corresponding results obtained.

3.1 Numerical details

The numerical details section provides a concise description of the general considerations, followed by an explanation of the boundary conditions considered and the mesh applied in the analysis.

3.1.1 General considerations

A finite element model was developed to simulate the L-joints in the attempt to predict the real behavior of the structure. The *Abaqus 2022*[®] software was used to carry out the analysis.

Two static numerical models were simulated using pine wood with bio-adhesive for the two configurations previously mentioned (Figure 2). To model the adhesive layer, cohesive elements were utilized, and their behavior was defined by a triangular traction-separation law. The properties of this law, including strength and fracture properties of the novel bio-adhesive, were determined by Corte-Real et al. [35]. For the wood substrates, a homogenous anisotropic layer was employed with the corresponding elastic properties

obtained from Table 1 [2]. Additionally, a layer of cohesive elements was positioned at a 0.15 mm distance of the interface of both upper and bottom substrates to allow for possible delamination of the wood. The triangular cohesive law governing these elements was defined based on the strength and fracture properties of natural pine wood, as determined through experimental analysis by Moura and Dourado [2]. A visual representation of the model can be seen in Figure 8.

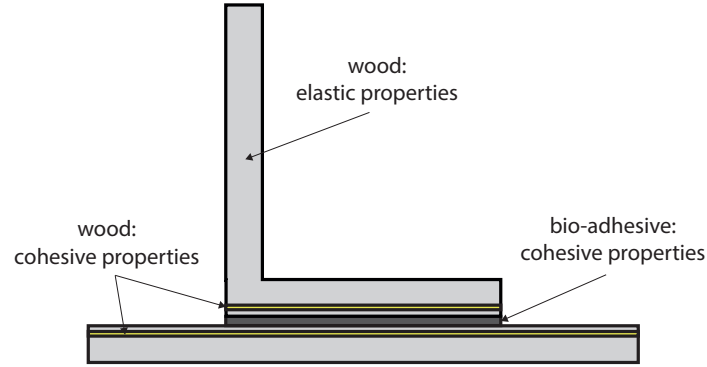


Figure 8: Cohesive elements placed in L-joint numerically tested.

3.1.2 Boundary conditions

To ensure an accurate simulation of the experimental testing conditions, the boundary conditions were carefully considered. Consequently, the horizontal and vertical directions of the two upper edges of the bottom substrate were fixed. The upper edge of the upper substrate was subjected to a vertical upward displacement while being constrained from any horizontal movement. This configuration is visually displayed in Figure 9.

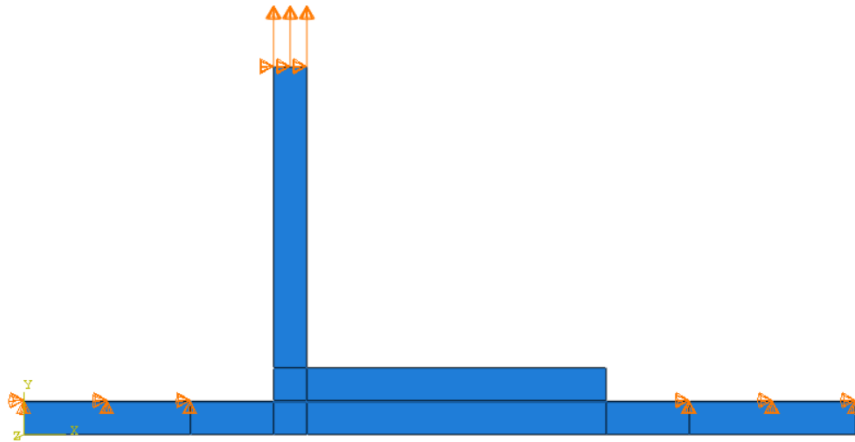


Figure 9: Boundary conditions applied to the L-joint.

3.1.3 Mesh

The simulations employed a high mesh density for both cases, ensuring convergence of the results. The mesh configuration comprised 11740 four-node bilinear plane strain quadrilateral elements (CPE4) and 700 four-node two-dimensional cohesive elements (COH2D4). In order to accurately capture bending phenomena, the thickness of the substrates was discretized using more than 20 elements. Furthermore, a mesh refinement strategy was implemented specifically in the right-side region of the adhesive to account for the anticipated occurrence of high localized stress effects in that particular area. Figure 10 presents the mesh configuration.

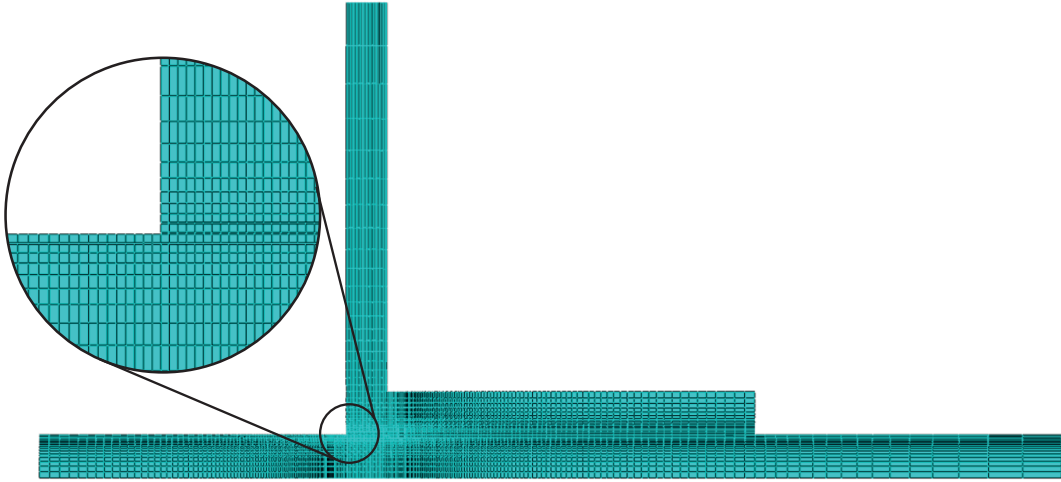


Figure 10: Mesh applied to the L-joint.

3.2 Numerical results

A strength analysis was performed on the two previously mentioned configurations, and the obtained results were compared with the experimental data. Additionally, a detailed study of the failure paths was conducted, which included analyzing the behavior of damage initiation and determining the type of failure observed.

3.2.1 Strength analysis

A strength analysis was performed using the FE software for both configurations, and the resulting $P - \delta$ curves were directly extracted and compared to the experimental data. Figure 11 compares the numerical and experimental results for the vertical configuration, while Figure 12 presents the comparison for the inclined configuration.

In the case of the vertical configuration, the numerical results indicate that the load failure occurred at 261.2 N with a displacement of 0.28 mm. As mentioned earlier, the

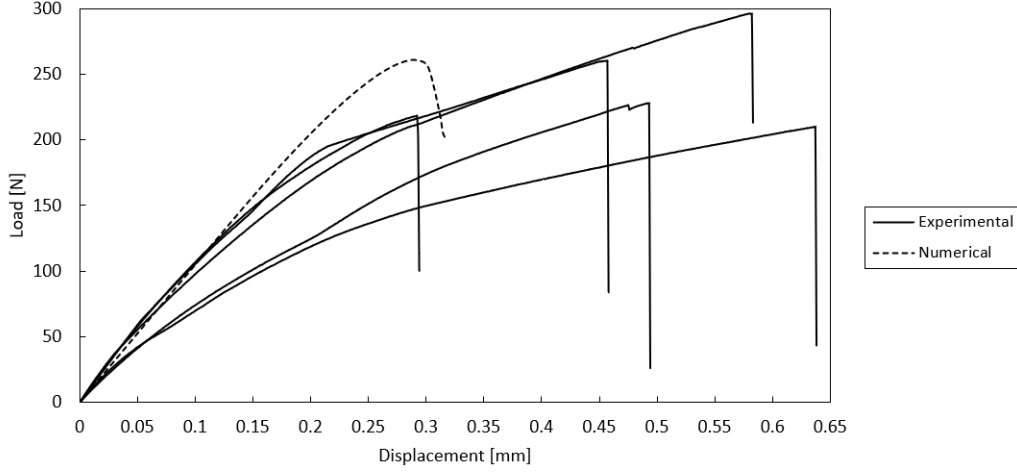


Figure 11: Comparison between experimental and numerical $P - \delta$ curves for the vertical configuration of the L-joint.

experimental failure load was 242.9 ± 32.4 N, with a corresponding displacement of 0.49 ± 0.12 mm. This suggests that the numerical model accurately predicts the failure load. However, in terms of displacement, the numerical results fall slightly short compared to the experimental values. Up to approximately 0.2 mm, the slope between the numerical and experimental results is very accurate. However, beyond that point, there is a noticeable deviation in behavior between the numerical and experimental results. One possible explanation for this discrepancy is that the experimental data indicates signs of plastic behavior, allowing for more displacement before failure. Unfortunately, this phenomenon was not taken into account in the numerical simulation due to lack of information to simulate that behavior. Despite the limitation of considering only the elastic behavior of wood, the numerical results still provide reasonably accurate predictions.

In the case of the inclined configuration, the numerical analysis indicated a load failure of 260.8 N with a displacement of 0.29 mm. The experimental testing resulted in a load until failure of 183.2 ± 23.6 N, with a corresponding displacement of 0.38 ± 0.03 mm. This discrepancy can be attributed to the presence of microcracks in the experimental testing that are not accounted for in the FEA. Consequently, the FEA tends to predict higher failure loads compared to the experimental results. Despite this difference, the overall findings constitute good strength predictions of the joint. Comparing the $P - \delta$ curves, it is evident that the numerical results closely match the experimental data, indicating a good agreement between the two approaches.

When comparing both cases, it is evident that the numerical results are very similar for both configurations in terms of failure load, displacement, and the shape of the $P - \delta$ curve. However, in reality, the two configurations exhibit distinct behaviors as described earlier in Section 2.4.1.

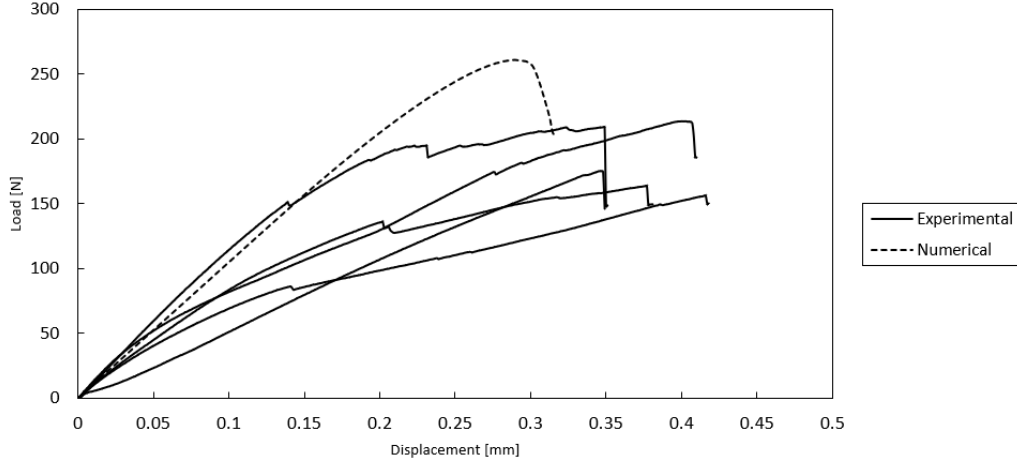


Figure 12: Comparison between experimental and numerical $P - \delta$ curves for the inclined configuration of the L-joint.

3.2.2 Failure mode

Regarding the numerical results, the predominant failure mode observed was delamination. To gain a deeper understanding of this phenomenon, the state variable SDEG was utilized to monitor the evolution of damage. When the SDEG analysis reveals a value of zero, it indicates the absence of any damage. As damage progresses, the SDEG value increases linearly, and when it reaches one, the cohesive element fails and disappears. Figure 13 illustrates the occurrence of delamination in the upper cohesive layer of the wood, with horizontal propagation. However, it was noticed that the numerical model did not precisely replicate the observed failure mode from the experiments. Upon thorough analysis, it was determined that the maximum principal stress criteria should be considered. By definition, the maximum principal stress failure predictor (MPSFP) states that if a component is subjected to a multiaxial stress system, the fracture will occur when the maximum principal stress exceeds the local strength. It was verified that, at a given displacement, the maximum principal stress value exceeded the local strength for the LR plane (16 MPa). This indicates that the crack propagated in that specific region, perpendicular to the direction of the maximum principal stress, as shown in Figure 14. Therefore, the failure path in the numerical analysis precisely matches the failure observed experimentally.

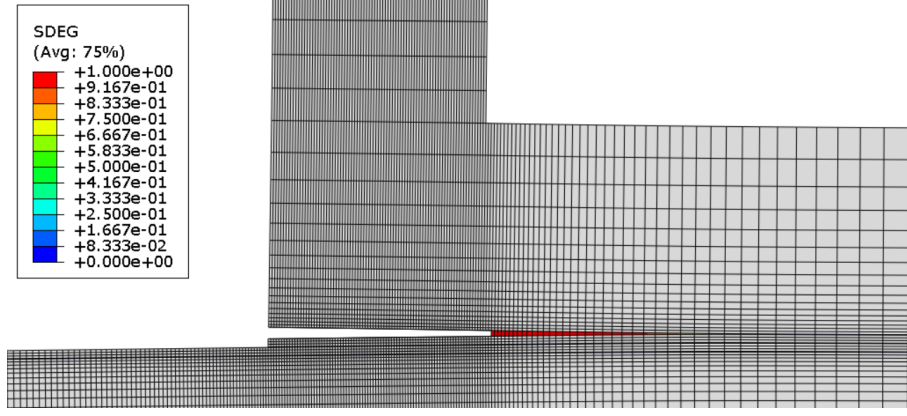


Figure 13: Damage analysis (SDEG) for L-joint.

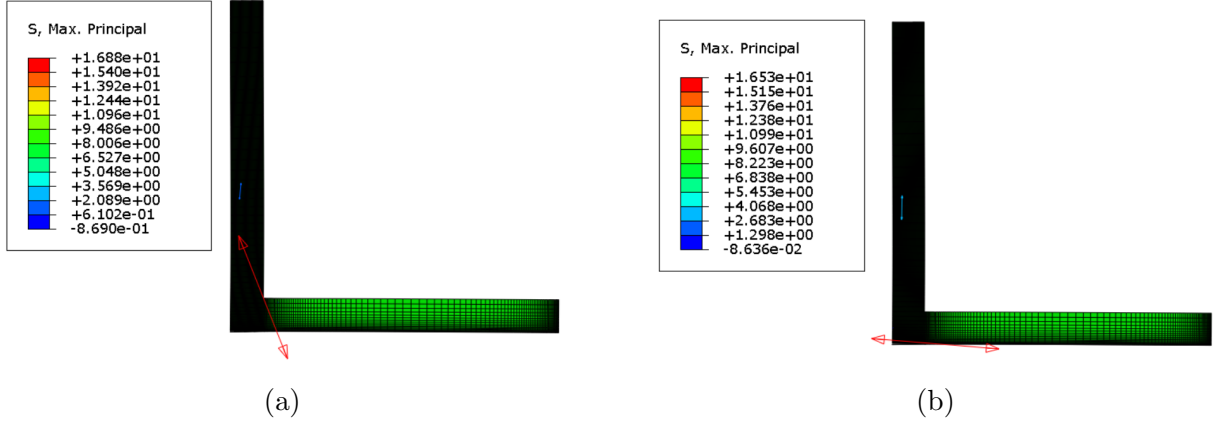


Figure 14: Maximum principal stress distribution in the FEA: (a) vertical configuration, (b) inclined configuration.

4 Conclusions

This study focused on investigating L-joints for the automotive industry by incorporating biomaterials, specifically pine wooden substrates with a novel polyurethane-based bio-adhesive. Two different configurations using pine wood were analyzed through strength analysis and failure mode. The study involved experimental and numerical analyses, which were compared to each other.

The findings emphasized the crucial role of fiber orientation in determining the overall behavior of the joint. The orientation of the fibers was found to impact the strength capability of the structure and influence the failure mode. Notably, the presence of fibers oriented along the direction of loading improved the strength and stiffness of the joint. Therefore, special attention should be given to the way that the wood is cutted and then implemented in the structure because of their anisotropic nature, and consequently impact in the overall behavior.

The joint testing results demonstrated that pine adhesively bonded L-joints could withstand a maximum load of 242.9 ± 32.4 N before failure, accompanied by a corresponding displacement of 0.49 ± 0.12 mm. It should be emphasized that these tests were conducted at a reduced scale. Nevertheless, these findings demonstrate promising performance for pine L-joints.

The numerical simulations performed in this study provide valuable insights and should be integrated into the design phase. They enable predictions of structural behavior and assist in optimizing joint configurations. However, it is essential to note that the experimental results are of utmost importance in evaluating the real performance of the structure. Experimental testing provides validation and helps uncover the intricacies and nuances that may not be captured by numerical simulations alone.

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