

Faculdade de Engenharia da Universidade do Porto

Fatigue in Steel Railway Wagons: Advanced Analysis for Improved Performance and Durability

Dinis Andrade Moraes Barata dos Santos



Master in Mechanical Engineering
Machine Design

Supervisor: Dr. Cláudio Horas

Second Supervisor: Prof. Dr. Abílio de Jesus

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Abstract

Rail freight transportation has gained renewed importance in recent years due to the increasing need for sustainable mobility solutions and the environmental targets established by the European Union. However, the reliability and structural integrity of freight wagons remain key challenges, particularly regarding the fatigue behavior of welded joints. These connections are highly susceptible to cyclic loading and frequently govern the service life of railway vehicles. Consequently, the development of more reliable fatigue assessment methodologies has become essential for improving maintenance planning and extending structural life.

This dissertation was developed within the framework of the Smart Wagons project, whose objective is to strengthen the Portuguese railway industry through the development of advanced freight wagons. The main objective of this work is to evaluate the fatigue behavior of welded joints in railway freight wagons using the equivalent structural stress approach and the Master S-N curve methodology. To achieve this objective, a comprehensive review of the fatigue phenomenon in railway freight vehicles was presented, together with the current state of the art regarding stress-based fatigue assessment methods. Then, a numerical fatigue reassessment was performed for different S355 welded details. This assessment was carried out using a customized post-processing tool integrated into the finite element workflow to extract the equivalent structural stresses and derive a Master S-N curve representative of railway welded connections. After that, the developed methodology was applied to a critical welded joint of the SmartWagons freight wagon using a global-local multiscale finite element approach, allowing the comparison between the proposed Master S-N curve and existing reference curves.

The results of the reassessment of the experimental data available in the literature enabled the derivation of an engineering fatigue resistance design curve, whose application to the SmartWagons case study resulted in a compliant fatigue assessment. These findings confirm the potential of the Master S-N curve approach as a reliable tool for the fatigue assessment of freight wagons and contribute to improving structural design and maintenance strategies for railway rolling stock. Finally, the dissertation outlines several directions for future work. The proposed developments are expected to contribute to more accurate fatigue life predictions and to the design of safer, lighter and more sustainable railway freight vehicles.

Keywords: Freight Wagons, Fatigue Assessment, Master S-N curve, Global-local analysis, Structural Integrity

Resumo

O transporte ferroviário de mercadorias assumiu uma importância crescente nos últimos anos devido à necessidade de soluções de mobilidade mais sustentáveis e às metas ambientais estabelecidas pela União Europeia. Porém, a fiabilidade e a integridade estrutural do material circulante continuam a constituir desafios importantes, particularmente no comportamento à fadiga das ligações soldadas. Estas ligações são suscetíveis a carregamentos cíclicos e, frequentemente, determinam a vida útil das estruturas ferroviárias. Assim, o desenvolvimento de metodologias de avaliação da fadiga mais fiáveis tornou-se essencial para melhorar a gestão da manutenção e prolongar a vida útil das estruturas.

O principal objetivo deste trabalho é a avaliação do comportamento à fadiga de ligações soldadas em vagões de mercadorias, utilizando a metodologia das tensões estruturais equivalentes e da curva Mestre S-N. Para tal, foi realizada uma revisão do fenómeno da fadiga em veículos ferroviários, bem como do estado da arte relativo às metodologias de avaliação da fadiga baseadas em tensões. Posteriormente, foi efetuada uma reavaliação numérica da fadiga para diferentes detalhes soldados em aço S355, recorrendo a uma ferramenta de pós-processamento desenvolvida para integração com modelos de elementos finitos, permitindo a extração das tensões estruturais equivalentes e a obtenção de uma curva Mestre S-N representativa de ligações soldadas ferroviárias. Por fim, esta metodologia foi aplicada a uma ligação crítica de um vagão de mercadorias, enquadrado no projeto Smart Wagons, recorrendo a uma abordagem numérica multiescala global-local, permitindo a comparação entre a curva Mestre S-N proposta e curvas de referência existentes.

A reavaliação dos dados experimentais disponíveis na literatura permitiu obter uma curva de projeto para a resistência à fadiga, cuja utilização no caso de estudo Smart Wagons conduziu a uma avaliação da fadiga em conformidade com os critérios estabelecidos. Os resultados obtidos demonstram o potencial da metodologia baseada na curva Mestre S-N como uma ferramenta fiável para a avaliação da fadiga de vagões ferroviários de mercadorias, que contribui para a melhoria das estratégias de dimensionamento estrutural e de manutenção do material circulante. Finalmente, são apresentadas hipóteses para investigação e desenvolvimento futuros, esperando-se que estes contribuam para previsões mais precisas da vida à fadiga e para o desenvolvimento de vagões de mercadorias mais seguros, leves e sustentáveis.

Palavras-chave: Vagão de Mercadorias, Análise de Fadiga, Curva Mestre S-N, Análise Global-Local, Integridade Estrutural

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A todos os que me acompanharam, obrigado.

Dinis Santos

*“Strive for perfection in everything you do.
Take the best that exists and make it better.
When it does not exist, design it.”*

Sir Henry Royce

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Chapter 1

Introduction

1.1 Background

The invention of the steam engine, one of the core elements of the industrial revolution of the XIX century, led to an exponential growth in goods transportation, expanding significantly the range of activity and reducing the transportation time. This change of paradigm, contrasting with the less effective local and slow system in use until then, was an important step towards the inter-connectivity between the main cities across the globe. As a product of this revolution, the transportation by rail was possible and determinant in every country, offering a more flexible and faster solution for goods transportation, shaping their economies and growth.

In recent decades, however, the European landscape for freight transportation has shifted significantly from the rails to the roads, as seen in the percentage of total cargo that each represents in Figures 1.1 and 1.2.

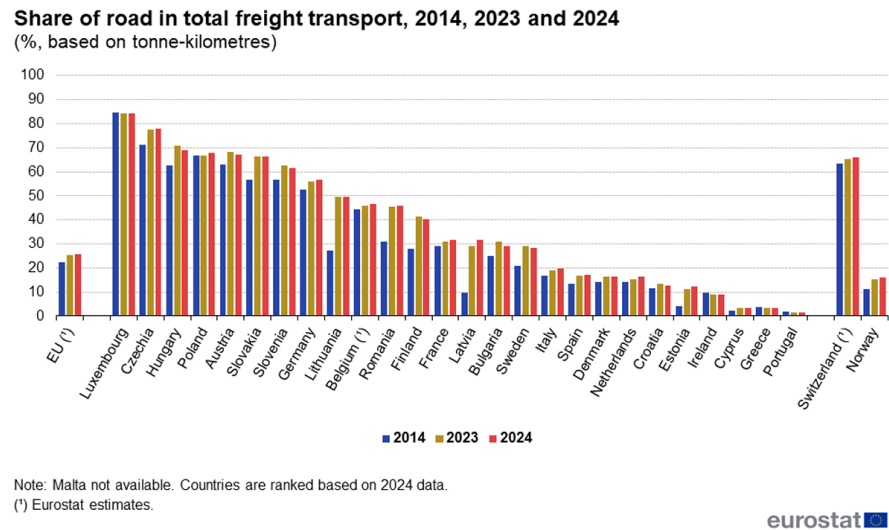


Figure 1.1: Share of road in total freight transport, 2014, 2023 and 2024 [1].

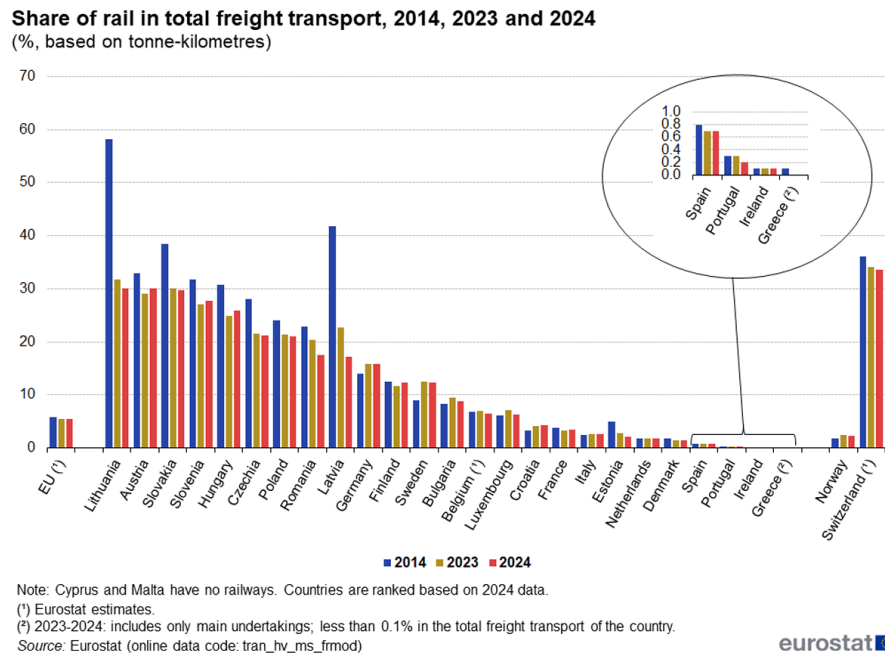


Figure 1.2: Share of rail in total freight transport, 2014, 2023 and 2024 [1].

The percentages shown in the previous graphs also take into account the inland waterways, maritime and air contributions for each country's cargo transportation. Comparing Portugal's evolution with the rest of the European Union countries, the main trend is the decrease in percentage of rail cargo. However, taking bigger countries as examples, such as Germany and Sweden, the recent efforts show an increase in the percentage of cargo transported by rail. Nevertheless, this increase was not achieved by the reduction of road freight transport, as this percentage also increased in the same time period. Therefore, even in these countries, the road freight transport trend is still observed.

The recent trend of road transportation is allied to the higher flexibility and big investments in motorways, presenting a lower cost upfront and being the main option for many companies. However, with this increasing trend there are other aspects that come into discussion, such as the environmental concerns. In fact, in terms of emissions, rail transportation is only responsible for 0.4% of total emissions related to transports, as seen in Figure 1.3.

Furthermore, rail cargo transportation consumes six times less energy and produces eight times less air pollution, resulting in twelve times fewer external costs to society when compared to road transport [2]. In accordance with this scenario and aiming to reduce the emissions associated with transportation, the European Union took big measures, such as the Green Deal [3] and the Fit for 55 package [4]. With transport accounting for 25% of all greenhouse emissions in the EU and 30% of those emissions related to freight transportation [5], the shift from road to rail and inland waterways was proposed. In addition, this

objective was emphasized by the European Union’s White Paper on Transport in 2019, integrating targets such as achieving climate neutrality by 2050 (Green Deal) and shifting 75% of the current road cargo transportation to rail and inland waterways.

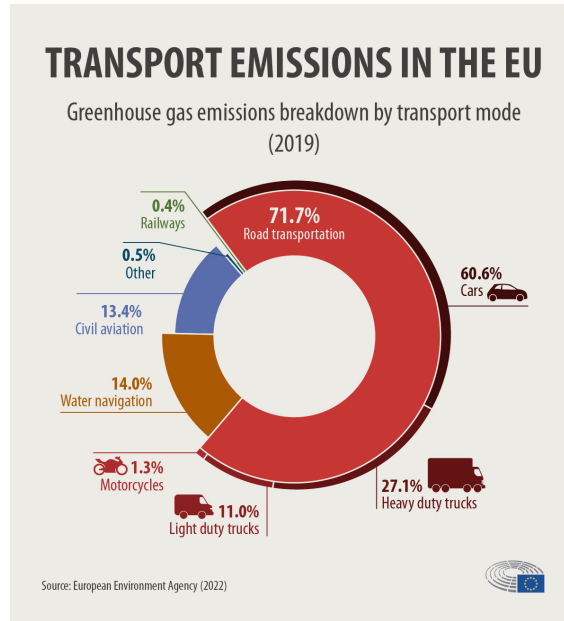


Figure 1.3: Transport emissions in the EU [6].

In order to ensure the baseline objectives already presented, there are many subjects to be addressed, such as interoperability inside the EU. In fact, before the proposition of the environmental and sustainability packages from the EU, as a solution for this subject the Trans-European Transport Network (TEN-T) was created, as presented in Figure 1.4. The TEN-T is vital for developing a coherent, efficient, multimodal, and high-quality high speed transport infrastructure across the EU.

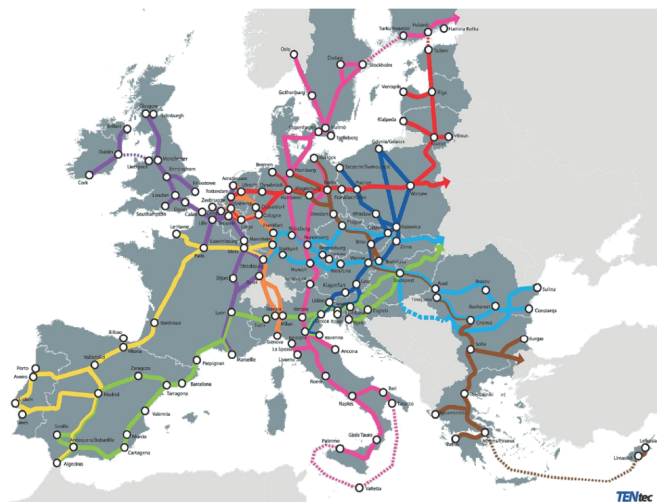


Figure 1.4: Trans-European Transport Network [7].

The TEN-T network comprises railways, inland waterways, short sea shipping routes, and roads, connecting urban nodes, maritime and inland ports, airports, and terminals. It facilitates the efficient movement of both passengers and freight, while also ensuring accessibility to employment opportunities and essential services and promoting trade and economic growth. Eliminating physical gaps, bottlenecks and missing links, the TEN-T strengthens the economic, social and territorial cohesion of the European Union and contributes to the development of integrated and seamless cross-border transport systems.

In addition, it places strong emphasis on reducing the environmental impact of transport and enhancing network safety and resilience, in alignment with the EU's broader sustainability objectives. Achieving this transition requires substantial investment in both infrastructure and vehicle capabilities. In this context, the upgrading of existing rail networks, the construction of new inter-modal terminals and the development of resilient freight wagons with longer operational life are key priorities.

Over the years, numerous projects have therefore fostered innovation and supported the modal shift from road to rail in Europe's freight transport sector. One example is the Shift2Rail project [8], which focused on doing research and innovation and commercial products, increasing the integration of innovative and advanced technologies into new solutions for the railways.

According to the last decades European trend, Portugal also shows bigger investments in road projects in comparison with rail related projects. However, in recent years, in alignment with the EU's recommendations, Portugal is also investing in freight rail, with projects like *Corredor Internacional Sul*, linking Sines to the Spanish border and improving the interoperability between the rest of Europe and the Atlantic [9], and the Autoeuropa and Setúbal port connection via rail that reduces emissions by 80% compared to the road transport for the same distance [10]. In addition, MEDWAY, the main private freight operator in the Iberian peninsula [11], is also building more inter-modal transfers across the country [12].

This thesis is related to the agenda "SMART WAGONSS – Development of Production Capacity in Portugal of Smart Wagons for Freight", nr. C644940527-00000048, investment project nr.27, financed by the Recovery and Resilience Plan (PRR) and by European Union- NextGenerationEU. The main objectives are to revive the railroad industry in Portugal and produce and maintain interoperable rail vehicles for freight transport in the European area, equipped with digital technology to optimize maintenance and operation, increasing their availability and improving the economic and environmental sustainability of the railroad.

One of the key factors of freight wagons is the design solutions used for their manufacturing, one example being the welds and respective material. In fact, the normally used materials for these wagons are regular structural steels, like the S355 steel, with great overall performance. Nevertheless, there are also high-strength steels being used in heavy duty structures and the possibility of using these steels in rail rolling stock is still theoretical. In addition, freight wagons are not only subject to high static loads, there are cyclic loads due to the loaded and unloaded states of the wagon and the operational conditions that induce another set of loading conditions. Furthermore, the dynamic loading induces fatigue in the freight wagons, with the welded connections being one of the relevant aspects to consider in fatigue analysis. In fact, cyclic loading over longer periods of operation may lead to implications in the maintenance of the railway vehicles, threatening the sustainability strategy and the available fleet.

Taking this into account and considering the effects of the welded joints on the rolling stock, the initial appeal for a high strength steel, that offers high static loading resistance, may be put in question. Nevertheless, the use of these steels would possibly lead to a reduction of material in the rolling stock, reducing the overall weight and reducing the power needed for the transport of the same amount of cargo. Through the optimization of this process, other goals, such as diminished energy consumption, reduced CO_2 emissions and increased load capacity, can be achieved.

In terms of fatigue, there are several ways to compute the theoretical life expected in a component, with different stress variants for each method, like nominal and hot-spot stresses. The nominal stress approach is the most adopted method as it does not take into consideration the local characteristics of the load transfer mechanism for the nominal stress itself. These characteristics will appear on the nominal S-N curves and therefore will be considered, however, the carried out approximations may lead to an excessively conservative result. One of the emerging analyses is the equivalent structural stress analysis applied with the Master S-N curve, proposed by Dong et al. [13]. This method leads to the compression of the dispersion of data points present on the nominal S-N curve around a mean curve, the Master S-N curve. In fact, this approach is already addressed in ASME [14] for pressure vessels and piping, and is employed in different applications, such as welded connections in bridges [15] and wind turbines [16]. This is an interesting method as the model considers the presence of the welded connection and, allied with mesh insensitivity in the analysis, allows the assessment of several different connections on one single curve, with the equivalent structural stress as the input. Thus, the method allies the representation of the load transfer mechanism with efficient models, compatible for different geometries.

The design constraints can be theoretically tackled using software equipped with the

finite element method (FEM), which proves to be a time and cost effective solution. However, the simulations have a limit of applicability and all important design issues that spark doubt should be addressed with experimental testing.

Ultimately, this approach contributes to the overarching goal of enhancing the sustainability of the railway industry. With these questions in mind, the next section presents the objectives set for this thesis.

1.2 Objectives

The main objective of this dissertation is to evaluate the fatigue behavior of welded joint on freight wagons using the Master S-N curve approach. This shall be achieved by creating and analyzing various types of welded joint models on a finite element method software, whose data are available in the literature, followed by a fatigue reassessment. The intermediate milestones of the present work are:

- Present the historical context of railway freight vehicles and the fatigue phenomenon to which they are subjected;
- Review the state of the art regarding stress-based approaches for fatigue life assessment;
- Fatigue reassessment of different welded joint details of S355 steel using finite element method models;
- Validate the results according to the present standards and recommendations;
- Derive a new Master S-N curve for railway welded connections and apply it to the SmartWagons case study.

1.3 Outline

In this section, a brief description of the chapters of this dissertation is presented. Overall, this work is divided into 5 chapters. Following the introductory chapter, this dissertation is structured as follows:

Chapter 2 - State of the art in Railway Freight Vehicles

This chapter provides an in-depth framework of the fatigue phenomenon in freight wagons. First of all, the global considerations in freight vehicle design are presented, followed by the fatigue occurrence during their operation. Subsequently, the chapter provides an overview of the methods used to evaluate fatigue in welded connections, such

as stress-based approaches and the comparison between the nominal S-N curve and the Master S-N curve. Lastly, the chapter focuses on fatigue in two kinds of steel and offers a brief review of the SmartWagons project, as well as its connection to the dissertation.

Chapter 3 - Normative Framework

In this chapter, the different standards applicable in the assessment of fatigue in railway freight vehicles are addressed in detail. The general and most relevant standards are presented, followed by other complementary applicable recommendations, such as the International Institute of Welding recommendations [17] and the DVS 1612 [18] standard, relevant for the evaluation of the welded joints on railway vehicles.

Chapter 4 - Master S-N curve-based approach

This chapter presents the results of the finite element analysis and is divided into 3 main sections. Firstly, the workflow for the fatigue assessment is presented. After that, the methodology used in each of the case studies is addressed, with the respective models and considerations taken into account, followed by the respective stress plots. The fatigue resistance values of each case study of the welded connections are from the literature, with the respective fatigue reassessment of the data. Lastly, the results are compared with the different normative approaches and validated according to the Master S-N curve.

Chapter 5 - Analysis of a Full-Scale Freight Railway Wagon

This chapter presents the application of the proposed methodology to a full-scale railway freight wagon, developed in the SmartWagons project. A finite element model is used to evaluate the fatigue behavior of the critical welded joints and the results obtained with the Master S-N curve-based approach are compared with those from the conventional nominal stress method through the evaluation of the corresponding utilization factors.

Chapter 6 - Conclusions and Future Work

In this final chapter, the thesis concludes with a synthesis of the key findings and pertinent observations. Recommendations for future work and ongoing research directions in the realm of this thesis topic are also addressed.

Chapter 2

State of the Art in Railway Freight Vehicles

Freight wagons are unpowered railway vehicles designed for various types of cargo, from raw materials to processed goods and waste disposals. In order to accommodate this wide range of requirements, each specialized freight vehicle has a different design solutions, resulting in various possible analyses. Nevertheless, each type of wagon needs to verify different aspects of its manufacturing depending on the operational focus. For example, a tank wagon will need to verify pressure resistance and fluid-related loading responses during operation but a container wagon will not have such a type of loading present.

Subsequently, regardless of the specialization of the rolling stock, there are always some specifications that remain relevant, such as axle load and maximum operating speed. In fact, these two specifications are inversely correlated, as an increase in the axle load, which represents the weight transmitted by each wheel set to the track, will reflect in a decrease of the design speed. Other factors should be taken into account when addressing this specifications, such as the track category. For example, the rail track category in Portugal is D which imposes a maximum of 22.5 tons per axle, according to EN 15528 standard [19].

In addition, considering the rest of the important aspects of the freight wagon, such as the bogie specifications and wagon design, all according to the European Railway Agency (ERA) through the Technical Specifications for Interoperability (TSI), it is possible to verify the static structural safety of the vehicles. For freight wagons, the particular specification is the TSI and, like all the other ERA and UIC (International Union of Railways) railway specifications, can be applied alongside other relevant standards as long as the latter does not contradict the specialized obligatory railway standards. The WAG-TSI states that the static safety verifications should contain yield stress, traction resistance,

instability and stiffness deformation criteria. It is also stated that the safety factor should be higher when dealing only with numerical calculations and no experimental validation.

Complementing the static analysis design, there are other relevant verifications for railway freight vehicles, such as fatigue assessment. The relevance of this phenomenon in engineering design and, in this case, in railway rolling stock is high due to factors such as the age of the current rolling stock, the increase in transported payloads and the recent aim to increase rail freight transportation. Consequently, this assessment needs a normative baseline and framework similar to the static analysis. Therefore, this phenomenon and respective analysis are addressed in the continuation of this chapter and the respective normative framework is presented in chapter 3.

2.1 Fatigue phenomenon in freight wagons

Similarly to every component that experiences significant cyclic loading, fatigue in railway freight vehicles is an important factor to be taken into account. In fact, the fatigue phenomenon is a process of cumulative structural degradation under cyclic or fluctuating stresses, leading to different types of behavior of the subjected components. This behavior is mainly influenced by the presence of material defects or possible crack-initiation sites, such as details with higher stress concentration factors. Therefore, this phenomenon is characterized by acting stresses significantly lower than the material's yield strength.

Another important consideration is the accumulated-damage. Depending on the fatigue life verification method, different considerations need to be made. In fact, the traditional infinite life verification will only verify the stress range value to be admissible for a certain number of completed consecutive cycles. However, considering the accumulated damage approach, there is the need to account for loading spectra, cycle counting and the Palmgren-Miner's law [20] for the accumulated damage, as presented in the EN 12663-2 [21] and EN 17149-3 [22] standards. Whereas both standards refer to accumulated damage, the former only states the possibility of this analysis, while the latter addresses in detail the methodology from the resistance side through the definition of S-N curves.

This approach is closer to real life loading scenarios and material behavior, proving to be of interest in the railway engineering panorama. In addition, it allows the determination of the remaining life on the components in years as well as evaluating different scenarios in terms of damage accumulation, as the infinite life assessment only returns a binary compliance response.

Subsequently, fatigue life will strongly depend on the accumulated damage and the crack propagation behavior. In freight wagons, fatigue life will be primarily influenced

by factors such as the load spectrum, oscillations and stress magnitude. In fact, the dynamic load spectrum can be subdivided into three types of loads: vertical, longitudinal and transversal.

Firstly, regarding vertical loads, these vehicles are subjected to a continuous spectrum of accelerations derived of the loading and unloading cycles and operating conditions. For design purposes, a dynamic vertical component of $\pm 30\%$ relative to the static load is often adopted to simulate track-induced vibrations [23]. Notably, Figures 2.1 and 2.2 shows an example of these failures, followed by a schematic representation of the failure mechanism in Figure 2.3.



Figure 2.1: Side view of failure in wagon SQDY 00060G [24].



Figure 2.2: Failed wagon at Gulgong [24].

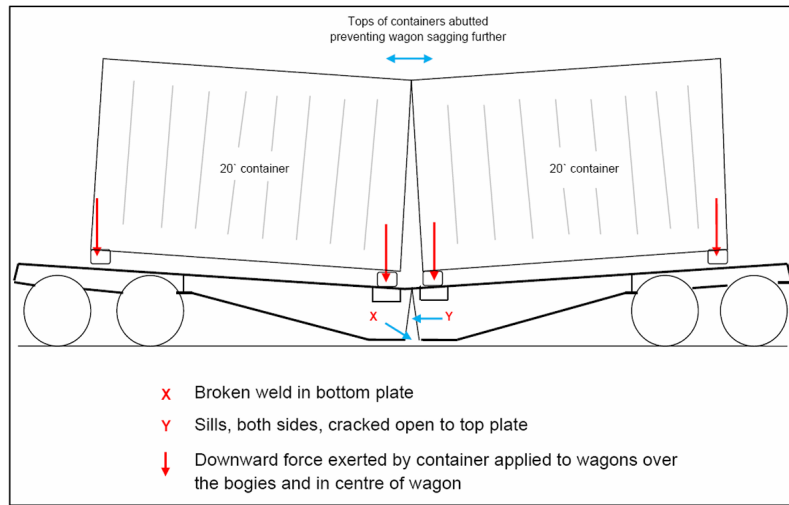


Figure 2.3: Representation of failure mechanism [24].

Similarly, transversal loads also take an important part in fatigue of the rolling stock. These forces act in a direction parallel to the rails and are usually present at the beginning of motion, braking, and logistical maneuvers in rail yards. Moreover, the dynamics of traction and compression in the longitudinal direction will induce more fatigue cycles on the framework, contributing to the crack growth rate da/dN , through the Paris law regime [25].

Lastly, the stress concentrations that appear on the freight wagons are usually the result of geometric discontinuities, such as container attachments, changes in the central section and welded connections. This geometries create potential fatigue initiation sites due to the magnification of the nominal stresses, often exceeding the material's fatigue threshold.

Overall, the combination of all these factors will lead to the stress spectrum that the freight wagon is subjected. Unlike laboratory experimentation, this spectrum is not composed of simple constant cycles and, therefore, the fatigue phenomenon is often dominated by variable amplitude loading (VAL).

2.2 Fatigue in metallic materials

The structural degradation of metallic materials under cyclic loading is a multi-scale phenomenon governed by localized plastic deformation, even when the global structural response remains strictly within the elastic regime. In polycrystalline metals, the application of repetitive stress cycles will trigger the movement and accumulation of dislocations

along preferred crystallographic slip planes. This process alters the material's local topography and drives the evolution of structural damage from the atomic level to macroscopic failure [26].

The fatigue degradation will then start at the microscopic level on every surface irregularity that was created by the slip due to the severe local stress concentration felt at those cracks [27]. The propagation of the crack will pass through different stages, with the first one being shear driven. The second stage will represent the moment where the crack has propagated and it encounters grain boundaries. In this stage, the propagation will be driven by alternating tensile stresses perpendicular to the crack plane. Therefore, the stage's duration will be highly dependent on the material's microstructure configuration and loading cycle. To evidence this dependence, Figure 2.4 presents the difference in fatigue life between different types of steels, when compared using notch stresses.

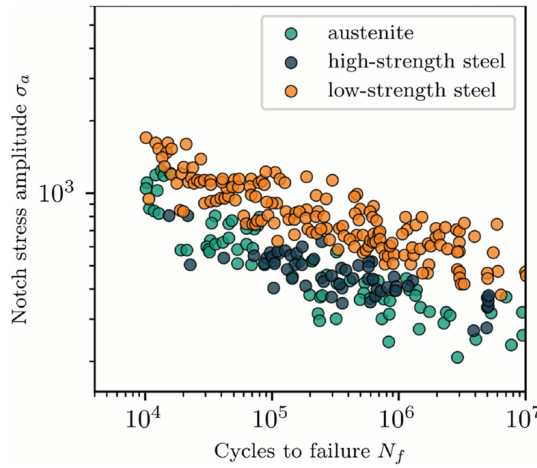


Figure 2.4: Influence of material strength on the endurable notch stresses with $r_{ref}=0.05$ mm [28].

Accordingly, two separate parts are presented: one addressing currently used structural steels and the other focusing on high-strength steels. This distinction is necessary in view of the aspects previously highlighted, as well as the recent studies on the use of high-strength steels in welded structures subjected to significant fatigue loading.

2.2.1 Current structural steels

Current used structural steels, such as the S355, represent the historical composition of heavy structural engineering and rolling stock manufacturing. These steels are characterized by a ferritic-perlitic microstructure and offer an optimized balance between ductility, fracture toughness and weldability. Due to the application in question, the choice for this material is understandable, even with the lower yield strength when compared to high

strength steels.

In terms of fatigue, these steels exhibit a relatively low notch sensitivity index, meaning that the reduction in fatigue strength due to geometric discontinuities is mitigated by localized plastic deformation, resulting in a higher notch tolerance [29]. This is possible given the well-documented cases of this material within the standardized FAT classes defined in EN 1993-1-9 [30].

Furthermore, current structural steels demonstrate high crack growth resistance and a high fracture toughness threshold [31]. In the stable crack growth regime, governed by the Paris law [25], the propagation of the crack will be relatively slow, providing a larger safety margin for defect detection during the maintenance operations, preventing the occurrence of a sudden brittle fracture.

Finally, these steels generally present hardening behavior under variable amplitude loading, elevating the local material resistance near the stress concentration zones. This phenomenon occurs due to the micro-plastification around the critical zones, delaying the propagation even more.

2.2.2 High-strength steels

In modern structural optimization, high-strength steels are often employed in order to reduce the mass and optimize the load-bearing capacity of the structure. The main characteristic of these alloys is the significantly higher yield strength when compared to the structural steels commonly used. For example, the S690 is a high strength low alloy steel and has a yield strength of approximately 690 MPa, nearly double when compared with the 355 MPa yield strength of S355. However, the advantages gained in static loading scenarios are diminished when the material is under fatigue loading, as the scaling of its characteristics under fatigue are not linear with the static strength gains.

In fact, the ultimate tensile strength of unnotched parent materials typically allows the fatigue limit to scale proportionally, a fundamental behavior that is heavily documented in baseline structural steel characterizations [32]. Nevertheless, the fatigue strength of the welded connections using S690 is practically identical to the fatigue strength of S355 [33]. According to the standard nominal S-N curves, a specific welded detail category applies universally regardless of the steel grade used. This happens due to the fact that the weld fatigue life will be dominated by the crack propagation, therefore, the static strength advantage may be eliminated on these welded connections [34].

Furthermore, the heat effects on the HAZ will create softer material properties, altering significantly the martensitic-bainitic structure of the S690. Considering that the fatigue cracks will preferentially nucleate on the heat-affected zone, this softer material will create an ideal scenario for crack initiation and rapid propagation.

Regarding the residual stresses from the welding procedure, high-strength steels can withstand much higher magnitudes of residual stresses. However, the higher resistance does not invalidate the effects of the residual stresses on the effective stress ratio during the loading cycle. Therefore, the presence of these stresses may still help to keep crack surfaces open during the dynamic loading spectrum and, due to the lower crack propagation resistance, accelerate the fatigue crack growth.

2.3 Fatigue in welded connections

Welded connections are one of the most vulnerable regions in a rail freight vehicle as they represent metallurgical discontinuity and a geometric notch. In addition, there are other factors that come into play, such as welding related defects and the heat-affected zone (HAZ). In fact, even the best welds are prone to welding defects and unwanted inclusions that will affect significantly the weld's resistance to both fatigue and static loading. Regarding the HAZ, the welding process acts as a heat treatment to the parent material, resulting in a more fragile region of the material, contributing even more to the discontinuity near the welded connection.

In addition, welded connections may also introduce residual tensile stresses, leading to an effective variation of the value of the stress ratio (R) in these joints. Therefore, the material can be subjected to a dynamic loading spectrum where the unloading phase of the cycle may still represent a tensile stress, keeping the crack tips open. However, this scenario will vary depending on the position and solicitation of the welded connections.

In Figure 2.5 a schematic representation of a welded connection is presented. The fatigue phenomenon usually initiates at the weld toe or root. In order to assess the integrity of each connection, the traditional standard visual inspections are often insufficient to detect sub-surface or other types of cracks. Therefore, aiming to maintain the observation of each connection and respective crack transition from stable to unstable, it is important to use more powerful methods. Accommodating to the aging rolling stock fleet and prototype testing, non-destructive methods are advisable, such as magnetic particle inspection (MPI) and ultrasonic testing (UT).

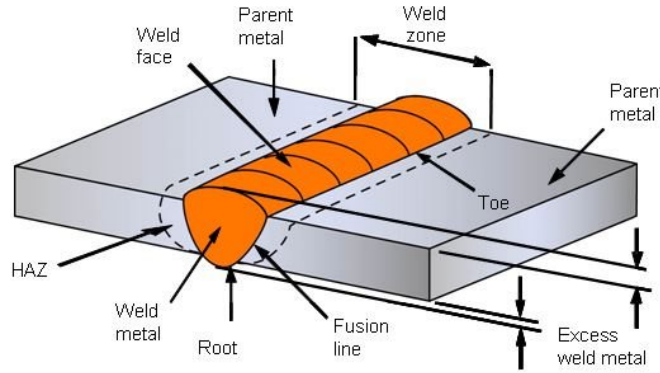


Figure 2.5: Butt weld diagram with relevant nomenclature [35].

In the presence of fatigue, welded connections may fail with a significant loss of structural integrity, such as the critical connections in a freight wagon. In fact, the underframe of the center sill is a critical structural component, since a failure in these joints can indicate a catastrophic collapse of the wagon [36]. Moreover, once a crack transverses the center sill, the structural redundancy is lost and the load will be transferred to the side sills, where the crack will rapidly propagate and the frame will experience buckling [37].

On the design phase, there are many methods to assess the fatigue life of the welded connections, relying on different approaches and quantities. Usually, stress-based approaches are employed, calculating the acting stress ranges and comparing them with experimental-derived cataloged S-N curves. These approaches are explained further in Section 2.3.1.

2.3.1 Stress-based approaches

There is a large variety of stress-based approaches, in this case, governed by standards such as EN 12663-2 [21], applicable for railway and freight wagons, and EN 1993-1-9 [30] for steel structures and welded connections. In this section, different types of stress-based approaches for freight wagons will be presented: nominal, hot-spot, notch and equivalent structural stress-based, respectively in this order.

Overall, these approaches can be divided into global and local methods. The global methods depend on global models and nominal stresses. Usually, shell elements are employed in the finite element analysis and the results lead to the tracing of nominal S-N curves. The models can vary from several meters to centimeters, depending on the structure and the nominal S-N curves include local characteristics, disregarding the detailed representation of weld profiles. The reliability of the generated results will depend on the

approximation between the real and cataloged details and the global model quality.

On the other hand, local methods are applied through global models and local sub-models. The analysis usually relies on shell or solid elements, with the latter usually being chosen due to the better definition of the load transfer mechanism allied to a higher computational effort. Furthermore, this analysis employs structural or notch stresses, with the possibility of also applying linear elastic fracture mechanics. The local fatigue model or the structural stress S-N curves will not include the local characteristics, therefore creating the necessity of detailed local sub-models. In fact, this is beneficial as models will be more detailed and the resulting fatigue parameter will already take the local parameters into account, depending on the method. Consequently, the resulting S-N curve will no longer need to approximate these characteristics, increasing the reliability of the analysis. Moreover, these models are usually employed at the millimeter scale or lower. The reliability of the results will now depend on the local sub-model and the global-local interaction. For small structures, the global model can be refined to the millimeter level.

Nominal stress approach

Firstly, the nominal stress approach is used for its simplicity of application and its important role in initial design validation. This approach calculates the average stress of the structural cross-section accounting only for macro-geometric effects, such as changes in profile or cut-outs of the material. However, this approach disregards the local stress concentration by the welded connection on the calculation of the nominal stress. Furthermore, these characteristics are then considered on the resistance of the respective nominal S-N curves, $\Delta\sigma_n$ vs N .

This approach will consequently lead to the categorization of the welded connections according to fatigue resistance classes. Each class has a specific description of the welded detail and relevant aspects to be analyzed on the connection to be studied and will have a corresponding S-N curve, following normative framework such as EN 1993-1-9 [30] and the IIW Recommendations [17]. This framework is addressed in detail in chapter 3. Nevertheless, the categorization is a subjective process that can lead to discrepancies in analysis from different sources, in addition to the diversity of resulting S-N curves for different details.

Hot-spot stress and notch stress approaches

Secondly, the hot-spot stress approach, also known as structural stress, calculates the stress at the critical weld toe by filtering out the non-linear stress peak. Moreover, although the stresses are calculated locally through detailed numerical modeling with all

the stress concentration effects of the detail, the effects regarding the weld profile are excluded. However, the presence of the weld profile is still considered. This approach proves to be effective for complex welded sub-models, where the calculation of the nominal stress is difficult to define. On the other hand, notch stresses consider the weld profile related effects, resulting in a numerical peak near this geometry.

Overall, these approaches require stress extrapolation from points located in the vicinity of the weld profile. Two possible approaches are: the surface extrapolation, where the chosen points are at specific distances from the weld, and extrapolation through the thickness of the welded plates. The results of these two extrapolation approaches are presented in Figure 2.6. Furthermore, if all of the stress concentrations effects are considered, including the ones derived from the weld profile geometry, it is possible to analyze the welded connection through a notch stress approach. This method consists in the refinement of a 3 dimensional solid element mesh and the stress results are obtained directly in the node that presents the maximum stress value. Therefore, this approach does not need an extrapolation. In summary, the hot-spot stress approach takes the stress values of the reference points and extrapolates the stress along a linear distribution up to the welded profile. The notch stress approach takes into account the non-linear nature of the stress distribution, due to the weld profile effects, leading to a stress peak near the weld profile. Figure 2.6 presents the respective results for each type of extrapolation previously mentioned for the hot-spot stress approach, alongside the representation of the values for both structural (hot-spot) and notch stresses.

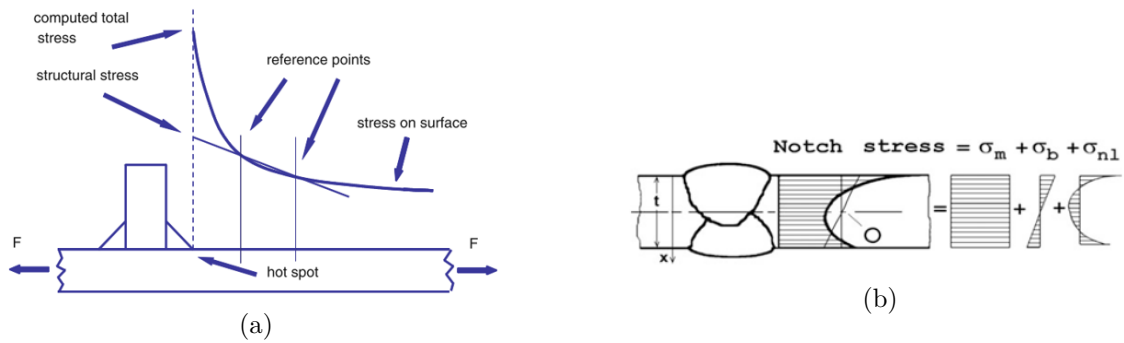


Figure 2.6: Extrapolating structural stress through the surface (a) and through the thickness (b) [33].

As Figure 2.6 presents, the non-linear components will diverge the notch stress and the structural stress. This difference will lead to the definition of notch stress, explained later in this section. As Figure 4.23b shows, when the non-linearity of the stress is taken into account, the referred stress is no longer the structural stress but instead the notch stress. Therefore, it is possible to write:

$$\sigma_{hot-spot} = \sigma_m + \sigma_b \quad (2.1)$$

$$\sigma_{notch} = \sigma_m + \sigma_b + \sigma_{nl} \quad (2.2)$$

With each of the stress components can also be written as:

$$\sigma_m = \frac{1}{t} \cdot \int_{x=0}^{x=t} \sigma(x) dx \quad (2.3)$$

$$\sigma_b = \frac{6}{t^2} \times \int_{x=0}^{x=t} (\sigma(x) - \sigma_m) \cdot \left(\frac{t}{2} - x\right) dx \quad (2.4)$$

$$\sigma_{nl} = \sigma(x) - \sigma_m - \left(1 - \frac{2x}{t}\right) \cdot \sigma_b \quad (2.5)$$

As shown in 2.2, the notch stress will consider all the stress components including the non-linear component. This stress is determined at the weld root, often referred to as the notch root in this context, taking into account the actual geometry as specified by the relevant normative framework, leading to a singular stress concentration. Another comparison between the notch stress and the hot-spot stress is presented in Figure 2.7.

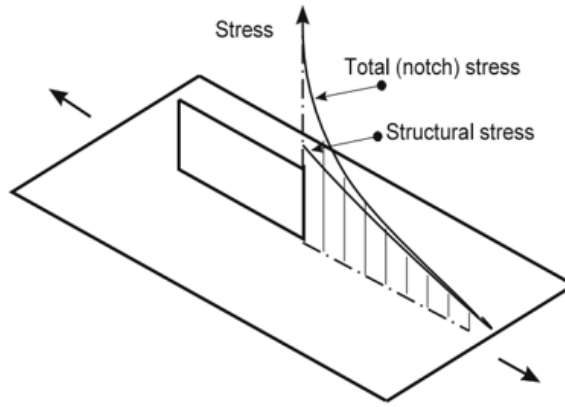


Figure 2.7: Notch and hot-spot stresses [33].

Therefore, this approach will be very sensitive to the geometry of the model, not only the macro-geometry, and to the size of the detail. In finite element analysis, the simulation needs to cover correctly all these small factors, leading to a model parameters dependent analysis, such as mesh size. However, whenever employing this approach to very small details or abrupt transitions will result in high stress values, often unrealistic. Thus, these values will not represent the true local stress state of the model.

Hence, a new approach was formulated, within the notch stress basis, employed recurring to the effective notch stress. This stipulated that the local stress can be defined

by an effective value, obtained via assuming a linear elastic behavior and maintaining the true geometry of the weld, but creating an effective value near the geometric discontinuity. Normally, this value will appear as an effective radius and for steels and aluminum alloys the value that produced the better results was $r = 1$ mm. This approach is consequently referred to as the effective notch stress approach.

Equilibrium-equivalent structural stress approach

Lastly, the equivalent structural stress approach, which was the one employed in this thesis, is based on the recovery of the nodal forces and moments of the crack plane along the weld line, ensuring the local equilibrium [13]. Hence, this approach is also referred as equilibrium-equivalent structural stress approach. For that matter, two stresses are first defined, the membrane and bending stresses, σ_m and σ_b respectively. These stresses are calculated from the nodal forces (f) and moments (m) per unit length, with t being the thickness, by the following expressions:

$$\sigma_m = \frac{f}{t} \quad (2.6)$$

$$\sigma_b = \frac{6m}{t^2} \quad (2.7)$$

The maximum structural stress is, as mentioned before, the sum of these two stresses. For proportional loading, the expression is the same but, instead of using the stress values, it is employed the stress range on the equation, giving the maximum structural stress range. Afterwards, employing this approach [13] requires the definition of certain quantities, such as the plate thickness effect and the bending ratio. The former considers the plate thickness and divides it by a reference thickness, usually equal to 1 mm [15], rendering the term dimensionless. The latter is defined by a ratio involving the absolute value of the membrane and bending stresses ranges, as shown in equation 2.8.

$$r_b = \frac{|\Delta\sigma_b|}{|\Delta\sigma_m| + |\Delta\sigma_b|} \quad (2.8)$$

The value of r_b will then be used for the determination of $I(r_b)^{\frac{1}{m}}$, where m is the exponent from the Paris law, representing a constant for each material. The $I(r_b)$ value is a dimensionless function of the bending ratio and is often presented with the exponent $\frac{1}{m}$. This function is the result of an integral along the crack through the thickness of the specimen and a stress intensity factor M_{kn} in dimensionless form. M_{kn} signifies the notch-induced magnification of the stress intensity factors as a/t approaches zero, with a and t being the crack length and thickness, respectively [13]. As a result, the considered thickness needs to be defined from the crack tip and not the entire specimen thickness,

ensuring that the integral goes from $a/t = 0$ to $a/t = 1$.

Moreover, in order to simplify the analysis using the equivalent structural stress approach, the calculation of $I(r_b)^{\frac{1}{m}}$ was approximated by a polynomial, with r_b as the only argument. Depending on whether load or displacement control is considered, the polynomial used for the calculation is different, similar to the calculations for mode I and mode III conditions. The expressions for $I(r_b)^{\frac{1}{m}}$ are shown in the following equations for mode I, in displacement (2.9) and load control (2.10), and mode III (2.11), respectively.

$$I(r_b)^{\frac{1}{m}} = 2.1549 \times r_b^6 - 5.0422 \times r_b^5 + 4.8002 \times r_b^4 - 2.0694 \times r_b^3 + 0.561 \times r_b^2 + 1.5426 \quad (2.9)$$

$$I(r_b)^{\frac{1}{m}} = 0.0011 \times r_b^6 + 0.0767 \times r_b^5 - 0.0988 \times r_b^4 + 0.0946 \times r_b^3 + 0.0221 \times r_b^2 + 0.014 \times r_b + 1.2223 \quad (2.10)$$

$$I(r_\tau)^{\frac{1}{m_\tau}} = 51.21 \times r_\tau^4 - 39.813 \times r_\tau^4 + 12.638 \times r_\tau^2 + 0.0654 \times r_\tau + 1.245 \quad (2.11)$$

$$m = 3.6 \quad (2.12)$$

$$m_\tau = 5 \quad (2.13)$$

$$r_\tau = \frac{|\Delta\tau_b|}{|\Delta\tau_m| + |\Delta\tau_b|} \quad (2.14)$$

Furthermore, the mode III expression for the definition of r_τ follows the same logic as the expression for r_b , with the main difference being the constant m . These expressions are derived from various experimental data sets and will also depend on which crack propagation mode is being considered. Normally, they are employed for the calculation of the respective mode equivalent structural stress and then are combined in order to account for more than one mode being present. Finally, these values are employed in the following equation in order to calculate the equilibrium-equivalent structural stress range for mode I and mode III, respectively.

$$\Delta S_S = \frac{\Delta\sigma_s}{t^{*\frac{2-m}{2m}} \cdot I(r_b)^{\frac{1}{m}}} \quad (2.15)$$

$$\Delta T_S = \frac{\Delta\tau_s}{t^{*\frac{2-m_\tau}{2m_\tau}} \cdot I(r_\tau)^{\frac{1}{m_\tau}}} \quad (2.16)$$

Then, the combined mode (I+III) can be calculated using:

$$\Delta S_e = \sqrt{(\Delta S_S)^2 + 3(\Delta T_S)^2} \quad (2.17)$$

Considering the resulting equivalent structural stress values with the constants present in the applicable standards, such as the ASME [14] reference values, it is possible to determine a S-N curve comparing this stress with the number of cycles, $\Delta \sigma S_S$ vs N . This curve is usually denominated Master S-N curve and is presented in detail on the next section [13].

The main advantages of taking on this approach are the mesh insensitivity and the collapse of the data set cloud around the Master S-N curve. In fact, for various types of mesh element size, only variations of around 3% are observed, presenting this quantification as an intrinsic parameter for fatigue life, until a mesh size of 2 times the thickness [15]. Moreover, the effects of the weld type, geometry and material are all implicitly present in the calculation of the equivalent structural stress, leading to the final data sets to present a narrow dispersion around the mean Master S-N curve when compared with a nominal S-N curve that presents a higher dispersion of data and has the necessity of different category curves for different details. However, there are certain considerations that this approach cannot take into account, such as the weld quality and the post-weld treatments that may have taken place. Welding reinforcements and severe discontinuities may also affect the real results.

2.3.2 Nominal S-N curve vs. Master S-N curve

The choice of the fatigue resistance curve to be employed on the analysis is a defining factor in the accuracy of the life prediction. For normal welded connections, the mean stress effect is not relevant when compared to the residual stresses effect from the welding procedure, leading to the latter being implicitly considered on the S-N curves. Furthermore, it is possible to define two main types of S-N curves, the traditional curves based on nominal stresses and the Master curve based on the equilibrium-equivalent structural stress.

The nominal S-N curve approach is the traditional method and requires the categorization of the evaluated joints into a specific FAT class, which is a fatigue class based on experimental databases. However, the categorization process may lead to a subjective step on the interpretation of standardized details, resulting in variable inputs on the same evaluation. Moreover, the definition of the nominal stress is difficult in structural components with relevant moment because the approach does not include the local characteristics of the load transfer mechanism. Therefore, these characteristics are then implicitly incorporated in the nominal S-N curves, leading to an increase scatter.

As mentioned in previous sections, the Master S-N curve is the result of the equilibrium-equivalent structural stress approach. Therefore, the method is derived from a mesh insensitive method, leading to a single universal curve. This is achieved by the implicit incorporation of the plate's thickness, the membrane and bending stresses, and the stress concentration due to geometry all being incorporated on the equivalent structural stress. Figures 2.8 and 2.9 present the difference between the nominal S-N curve and the Master S-N curve for the same dataset.

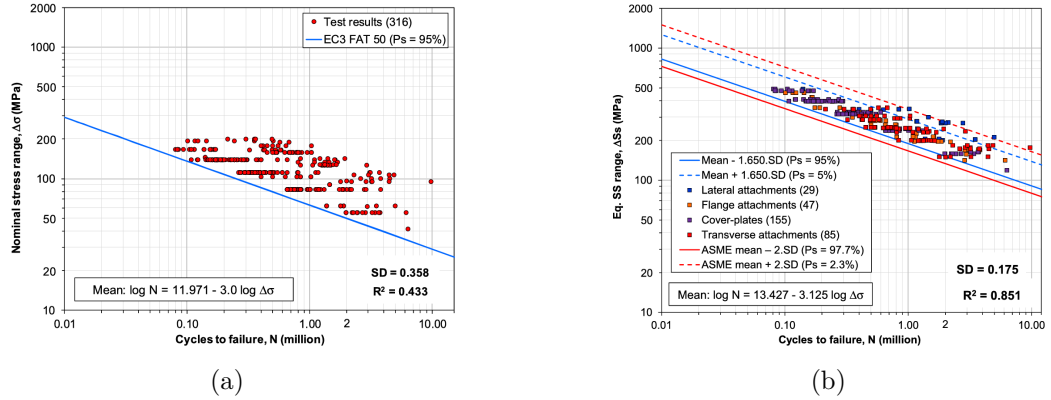
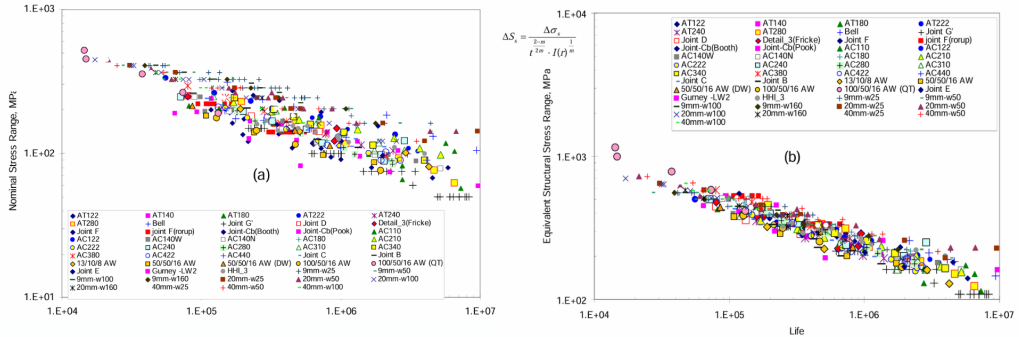


Figure 2.8: Comparison of the nominal S-N curve and the Master S-N curve for welded bridge details (adapted from Alencar et al. [15]).



mesh insensitivity and the failure analysis quality. Finally, in railway fatigue assessment, this method can increase the reliability of maintenance, based on models that consider accumulated damage and offer a better representation of the load transfer mechanism, preventing serious accidents involving rolling stock.

2.4 SmartWagons project

The advanced material characterization and high-fidelity fatigue assessment methodologies detailed in the previous sections find a direct, high-impact application within the framework of the SmartWagons Project, specifically designated as "SMART WAGONS - Desenvolvimento de capacidade produtiva em Portugal de vagões inteligentes para mercadorias". Co-funded by the Portuguese Recovery and Resilience Plan (PRR) under the NextGenerationEU initiative, this large-scale consortium is led by MEDWAY - Maintenance & Repair and brings together leading national R&D institutions, prominently featuring the Faculty of Engineering of the University of Porto (FEUP) alongside strategic partners such as INEGI, IST and ISQ [38]. The core objective of the SmartWagons agenda is to revitalize the national industrial capacity for the development and manufacturing of technologically intensive freight rail vehicles, aligning the European logistics sector with modern paradigms of digitalization, decarbonization and structural optimization [39].

Recently, the project designed an articulated half of the geometry of a Sggrss 80 freight wagon and evaluated its behavior under static, fatigue and modal analysis [40]. This articulated container wagon is presented in Figure 2.10 alongside the respective model in finite element software Ansys.



Figure 2.10: Example of Sggrss wagon (a) and generated model in Ansys (b) [41].

This thesis aims to add more analysis to this freight wagon, mainly on the assessment of the welded connections, as already presented in previous sections. For that matter, welded connections of S355 will be analyzed using the equilibrium-equivalent structural stress approach from the literature and from the wagon in study itself.

2.5 Concluding remarks

In this chapter, the fatigue phenomenon in railway freight vehicles was comprehensively characterized. Historically, there have been several incidents involving rolling stock, leading to a better fatigue life plethora of methods and better assessment of the failure data.

The classical stress-based approaches, such as the nominal stress approach, take on the problem on a global scale, often leaving out important considerations, especially when dealing with the nature of welded connections. This nature is characterized by the surface roughness, the heat-affected zone behavior and the residual stresses, all resulting from the welding procedure. Therefore, local approaches have been focus of recent research in the fatigue assessment of this connections.

The highlighted approach was the equilibrium-equivalent structural stress approach, based on the nodal forces and moments and aiming to achieve local equilibrium, hence the name. This method allows for the creation of the Master S-N curve that gets some of its advantages from the stress itself, such as the mesh insensitivity and the broader considerations regarding the connections characteristics. Hence, the dataset will present a narrower dispersion when compared to classical approaches such as the nominal S-N curve.

Finally, some considerations regarding each type of possible material possibilities were presented, highlighting the better resistance to fatigue crack initiation from high-strength steels, such as S690, when compared to current structural steels like S355. Moreover, this resistance is then balanced by the lower resistance to crack propagation from high-strength steels when compared to structural steels. These characteristics may then eliminate some of the advantages of using steels like S690 in welded connections of railway rolling stock. The SmartWagons project aims to study of the application of the S355 steel, with the previous analysis being of utmost importance in design consideration. The investigation of the behavior of high-strength steels, like the S690, in current applications is of utmost importance.

Chapter 3

Normative framework

The normative framework presented in this chapter defines the structural basis adopted for the assessment of the freight wagon under study. Since wagon design and verification must comply with both general railway engineering requirements and wagon-specific structural provisions, the analysis is based on the standards governing structural strength, load cases, validation procedures and fatigue assessment for freight rolling stock.

For that matter, the rail vehicle design is governed by many standards, each one focused on a part of the process. However, this kind of application can only be made if the standards to be applied are compatible with the main railway standards, following the same assessment criteria and philosophy. For the design of the rolling stock, as mentioned in the previous chapter, aspects such as the rail category and operating masses need to be considered. In fact, the standards presented in the following list are of general interest for the design process:

- EN 15528: "Railway applications - Line categories for managing the interface between load limits of vehicles and infrastructure" [19].
- EN 15663: "Railway applications - Definition of vehicle reference masses" [42].
- EN 15839: "Railway applications - Testing for the acceptance of running characteristics of railway vehicles - Freight wagons - Testing of running safety under longitudinal compressive forces" [43].
- EN 14363: "Railway applications - Testing and simulation for the acceptance of running characteristics of railway vehicles - Running behavior and stationary tests" [44].
- EN 16325: "Railway applications - Braking - Control systems" [45].

- EN 15085-3: "Railway applications - Welding of railway vehicles and components - Part 3: Design requirements" [46].

In addition to the previous list, the TSI standard is the most relevant for the main objectives of this dissertation and for the design of railway freight vehicles. Moreover, there are other standards that govern the design process of the railway rolling stock on the path of this thesis that can be considered due to the TSI specifications, such as the EN 12663-2 [21], EN 1993-1-9 [30], EN 17149-3 [47] and EN 13749 [48] standards and other applicable documents, such as the IIW Recommendations [17] and the DVS 1612 [18].

The Technical Specifications for Interoperability (TSI) [49] constitute the regulatory framework that harmonizes the essential technical and operational requirements of the European railway system. This framework should be understood as the legal-technical mechanism that translates the interoperability objectives of the EU into enforceable engineering constraints across subsystems such as rolling stock, infrastructure, maintenance and operation. From an engineering perspective, the TSI documents are organized by subsystem and define the boundary conditions that railway vehicles must satisfy in order to be authorized for operation within the European network. For rolling stock, the TSI establishes requirements and verification principles related to structural integrity, running dynamics, braking performance, loading gauge compatibility, axle load, safety and interfaces with infrastructure. Consequently, the design of a freight wagon cannot be treated as an isolated structural problem but must be developed within the interoperability envelope imposed by the applicable TSI provisions and related European standards. For structural and fatigue assessment work, this is particularly relevant because the TSI framework links global vehicle design with safety assurance, service performance and certification, ensuring that the final solution is not only structurally adequate, but also interoperable and fit for service across the European rail domain.

Nevertheless, the importance of the TSI in the design process does not exclude the need to use the previously mentioned standards and recommendations, regardless of their own importance. Consequently, the design framework is broader than the information present in the WAG-TSI. In order to present this framework, the following sections comprise these standards, recommendations and guidelines on freight vehicle projects, respectively.

3.1 EN 12663-2 [21]

The EN 12663 [21] standard, entitled "Railway applications – Structural requirements of railway vehicle bodies", establishes the mandatory regulatory framework for the structural assessment and verification of railway vehicle bodies within the European network.

The standard is divided into two distinct parts: Part 1 governs locomotives and passenger rolling stock, whereas Part 2 is strictly dedicated to freight wagons. Consequently, whenever EN 12663 is referenced in the context of the freight vehicle under study in this dissertation, the precise designation is EN 12663-2 [21], representing the primary normative baseline for the structural validation of freight rolling stock.

A fundamental prerequisite of EN 12663-2 [21] is the structural categorization of the freight wagon based on its intended operational profile and shunting methods. The standard distinguishes between two main categories: Category F-I, which encompasses wagons subjected to regular shunting impacts and hump shunting, and Category F-II, designated for vehicles excluded from hump shunting operations. This classification directly dictates the magnitude of the mandatory design loads.

In order to apply the structural requirements of EN 12663-2 [21] to the wagon under study, it is first necessary to define the design masses that govern the load cases. According to the adopted framework, the masses acting on the structure are divided into three main categories: the design mass of the vehicle, m_1 , the design mass of one bogie, m_2 , and the normal design payload, m_3 . The vehicle design mass, m_1 , corresponds to the mass of the wagon body in working order, excluding the bogies, while m_2 includes the mass of all equipment below and including the suspension system. Finally, m_3 represents the payload mass to be transported under normal operating conditions. These masses are particularly important because they provide the basis for the definition of the vertical forces and for the formulation of the subsequent static, stiffness, fatigue and dynamic verification cases.

Table 3.1: Definition of the design masses.

Definition	Symbol	Description
Vehicle design mass	m_1	Vehicle design mass according to EN 15663 [42], excluding bogie masses.
Design mass of one bogie	m_2	Mass of all equipment below and including the vehicle suspension. The mass of the connection elements between the vehicle body and the bogie is distributed between m_1 and m_2 .
Normal design payload	m_3	Normal design payload according to EN 15663 [42].

Once the design masses are defined, they are converted into equivalent forces through multiplication by the acceleration of gravity, g , and combined with the longitudinal and transverse actions required for structural verification. The payload mass m_3 is limited by the maximum admissible axle load, which for Portugal is 22.5 t due to the line category, and by the maximum container mass, 36 t , which leads to the following constraint:

$$m_3 \leq 22.50 - (m_1 + m_2), \quad \text{with } m_3 \leq 36 \text{ t} \quad (3.1)$$

This formulation establishes the admissible payload for each wagon platform and directly affects the vertical loading cases considered in the numerical model. In addition to the self-weight and payload-induced vertical loads, EN 12663-2 [21] also requires the consideration of longitudinal compressive and tensile loads in the buffers and coupler area, as well as loads transmitted through the body-to-bogie connection. Based on these actions, the standard defines a set of load combinations for the demonstration of static strength, structural stability, stiffness, fatigue resistance and dynamic stability. For fatigue verification, the load combinations are based on alternating vertical and transverse accelerations, written as

$$F_V = \pm 0.3 \times g \times (m_1 + m_3) + g \times (m_1 + m_3), \quad (3.2)$$

$$F_H = \pm 0.2 \times g \times (m_1 + m_3), \quad (3.3)$$

including their simultaneous application. Altogether, these expressions define the loading framework adopted in the present dissertation and provide the direct link between the EN 12663-2 [21] prescriptions and the finite element model used for the structural assessment of the wagon.

The validation methodology prescribed by EN 12663-2 [21] relies on a binary approach that categorizes loading scenarios into exceptional load cases and normal service load cases. Exceptional load cases represent extreme, low-frequency events, such as shunting impacts or emergency braking, where the structural integrity must be preserved under ultimate limit state conditions. The acceptance criteria for these cases mandate that the calculated equivalent stress must not exceed the material's lower yield strength, thereby preventing any permanent macro-deformation. Furthermore, the standard places a critical emphasis on elastic stability, requiring rigorous verification against structural buckling in thin-walled sections or slender chassis components, ensuring that the critical buckling load is never exceeded under exceptional compression. On the other hand, normal service load cases simulate the high-frequency cyclic actions encountered during routine operations, including vertical payload variations, dynamic track-twist displacements and aerodynamic or braking fluctuations. For these service conditions, the acceptance criteria shift toward ensuring that the structural utilization remains within safe fatigue margins, safeguarding the vehicle body for its entire design life.

To demonstrate compliance, EN 12663-2 [21] enforces a validation framework that combines analytical or numerical methods with experimental testing. While finite element method simulations are widely accepted for the initial design and optimization phases, the standard defines a clear hierarchy where numerical models must be validated through

physical testing unless a high degree of geometric and structural similarity to an existing proven design can be demonstrated. This empirical validation involves static proof-load testing of the prototype vehicle body, instrumented with strain gauges at critical stress concentration points, and dynamic impact tests to measure the real structural response under high strain-rate buffer collisions [21].

A key limitation of EN 12663-2 [21] is its focus on global structural verification and minimum strength requirements. Although it successfully defines the global nominal load cases and boundary conditions for the wagon body, it also leaves a technical gap regarding the localized structural assessment of critical details, such as welded connections. The standard does not provide detailed guidelines for the local fatigue assessment of welded joints, nor does it explicitly account for complex multi-axial stress fields, local notch effects or welding execution defects. Consequently, EN 12663-2 [21] must be understood as the foundational starting point and the baseline of the normative framework for railway applications, which must be systematically supplemented by specific fatigue standards, such as EN 1993-1-9 [30] or advanced cumulative damage methodologies as established in EN 17149 [47] to ensure a comprehensive durability assessment of the welded connections.

3.2 EN 1993-1-9 [30]

Following the last topic, steel structures designed in accordance with the Eurocode framework need to be in accordance with the EN 1993-1-9 [30] standard. The formal definition for this standard and respective part is "Eurocode 3: Design of steel structures - Part 1-9: Fatigue". This standard mainly focus its content on the fatigue design rules for steel structures and welded details [17, 30].

In fact, this standard provides a general fatigue design methodology based on the stress-life approach, where the fatigue performance of a detail is assessed through the comparison between the applied stress range and the resistance associated with a given detail category.

For welded railway structures, the application of EN 1993-1-9 [30] is important due to the fact that welded joints are typically the most fatigue-sensitive regions of the wagon body and underframe. In these points, fatigue damage often initiates at welded toes, weld roots or other local discontinuities where the stress field is intensified. Consequently, the use of a suitable design stress concept is essential for a reliable fatigue assessment [17, 23].

This standard also permits the use of different stress-based assessment approaches, depending on the considered detail and the level of detail of the numerical model. This

opens the possibility for the use of both global and local models, each focusing on a different modeling scale [17, 30]. This flexibility is particularly useful in freight wagon analysis as the global finite element models are often combined with local more refined and faithful sub-models to identify critical welded regions and quantify the corresponding stress ranges.

A central concept presented in EN 1993-1-9 [30] is the fatigue resistance curve, commonly referred to as the S-N curve or Wöhler curve, which relates the stress range to the number of cycles to failure, $\Delta\sigma_n$ vs N . The fatigue strength of a structural detail is expressed through a reference stress range at a specified number of cycles. The design procedure should also account for the accumulation of fatigue damage under variable-amplitude loading, usually assessed through the Palmgren-Miner damage accumulation law [20]. Thus, this framework is consistent with the empirical observation that welded details do not usually fail by yielding, but rather by crack initiation and propagation under repeated cyclic loading, as expressed on the previous Chapter 2 [30, 50]. Figure 3.1 presents the already mentioned S-N or Wöhler curves for different detail categories.

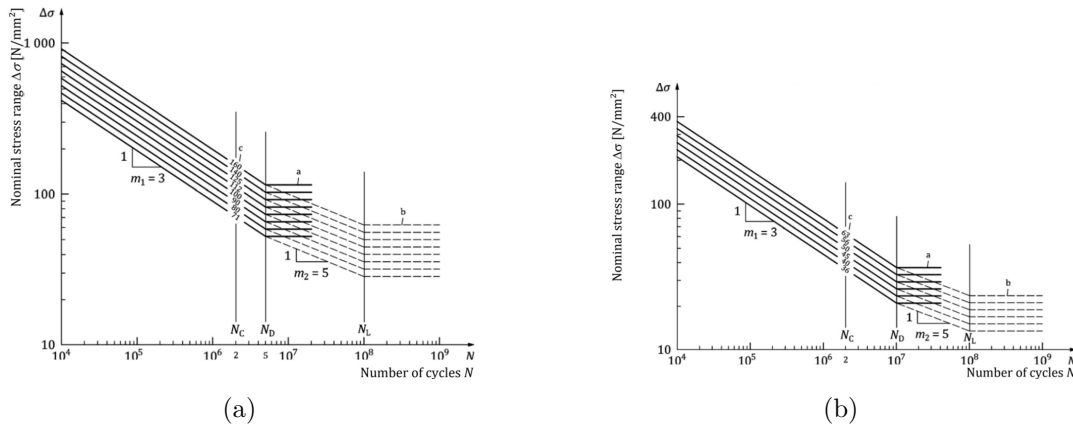


Figure 3.1: Nominal S-N curve present in EN 1993-1-9 for detail category 71 and above (a) and detail category below 71 (b) [30].

As the previous figures show, there is a total of 14 fatigue categories for nominal normal stresses. Each category is defined by a corresponding number that identifies the characteristic value of the curve at 2 million cycles. For example, the curve 100 is the fatigue class where the stress range is equal to 100 MPa at 2 million cycles. As it is also shown, these curves have a slope, m , equal to 3. After the 5 million cycles mark, this standard stipulates a value for the slope of m equal to 5. This is a very reviewed topic in the literature as some authors argue what is the optimal slope and where the knee point should be placed on the curve, for different applications [51–53]. For the case of shear stress loading spectra, there are only 2 categories, 80 and 100 [15, 30].

The EN 1993-1-9 [30] standard presents the methodology of different stress-based approaches, notably the nominal stress, hot-spot (structural) stress and effective notch stress approaches. The latter methodology and validation is mentioned very briefly and is not taken into such detail as to use this standard as a baseline for the application of these methods. However, certain cases are already stipulated in the standard with the preferable approach, such as standardized details with the nominal stress approach and reference welding configurations with the hot-spot stress approach.

These methodologies are presented in order to offer a wider choice of approaches as the fatigue categories are subjected to the engineer's vision of the details. This happens because each of the categories has a description after the corresponding welded connection type and the assessment of this description may differ across different engineers. Figure 3.2 shows part of the fatigue category definitions with the respective description, present in this standard.

Table 10.4 — Transverse welded butt joints (full penetration)

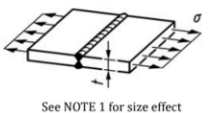
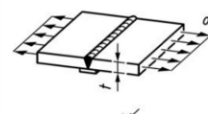
Detail category ^a	Constructional detail	Symbol	Description	Supplementary Requirements
112			① Splices in plates and flats of same thickness, welded from both sides, ground flush	Weld all-around ground flush with plate surface in direction of stress. Misalignment ≤ 5% of plate thickness, see NOTE 2.
90	See NOTE 1 for size effect		② as aforementioned, but as-welded with flank angle ≥ 150°	②: Welded in welding position PA acc. to EN ISO 6947. ②③: Weld ground flush at plate edges in direction of stress, where relevant, after removing weld run-off pieces. Misalignment ≤ 5% of plate thickness, see NOTE 2.
80			③ as aforementioned, but as-welded with flank angle ≥ 110°	
71			④ Splices in plates and flats of same thickness, welded from one side on permanent root backing, tack welds terminate not closer than 10mm to the plate edges	See ② and ③. Tack welds should attach the root backing inside the groove.
50	See NOTE 1 for size effect		⑤ as aforementioned, but tack welds terminate closer than 10mm to the plate edge and/or if a good fit cannot be guaranteed	

Figure 3.2: Fatigue categories descriptions present in EN 1993-1-9 [30].

When applied together with EN 12663-2 [21], EN 1993-1-9 [30] provides a coherent framework for both global structural verification and local fatigue assessment. The EN 12663-2 [21] standard establishes the minimum structural requirements for the wagon body, whereas EN 1993-1-9 [30] defines the fatigue verification methodology for the welded details that may control the service life of the structure. This combination is particularly relevant for freight wagons, where the global strength of the body and the local fatigue

resistance of welded joints must be assessed in a consistent and complementary manner [21, 30].

For this reason, EN 1993-1-9 [30] is commonly used with detailed finite element analysis and engineering judgment to identify critical details, define representative stress ranges and evaluate the expected fatigue performance. However, there is always room for more data to complement the already existing standards.

3.3 IIW Recommendations [17]

The fatigue assessment of welded steel structures is strongly supported by the recommendations published by the International Institute of Welding (IIW) [17]. These recommendations constitute one of the most widely used international references for the design and verification of welded joints under cyclic loading. The IIW recommendations also provide a technically consistent framework for fatigue evaluation that integrates stress definition, detail categorization, S-N curves, statistical interpretation and practical guidance for welded structural details [33]. All of the reference values, methodologies and graphs are supported by a great database specialized in welded connections, as the name of the institute suggests.

The relevance of the IIW recommendations [17] is particularly high for welded joints, which are predominant in freight wagon underframes and bodies and where structural durability is often governed by fatigue rather than by static strength. Furthermore, these guidelines are used to support fatigue design, fatigue life prediction and the assessment of welded details subjected to complex service load spectrum.

The IIW recommendations propose a very similar baseline for fatigue assessment when compared with the EN 1993-1-9 [30] standard. Similarly, the fatigue performance of a welded joint is classified through a fatigue strength category, commonly denoted as FAT, which defines the reference stress range at a specified number of cycles, usually 2 million cycles. This classification is essential because it allows welded details with similar geometric and loading characteristics to be assessed on a consistent basis.

The IIW framework differentiates several stress evaluation concepts, each of which is appropriate for a different level of detail modeling scale. The nominal stress method is suitable when the joint geometry and the load path are sufficiently well defined, as already discussed. The structural hot-spot stress method is typically used when fatigue cracks are expected to initiate at the weld toe and when the local geometric stress concentration must be captured without modeling the weld profile in full detail. The notch stress method may

be employed when the joint geometry requires a more refined representation. The choice of method must be consistent with the detail category and with the available experimental calibration basis [33, 50].

Another important contribution of the IIW recommendations [17] is the treatment of fatigue improvement methods. The guidelines include the effects of post-weld treatments such as grinding, TIG dressing and high-frequency mechanical impact (HFMI). All of these treatments may significantly improve the fatigue strength of welded joints when properly implemented and controlled, proving their relevance in certain projects where that may be necessary. These improvement methods are particularly relevant for high-strength steel structures, where the local fatigue resistance of welded details can govern the overall structural performance [54].

In the context of this dissertation, the IIW recommendations [17] are used as a complementary normative basis to EN 1993-1-9 [30] and EN 12663-2 [21]. While EN 12663-2 [21] establishes the minimum structural requirements for the wagon body and EN 1993-1-9 [30] provides the general fatigue design framework for steel structures, the IIW recommendations [17] offer the most detailed and practice-oriented guidance for the fatigue assessment of welded joints. This combination is particularly effective in freight wagon analysis where global structural compliance and local weld fatigue verification must be treated in a coherent manner [21, 30, 33].

Nevertheless, this recommendations having a similar approach to the problem at hand also improve said coherence. In fact, the survival probability is the same, the baseline is also similar and the characteristic curves are defined on the same principles. Figure 3.3 presents the nominal S-N curves of the IIW recommendations [17] and it is relevant to note that the reference class values are similar, in this case, for structural steels when compared with the EN 1993-1-9 [30].

The main advantages, apart from the ones already mentioned, are the wider collection of welded detail geometries and, therefore, more information and guidance in the design and validation stages. In addition, one of the differences resides on the definition of the knee-point, where the slope will change on the S-N curves. The IIW recommendations [17] define the knee-point at 10 million cycles, therefore leading to a slightly different curve on the high end of the high cycle fatigue spectrum. In terms of shear stress cyclic loading, this document also accounts for two values of FAT classes, once again similar to the EN 1993-1-9 [30] approach. Finally, when employing each of the methods present in this document, the conservative approach is implicit as the other methodologies will lead to more conservative curves due to the higher stress concentration factors assumed.

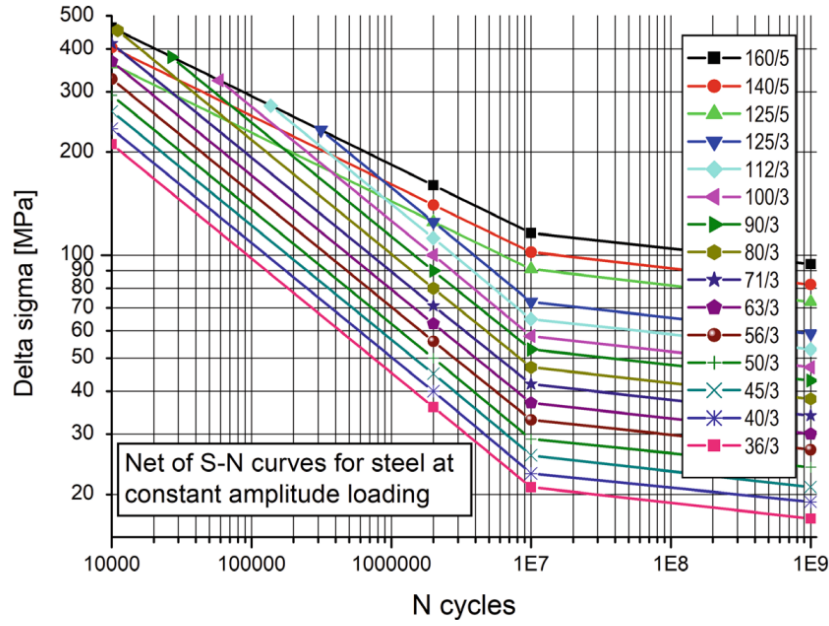


Fig. 3.2 Fatigue resistance S-N curves for steel, normal stress for high cycle applications

Figure 3.3: Nominal S-N curves present in the IIW recommendations [33].

Therefore, the IIW recommendations are a fundamental engineering reference for interpreting fatigue performance in welded railway structures, in this case, freight vehicles. Their application supports the identification of critical details, the selection of appropriate stress concepts, the assignment of fatigue categories and the estimation of service life under realistic cyclic loading conditions. For welded freight wagons, the IIW recommendations are an indispensable tool for durability assessment and design validation [23, 33].

3.4 EN 17149-3 [47]

The "EN 17149-3 - Railway applications - Fatigue strength assessment of railway structures based on cumulative damage" standard establishes a methodology for the fatigue assessment of railway structures based on the cumulative damage approach. This standard considers a methodology based on the application of a representative load spectrum, which is not defined in this standard, providing a more realistic assessment of the fatigue behavior of railway vehicles under operating loading conditions. The methodology is applicable to both base material and welded joints, considering the Palmgren-Miner linear damage accumulation rule. The assessment procedure comprises the stress history, cycle counting using the rainflow algorithm, mean stress correction, elimination of low-amplitude stress cycles, extrapolation of the measured or simulated load spectrum to the design service life and calculation of the accumulated fatigue damage. The structural detail is considered acceptable when the accumulated damage does not exceed the allowable damage defined

by the standard [47].

The EN 17149-3 [47] standard defines fatigue resistance through S-N curves characterized by fatigue strength classes expressed in terms of the reference normal stress range, $\Delta\sigma_C$, or shear stress range, $\Delta\tau_C$. For welded joints, the standard specifies different S-N curve slopes, knee points and cut-off limits to accurately represent the fatigue behavior under variable amplitude loading (VAL). Both the modified Miner rule and the consistent Miner rule are included as acceptable cumulative damage models.

This standard also introduces correction factors to account for the influence of post-weld improvement techniques, weld quality and inspection level on fatigue strength. In addition, fatigue verification incorporates partial safety factors associated with the applied loading, γ_L , and the fatigue resistance, γ_M . The latter is composed of factors related to the structural safety category, inspection and maintenance strategy, and the level of experimental validation supporting the design.

Comparing with traditional safe-life methodologies, such as DVS 1612 [18] or the combined application of EN 12663-2 [21] and EN 1993-1-9 [30], EN 17149-3 [47] explicitly adopts a cumulative damage approach that provides a more representative description of the operational loading experienced by railway vehicles throughout their service life. Consequently, this standard represents the alignment of vehicle analysis standards with other broader analyses documents, such as EN 1993-1-9 [30] and the IIW Recommendations [17], representing a significant evolution and convergence in the normative state of the art.

3.5 DVS 1612 [18]

The fatigue assessment of welded joints in railway vehicle construction is also supported by the German technical code DVS 1612 which specifically addresses the design and endurance strength analysis of steel welded joints in rail vehicles. The full denomination of this guideline is: "Technical code DVS 1612: Design and endurance strength analysis of steel welded joints in rail-vehicle construction". Similarly to what happens with the IIW recommendations, the DVS 1612 is a technical guideline presented by the German Welding Society. This code is of particular relevance for freight wagon structures because it provides a dedicated framework for the verification of welded steel joints under combined static and cyclic loading [18].

The DVS 1612 [18] guideline is especially important in the context of freight wagons because welded joints are the dominant structural joining technology in wagon underframes, bodies and local attachments. In such structures, fatigue failure is often governed by the

welded detail itself rather than by the nominal strength of the parent material. Therefore, it focuses on welded joints typical of rail applications and provides fatigue strength values, static verification rules and geometry-dependent assessment procedures tailored to this specific area. In addition to the EN 12663-2 [21], this document is another technical code directly aimed to the railway operations.

A central characteristic of DVS 1612 [18] is its reliance on the nominal stress concept. In this approach, the structural detail is assessed using the nominal stress without explicitly modeling the weld toe profile or local notch geometry. This makes the method suitable for standard welded details in railway structures where simplified fatigue verification is required. The code assigns fatigue strength values to detail classes representative of rail-vehicle welded configurations, consequently allowing a direct comparison between the calculated stress range and the permissible endurance strength. In fact, these values were already consolidated in previous versions, such as the DV952, and were then reviewed and harmonized in order to coexist with other railway standards for welded connections like the EN 15085-3 [46].

Unlike some other fatigue frameworks, the DVS 1612 [18] does not rely on a cumulative damage calculation in its base formulation. Instead, the assessment is based on endurance strength verification, which means that the structural detail is checked against an allowable fatigue threshold associated with the specified welded configuration and material grade. This makes the code particularly convenient for engineering applications where a clear acceptance criterion is required and where the stress spectrum can be represented by simplified design load cases. The simplification of the load cases is always done on the conservative side, however, testing under real operating conditions adds to the verification procedures with relevant certainty.

The code is applicable to steel welded structures used in rail vehicles and is intended for details manufactured from standard structural steels, mainly covering S235 and S355 steels as they are the most commonly used. One of the constraints of this code is a welded plate thickness of at least 2 mm, however, in freight rolling stock there is no use of structural components with a thickness of at least 2 mm.

In terms of methodology, the DVS 1612 [18] provides both fatigue and static stress checks for welded steel joints. The static assessment is based on the evaluation of the stress state at the reference location. On the other hand, the fatigue assessment is carried out against the allowable endurance strength defined by the code. This dual verification is useful in freight wagon analysis as it ensures that the same welded detail is safe under service loading and also durable over the intended operational life [18].

The main steps of this methodology are the following:

1. Define all the load cases according to EN 12663-2 [21] or EN 13749 [48].
2. Fatigue life will be verified through nominal longitudinal, transverse and shear stresses, using the finite element method, but always with the practical rule of evaluating the stresses on a zone far enough from the weld line in order to avoid local stress concentration effects.
3. The detail is classified on a notch case line and under a weld performance class, CP, from the EN 15085-3 standard [46].
4. The applied stress is then compared with the admissible stress obtained through the MKJ diagrams.
5. Further validation can be done using an equivalent approach in FEM, with measurements in static and dynamic tests and with experimental verification.
6. In the case of critical uncertainty, a real operating conditions test must be carried out.

One example of the tables present in this guideline is presented in Figure 3.4. It is worth noting the contrast of information relative to the other tables previously presented.

Finally, the Moore-Kommers-Jasper, MKJ, diagrams are reference graphs used in this technical code to indicate the admissible stress under fatigue of the welded connection and the base metal in function of the stress ratio, R , and the welded detail. These diagrams show what is the maximum admissible stress for a fatigue reference life for each detail class and type of solicitation. Including the safety factor, the reference fatigue life is usually in the order of 2 million cycles. Figure 3.5 shows an example of a MKJ diagram.

Appendix B: Construction-type catalogue
Table B-1.1: Butt joints with stresses longitudinal to the weld.

No.	Joint and seam formation Representation	Description	Type of weld	Weld no. to DIN EN 15085-3	Dressed weld face	Type and scope of test	Weld perfor- mance class according to DIN EN 15085-3	Notch case line	Remarks
1.1.1		Full penetration weld on both sides and full penetration weld on one side with backing run	Butt weld in V joint with backing run	2c	Yes ^{a)}	100 % NDT V	CP A	B	
1.1.2		Full penetration weld on both sides and full penetration weld on one side with backing run	Single-bevel butt weld with backing run	3c	No	10 % NDT V	CP B & CP C1	C ^{b)}	
1.1.3		Full penetration weld on both sides and full penetration weld on one side with backing run	Double-bevel butt weld	6	No	Visual inspection	CP C2	E1	
1.1.4		Full penetration weld on both sides ^{a)}	Square butt weld at both sides ^{a)}	1c ^{a)}	Yes ^{a)}	100 % NDT V	CP A	B	
1.1.5		Full penetration weld on both sides ^{a)}	Square butt weld at both sides ^{a)}	1c ^{a)}	No	Visual inspection	CP B & CP C1	C ^{b)}	
1.1.6		Full penetration weld on both sides ^{a)}	Square butt weld at both sides ^{a)}	1c ^{a)}	No	Visual inspection	CP C2	E1	
1.1.7		Full penetration weld on one side with temporary backing ^{a)}	Square butt weld ^{b)}	1b ^{b)}	No	10 % NDT V	CP B & CP C1	D	
1.1.8		Full penetration weld on one side with temporary backing ^{a)}	Single-V butt weld	2b		Visual inspection	CP C2	E1	
1.1.9		Full penetration weld on one side with temporary backing ^{a)}	Single-flanked single-V butt weld	2d		Visual inspection	CP B & CP C1	E1	
1.1.10		Full penetration weld on one side with permanent backing ^{a)}	Single-bevel butt weld	3b	No	10 % NDT V	CP B & CP C1	E1	
1.1.11		Full penetration weld on one side with permanent backing ^{a)}	Single-flanked single-bevel butt weld	3d		Visual inspection	CP C2	E1+	
1.1.12		Full penetration weld on one side with permanent backing ^{a)}	Single-flanked single-bevel butt weld	3d		Visual inspection	CP C2	E1+	
1.1.11		Full penetration weld on one side without backing	Square butt weld ^{b)}	1a ^{b)}	No	10 % NDT V	CP B & CP C1	E1	
1.1.12		Full penetration weld on one side without backing	Single-V butt weld	2a		Visual inspection	CP C2	E1+	
1.1.12		Full penetration weld on one side without backing	Single-bevel butt weld	3a		Visual inspection	CP C2	E1+	

Figure 3.4: Construction-type catalogue present in DVS 1612 [18].

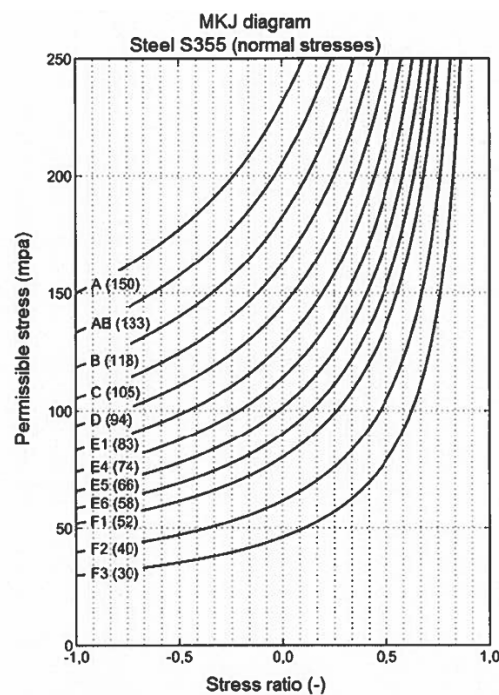


Figure 3.5: MKJ diagram for admissible normal tensile stress for S355 steel present in DVS 1612 [18].

3.6 EN 13749 [48]

The "EN 13749: Railway applications – Wheelsets and bogies – Method of specifying the structural requirements of bogie frames" [48] is a key standard for the structural specification and validation of bogie frames in railway applications. In the context of railway vehicle design, it provides the methodological basis for defining the structural requirements of the frame, the associated load cases, the acceptance criteria and the verification routes to be followed during both analysis and testing. Its role is particularly important as it connects the design process with regulatory compliance, ensuring that the bogie frame is assessed under representative service and exceptional loading conditions.

The standard is especially relevant for railway vehicles operating under the interoperability framework of the European rail system, where structural integrity, safety and repeatability of validation procedures must be demonstrated in a standardized manner. EN 13749 [48] therefore serves as a central reference for the structural verification of bogie frames, including designs for freight applications and for configurations subjected to significant cyclic loading.

A fundamental contribution of EN 13749 [48] is the separation of loading into exceptional load cases and normal service load cases. This distinction reflects the different structural demands imposed on the bogie frame. Exceptional loads are associated with infrequent but severe events and are primarily used to verify resistance against yielding, instability and ultimate failure. For exceptional loading, the standard establishes acceptance criteria based on the relationship between the calculated stress state and the relevant material resistance limits. Therefore, the structure must remain below the yield strength under permanent deformation checks and must preserve an adequate safety margin relative to ultimate strength. The analysis is not limited to numerical stress values, but is framed within a safety-based design philosophy that accounts for uncertainty in loading, geometry and material behavior [48].

On the other hand, normal service loads represent the repeated operational actions that govern fatigue behavior over the service life of the structure. For normal service loading, the standard allows the use of either an endurance limit approach or a cumulative damage approach, depending on the material behavior and the nature of the stress spectrum. When a defined fatigue limit is not available or when the operational history involves variable amplitude loading, cumulative damage methods become necessary. Usually, this is employed using the Palmgren-Miner law so that accumulated damage remains within acceptable limits over the design life [48].

Another important feature of EN 13749 [48] is the link to fatigue testing. The standard

defines validation procedures that may include static and fatigue tests on the bogie frame, with prescribed loading conditions that reproduce relevant service effects such as vertical forces, transverse forces, braking-related loads and track twist. This makes the standard particularly useful when numerical predictions need to be supported by experimental evidence. This is often the case whenever the carried out work deals with finite element analysis and structural validation.

Regarding welded connections, EN 13749 [48] is also highly relevant as bogie frames frequently contain critical fatigue welded details. In such cases, global structural compliance is not sufficient on its own because local stress concentrations, weld toe effects and residual stresses must also be considered. Therefore, the standard provides the structural framework within which more detailed welded joints fatigue assessments can be carried out using complementary methodologies.

3.7 Concluding remarks

In this chapter, the main normative baseline for the design and fatigue assessment of railway freight vehicles was presented. From all the standards mentioned, the EN 12663-2 [21], EN 1993-1-9 [30], IIW recommendations [17], EN 17149-3 [47], DVS 1612 [18] and EN 13749 [48] were the documents explored in more detail.

Firstly, EN 12663-2 [21] is a standard defined in the TSI for railways and therefore is a reference document in the area. It explores the methods in a superficial manner, but mainly focuses on the structural validation of the freight wagon's body. Consequently, it does not offer S-N curves or definitions of the fatigue actions, however, it allows for the use of other complementary and non-exclusive standards.

Afterwards, EN 1993-1-9 [30] was presented as the reference standard in metallic construction. It offers approaches based on nominal and hot-spot stresses and assesses the topic of the welded connections with detail. In addition to these documents, the IIW recommendations [17] offer the same baseline, failure probability and similar nominal S-N curves when compared to the EN 1993-1-9 [30] standard. However, due to the specialization in welded connections, it offers a wider range of geometric details to categorize the welded connections and, therefore, reduces slightly the subjective step of validation. It also changes the knee-point in the S-N curves to 10 million cycles.

EN 17149-3 [47] represents a significant advancement in the fatigue assessment of railway structures by introducing a cumulative damage approach based on a representative load spectrum, moving beyond the traditional safe-life methodologies adopted in standards

such as EN 12663-2 [21] and DVS 1612 [18]. By combining cycle counting techniques, damage accumulation according to the Palmgren–Miner rule, and calibrated partial safety factors, the standard provides a more realistic representation of the operational loading conditions experienced by railway vehicles throughout their service life.

The DVS 1612 technical code is also presented as a reference guideline with direct application on railway engineering. This code also has its system for categorization of details and validation, mainly in MKJ diagrams, which diverge from the rest of the documents.

Finally, EN 13749 [48] is also addressed and is an indispensable regulatory cornerstone for ensuring the structural integrity and operational safety of bogie frames within the railway sector. It establishes a rigorous methodology that defines exceptional and normal service load scenarios, and by requiring robust validation based on stress control, elastic stability and fatigue resistance, ensuring that the introduction of new designs, including innovative lightweight solutions, effectively mitigates the risk of catastrophic mechanical failure throughout the service life of the running gear.

In summary, EN 1993-1-9 [30] establishes the general Eurocode fatigue framework, while the IIW recommendations [17] provide a broader international perspective with various stress concepts. In contrast, DVS 1612 [18] serves as a rail vehicle-specific german technical code, precisely tailored to welded steel railway construction. As such, it is especially valuable for benchmarking, as well as for interpreting the fatigue resistance of welded details in freight wagons.

Chapter 4

Master S-N curve-based approach

This chapter is dedicated to the presentation of this dissertation's case studies, obtained from fatigue resistance data of S355 welded connections available in the literature, and the respective numerical fatigue reassessment according to the Master S-N curve approach. It also provides a detailed description of the geometrical models and the finite element analysis characteristics in Ansys Workbench. Furthermore, all of the data considered are related to fatigue resistance tests that were carried out until the complete rupture of the specimen.

4.1 Fatigue analysis methodology

The methodology used in the analysis of the case studies was different depending on the respective starting points. In fact, the data addressed in this dissertation is from the literature that, depending on the article, had points referring to nominal or equivalent structural stresses, both in amplitude and range. In order to employ the Master S-N curve approach, ΔS_S vs. N , all the data need to be presented in equivalent structural stress range. This numerical reassessment from nominal stresses is presented in further detail in 4.2. The final aspect to consider is the presence of a FEM model, which is not contemplated for all the articles presented as some geometries have similar thicknesses and the load transfer mechanism will result in similar results to the ones obtained through approximations. Nevertheless, the validation for the application of the approximations was carried out using FEM models.

Nevertheless, the difference in the starting point of the analysis of all the assessed data should be taken into account individually, allowing for a general assessment depending on the welded detail. Furthermore, each of the presented case studies corresponds to one welded connection, with different thicknesses, divided into the following categories:

- Case study 1 - Butt welds;
- Case study 2 - Transverse welds;
- Case study 3 - Cruciform welds;
- Case study 4 - Lap-joint;
- Case study 5 - Square tube-to-plate.

In all of the case studies, the data are divided by its source, allowing for a dedicated analysis of each data set, contrasting with the general considerations. As already mentioned in previous chapters, the equivalent structural stress approach allows for a better definition of the load transfer mechanism of the welded connection. Considering the load transfer mechanism, a welded joint can be categorized into load-carrying and non-load-carrying. The former corresponds to the welded details where the load is transferred through the weld profiles, while the latter corresponds to the connections where the load is transferred through the parent profiles. This categorization is important as the non-load-carrying connections will present a better fatigue life due to not being so solicited. This is also an important consideration in the application of the FEM models, which were not developed in the context of this dissertation, only modified. Finally, Figure 4.1 presents the workflow of the Master S-N curve-based approach of this thesis.

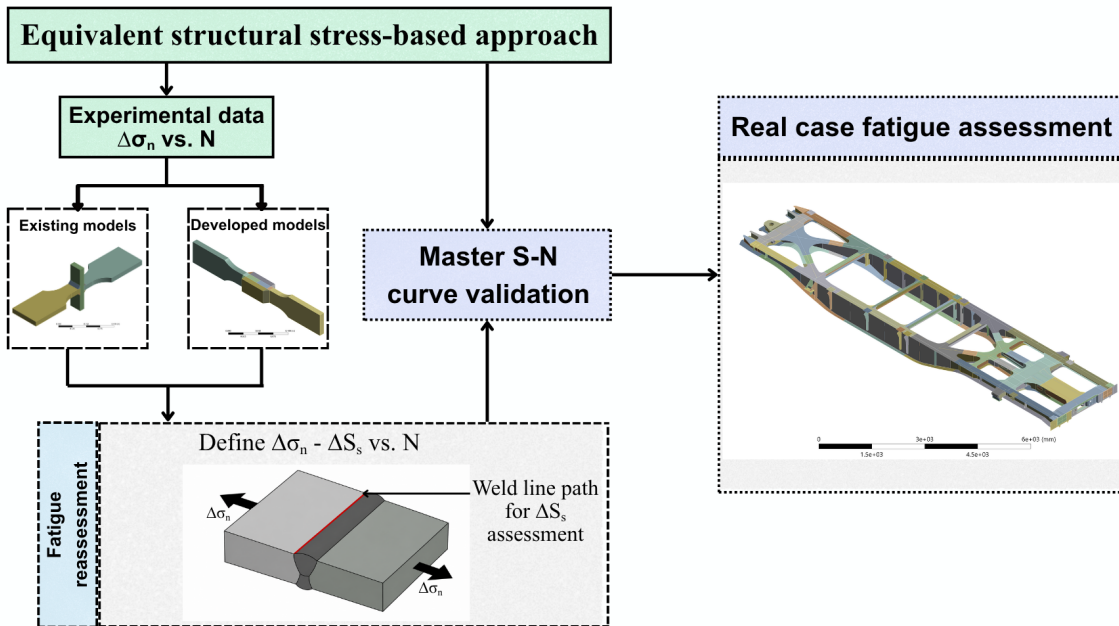


Figure 4.1: General workflow for establishing the Master S-N curve-based approach.

4.2 Numerical fatigue reassessment

The numerical fatigue reassessment performed in this work aimed to evaluate the fatigue performance of the welded details, available in the literature. As already mentioned, the transition from nominal stress to equivalent structural stress is particularly relevant in welded railway structures, with the latter combining the membrane and bending components of the stress field into a unified fatigue parameter. This allows different welded configurations to be assessed within a common framework and, therefore, makes this method especially suitable for comparing critical details with different local geometries and load paths.

In this context, the reassessment procedure is performed by considering the structural (hot-spot) stresses along the weld line that result from the presence of a nominal stress. The value of the nominal stress is applied in the FEM models through a force in the extreme ends of the welded specimen. The loads are chosen in order to create a load-controlled scenario or a nominal stress state that corresponds to the experimental conditions reported in the respective sources. This process leads to data sets of structural stress ranges and the respective membrane and bending components, needed for the definition of the equivalent structural stress. In addition, the shear membrane and bending stresses are also a result of the FEM model analysis, allowing for analyses of normal loading, Mode I, shear loading, Mode III, and for the combined Mode I+III. These outputs are possible due to the implementation of an Ansys APDL macro script package in Ansys Workbench, developed by Alencar et al. [55]. This implementation is explained in detail in Section 4.2.1.

Finally, the data for the equivalent structural stresses allow for the plot along the weld line. Then, the mean value is considered disregarding the nodes close to the start and end of the weld line, filtering out numerical anomalies, as presented in the plot of each case study. The values are compared with the ones that were obtained through manual calculation, employing equations 2.10 to 2.17 in Chapter 2, and it is possible to conclude that each of the calculation paths leads to similar results, as addressed in Table 4.1 for the comparison along the weld line and in Table 4.2 for the comparison of the mean value of the ratio of every model.

Moreover, regarding Table 4.2, for each model the simulation was run considering 3 different values of nominal stress, σ_n , that result from load-controlled conditions. The analyses were repeated for 3 different values of the nominal stress in order to verify that all the results remained within the elastic domain. That can be achieved by verifying that all the results for the analyzed nominal stresses resulted in similar ratios $\Delta\sigma_n/\Delta S_S$. Each case is identified by a letter from the source article author's name followed by a letter for

Table 4.1: Ratio between calculations and macro results along the weld line of a butt weld with 10 mm of thickness.

Distance (m)	Ratio $\Delta S_{S-manual}/\Delta S_{S-macro}$		
	Mode I	Mode III	Mode I+III
0.0000	1.0008	1.0000	1.0008
0.0025	1.0008	1.0000	1.0008
0.0050	1.0008	1.0000	1.0008
0.0075	1.0008	1.0000	1.0008
0.0100	1.0008	1.0000	1.0008
0.0125	1.0008	0.9961	1.0008
0.0150	1.0008	1.0000	1.0008
0.0175	1.0008	1.0000	1.0008
0.0200	1.0008	1.0000	1.0008
0.0225	1.0008	1.0000	1.0008
0.0250	1.0008	1.0000	1.0008

the type of welded detail and a value for the specimen's thickness. The letters Z and K refer to articles from Zhao et al. [56] and from Kassner et al. [57]. The abbreviations B , T and C refer to butt, transverse ("T"-joints) and cruciform welds respectively. Lastly, there are two values referring to a cruciform joint of 10 mm thickness from Kassner et al. [57], where the second one was obtained employing the same methodology but with linear solid elements instead of quadratic solid elements just like the rest. This difference is addressed in detail in 4.2.4.

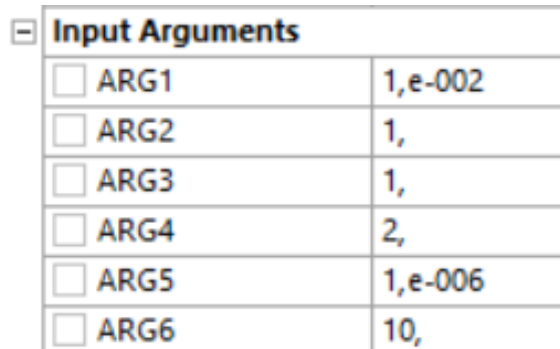
Table 4.2: Ratio between calculations and macro results for different models.

Source-Model	Ratio $\Delta S_{S-manual}/\Delta S_{S-macro}$		
	Mode I	Mode III	Mode I+III
Z-B10	1.0008	0.9987	1.0008
Z-B12	1.0008	0.9828	1.0008
Z-B13.8	1.0008	0.9935	1.0008
Z-T6	1.0008	0.5867	1.0004
Z-T8	1.0007	0.5475	1.0008
Z-C10	1.0008	0.9672	1.0008
K-B10	1.0008	0.9897	1.0008
K-C10	1.0008	0.9993	1.0008
K-C10-lin	1.0008	0.5603	0.9998

4.2.1 Ansys Workbench macro implementation

In order to implement the Ansys APDL script into the Ansys Workbench workflow, some of the definitions need to be changed. Firstly, the *Static Structural* module is created in the Workbench interface, followed by the creation of the model and the definition of the material properties in *Engineering Data*. For this dissertation, the S355J steel properties were used, as this is one of the structural steels most commonly used in rail freight vehicles. After that, in the Mechanical interface, the model properties are specified, such as mesh size and boundary conditions, alongside the macro script developed by Alencar et al. [55].

To start the changes for the implementation, inside the *Analysis Settings* module, *Retain Files After Full Solve*, *Nodal Forces*, *General Miscellaneous* and *Save MAPDL db* need to be changed to *Yes*. Then, 3 scripts are uploaded into the interface: one to assess the model characteristics, one to run the equivalent structural stress-based approach and one to export the results onto an Excel file. The first script needs 6 input arguments that correspond to the experimental conditions and model characteristics, as presented in Figure 4.2. The acceptable values for each argument and their respective definitions are presented in Table 4.3 [55].



Input Arguments		
☐	ARG1	1,e-002
☐	ARG2	1,
☐	ARG3	1,
☐	ARG4	2,
☐	ARG5	1,e-006
☐	ARG6	10,

Figure 4.2: Example of input arguments for the macro script.

From Table 4.3, it should be noted that *ARG1* and *ARG6* need to specify the same value of thickness whenever the integration is aimed at the whole thickness and not just a portion. These arguments are relevant to specify the integration range of the equivalent structural stress approach that, as seen before, starts at the crack tip if the initial crack length, a_0 , is different from 0. The nominal scaling factor refers to the stress ratio of the loading spectrum, allowing for an analysis with different ratios without post-simulation treatment, for example $ARG5 = 2 \times 10^{-6}$ that represents a stress ratio of $R = -1$ in MPa.

Therefore, taking Figure 4.2 as an example, the inputs represent a model with 10 mm of thickness, consequence of the model's units, using quadratic solid elements in an open weld line path under load-controlled conditions. A nominal scaling factor equal to 1×10^{-6}

Table 4.3: Equivalent structural stress macro options

Argument	Description	Value	Meaning
ARG1	Thickness	-	-
ARG2	Element type used in FEA	1	Quadratic Solid 20-node elements
		2	Quadratic Shell 8-node elements
		3	Linear Solid 8-node elements
		4	Linear Shell 4-node elements
ARG3	Weld line path	1	Open path
		2	Closed path
ARG4	Boundary conditions type	1	Displacement-controlled conditions
		2	Load-controlled conditions
ARG5	Nominal stress scaling factor	-	-
ARG6	Failure criterion	-	Reference thickness value

will return the data in MPa without considering the numerical effects of the stress ratio, R , and the reference thickness is 10 mm, equal to the nominal thickness of the specimen and, consequently, considering $a_0 = 0$.

Moreover, 3 named selections need to be created in order for the scripts to access the model data: weld lines, cracked weld part and start node. The first named selection is composed of the edges that represent the weld lines to be analyzed using the macro. The cracked weld part is composed of the volumes of the solids that define the combined solid geometry that would separate if a crack occurs in the direction of the thickness, relative to the selected weld lines. If the model uses shell elements instead of solid elements, this named selection should be named cracked weld areas and would be composed of faces instead of solid volumes. Lastly, the start node is the node where the analysis will start, normally corresponding to the first node of the weld lines named selection. This node is vital to the process because it corresponds to the origin of a new coordinate system. The images that illustrate these named selections will be presented for each case study that has a FEM model in the following sections.

Finally, it is also necessary to create a new auxiliary coordinate system, with its origin coincident with the start node named selection. The most important part of the definition of this new system is the directions of the axes that must comply with the position of the start node in the model's geometry. Consequently, the y-axis should coincide with the direction of the thickness and the z-axis should be tangential to the weld.

With this methodology in mind, the FEM models for the respective sources will be presented with the respective named selections and auxiliary coordinate system. If it is the case that more than one of the sources in a case study has a FEM model that was used in this dissertation, then the presentation of the previously mentioned characteristics

will only be presented once.

4.2.2 Case Study 1 - butt welds

Butt welds were addressed in the first case study. Recalling Figure 2.5, a butt weld is a welded connection between the two tops of two different solids. For this detail, 3 different sources were considered: Zhao et al. [56], Kassner et al. [57] and Wang et al. [58]. For the first two articles, there are FEM models available to implement the already addressed methodology.

Zhao et al. [56]

This article establishes and validates an effective notch stress approach for the fatigue assessment of metro bogie frames in S355J2W. The fatigue tests were made with a stress ratio $R = -1$, where the cracks appeared in the weld toe and weld root. From the article, the nominal S-N curve data was collected and applied through a load-controlled FEM simulation in Ansys Workbench. However, it is worth noting that the source does not state the welding quality according to relevant standards.

This article has nominal stress data relative to 4 butt welds, where the first three correspond to the same welding process, only differing in the specimen's thickness, and another dataset relative to a TIG welded detail. For each of the models, 3 different loads were applied in the simulation software, as mentioned before, in order to account for all the different nominal stress data points, relating both through the model's thickness. Moreover, the butt weld dimensions are detailed in Figure 4.3. These dimensions were then replicated in the model presented in Figure 4.4, where it is also possible to see the auxiliary coordinate system in the start node, as previously mentioned.

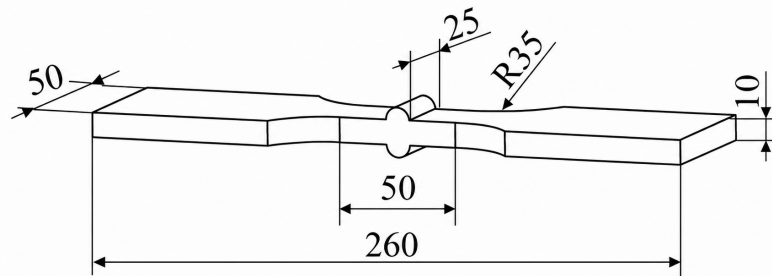


Figure 4.3: Butt joint dimensions in mm (adapted from Zhao et al. [56]).

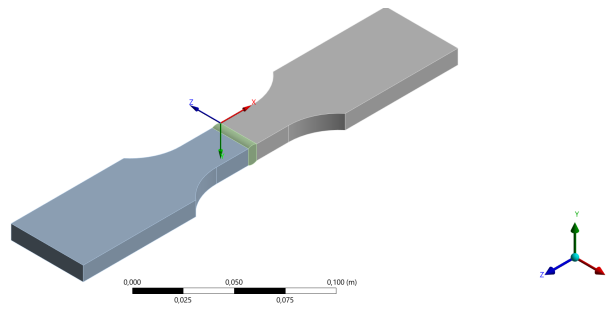


Figure 4.4: Butt joint model in Ansys Workbench, with 10 mm thickness.

Furthermore, the weld line in which the analyses will be carried out is presented in Figure 4.5 and the respective cracked weld part is shown in Figure 4.6. The boundary conditions applied were the load that corresponds to the nominal stress data from the source in one extreme and a zero-displacement condition at the opposite end, both applied to the respective faces. The mesh was generated using quadratic solid elements and, finally, the respective input arguments of the APDL script were introduced. The process was then repeated for all the other models. Finally, the resulting mesh and the von Mises stress plot are presented in Figure 4.7.

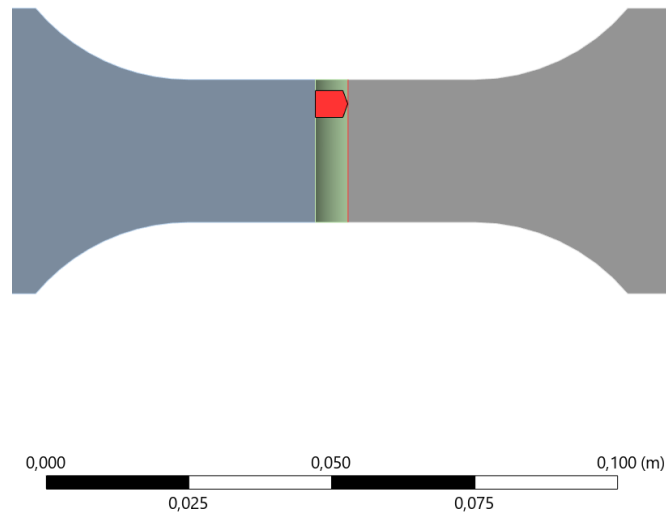


Figure 4.5: Butt joint named selection: weld line.

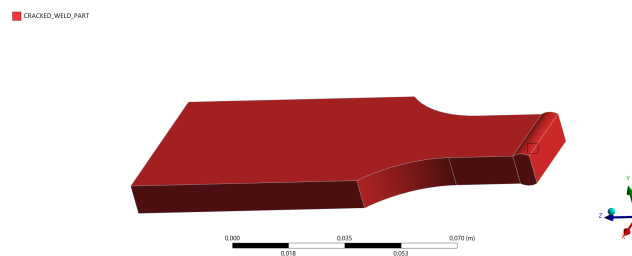


Figure 4.6: Butt weld named selection: cracked weld part.

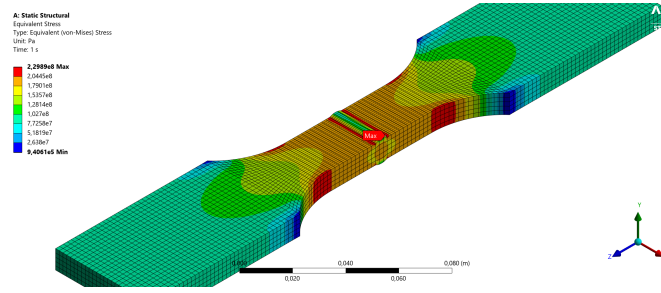


Figure 4.7: Butt weld von Mises stress plot for a load of 41.38 kN.

From the FEM analysis, the nominal stress data was then reassessed as structural stress and equivalent structural stress data along the weld line, for each simulated load magnitude. Furthermore, the distribution of the equivalent structural stresses for the combined Mode I+III along the weld line, shown in Figure 4.8, has the already-discussed stress peaks near the start and end of the analysis path.

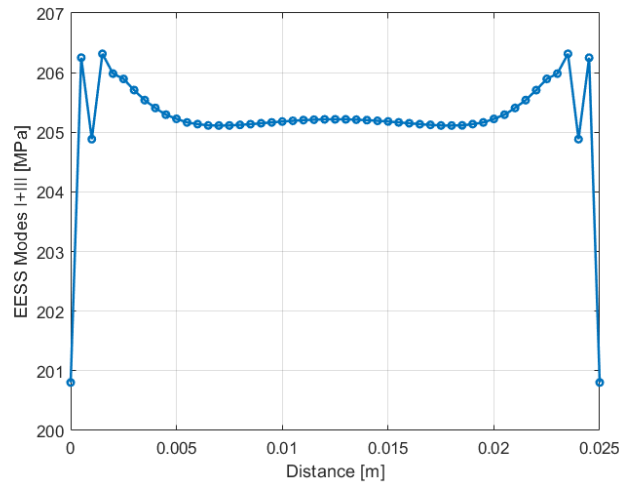


Figure 4.8: Equivalent structural stresses plot along the weld line for a butt weld with 10 mm thickness.

This numerical phenomenon needs to be addressed as the considerations of the first and last values in the mean value will lead to an unrealistic distortion of the real mean value of each detail under certain conditions. The values presented in the previous figure refer to a nominal stress condition of 150.556 MPa, implemented through a force of 37639 N of magnitude.

In terms of present opening modes, Mode I is clearly dominant as the uniaxial nature of the simulation is addressed. In fact, the values for Mode I and Mode I+III are very similar, with differences lower than 1 MPa. This is a direct consequence of the relative absence of crack opening due to shear, when compared with tensile loads, with values for Mode III around 1 or 2 MPa. Moreover, the geometry of the butt weld connection is

naturally more prone to tension when subjected to a uniaxial load-controlled scenario in the direction perpendicular to the welded surfaces. In Table 4.4 the results from the macro script are presented for 3 nodes in the middle part of the weld line, where the difference in order of magnitude for each mode is evident.

Table 4.4: Macro script outputs for 3 representative nodes.

Parameter / Node Number	15009	21946	15013
Structural stresses [MPa]	150.4	150.4	150.4
Membrane stresses [MPa]	150.4	150.4	150.4
Bending stresses [MPa]	-1.159×10^{-5}	-1.349×10^{-5}	-4.093×10^{-6}
Shear str. stresses [MPa]	-6.300×10^{-1}	-5.179×10^{-1}	-4.595×10^{-1}
Shear membrane stresses [MPa]	-6.300×10^{-1}	-5.179×10^{-1}	-4.595×10^{-1}
Shear bending stresses [MPa]	3.481×10^{-7}	3.250×10^{-7}	-4.497×10^{-7}
ESSP Mode I [MPa]	205.117	205.131	205.148
ESSP Mode III [MPa]	-1.010	-0.830	-0.736
EESS Modes I+III [MPa]	205.125	205.136	205.152
Distance [mm]	8.00×10^{-3}	8.50×10^{-3}	9.00×10^{-3}

The results of this fatigue reassessment are presented in Table 4.5, with the comparison of each of the mean values of the equivalent structural stress with the respective nominal stress scenario. All the simulated values gave the same ratio for the same thickness and, comparing with the TIG welded specimen, the ratio between equivalent structural stress and nominal stress range is also very similar. Moreover, the analyses are carried out in purely elastic regime, meaning that it is possible to extrapolate these values between themselves and verify that it is valid to assume the same ratio for the same details and thickness. In order to fill the rest of the values of the original nominal S-N curves from the source, this extrapolation approach was adopted and repeated for the other case studies as well.

Table 4.5: Results for the fatigue reassessment of Zhao et al. [56] butt welds.

Thickness [mm]	$\Delta\sigma_n$ [MPa]	$\Delta\sigma_s$ [MPa]	Ratio $\Delta\sigma_s/\Delta\sigma_n$
10	150.56	205.43	1.365
	165.51	225.84	
	185.4	252.98	
12	160.25	227.79	1.421
	174.79	248.45	
	189.1	268.79	
13.8	160.73	235.77	1.467
	176.24	258.52	
	190.63	279.62	
10 (TIG)	170.38	232.52	1.365
	189.35	258.41	
	220.44	300.83	

Kassner et al. [57]

Similar to the last article, these butt welds were tested under load-controlled conditions and the fatigue reassessment followed the same methodology. Furthermore, this source also has FEM models that allowed for the implementation of the macro scripts in the Ansys software. Nevertheless, this article distinguishes itself from the previous one with the presentation of the weld quality according to DIN EN ISO 5817 [59] and DIN EN 15085 [46]. In fact, one of the main focuses of the article is the assessment of the impact of weld quality in fatigue of welded connections in railway vehicles. The majority of the welds were classified as quality level B and some as quality level C. This categorization allows for the comparison between these results and other relevant data from articles that do not mention weld quality. Finally, this article considers a stress ratio $R = 0$ with constant amplitude until 10^7 cycles or complete failure of the specimen.

The butt weld geometry considered in this article is presented in Figure 4.9. Kassner et al. consider only one value for the thickness of the butt weld under study, equal to 10 mm. All the specimens were welded through the MAG process and the article also offers a comparison with welded connections with thermal stress relief, discussing the results.

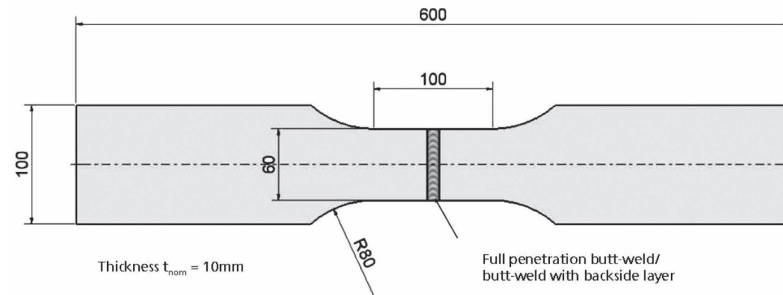


Figure 4.9: Butt joint dimensions in mm (adapted from Kassner et al. [57]).

Following the same workflow as before, the results relative to this article revealed that a butt weld with the same thickness will output almost the same ratio between the equivalent structural stress range and the nominal stress range. In this case, the ratio was equal to 1.3645, the same value as in the previous article for the same thickness and approximately the same value when compared with the TIG specimen. This proximity of the values can be seen in the expression for the equivalent structural stress, presented in equation 2.15.

In fact, as the value of $I(r_b)^{\frac{1}{m}}$ is dependent on a polynomial expression that itself depends on the bending ratio, r_b , and for butt welds the latter tends to the term of order 0, for the same thickness and m value, the ratio between the stresses will be the same. Therefore, this type of analysis may be extrapolated to butt welds with different values

of nominal stress and thicknesses, always considering that the analysis is made inside the elastic domain. The results from the macro (structural stresses) and respective nominal and equivalent structural stresses are presented in Table 4.6.

Table 4.6: Results for the fatigue reassessment of Kassner et al. butt welds.

$\Delta\sigma_n$ [MPa]	$\Delta\sigma_s$ [MPa]	ΔS_S [MPa]	Ratio $\Delta S_S/\Delta\sigma_n$
150	150.70	204.68	1.365
152	152.70	207.40	
160	160.74	218.32	
180	180.83	245.61	
190	190.88	259.26	
191	191.89	260.62	
200	200.93	272.90	
220	221.02	300.19	
237	238.10	323.39	
240	241.11	327.48	
246	247.14	335.67	
280	281.30	382.06	
285	286.32	388.88	
290	291.34	395.71	
300	301.39	409.35	

Wang et al. [58]

This article focuses on the fatigue fracture mechanisms and property evaluation of butt-welded connections in high-speed trains bogies. The analysis is divided into two scenarios, the as-welded specimens and the smooth-welded specimens. The difference between the two resides in the welding process and post-weld treatment of the weld, with the as-welded specimens having welding reinforcement and the smooth-welded specimens not having reinforcement, followed by surface polishing. For the analysis of this dissertation, only the as-welded specimens were considered. Furthermore, the fatigue data was obtained by applying a stress ratio $R = -1$ with 20 Hz frequency. All the specimens were welded using the MAG welding process, double-side three-pass. One of the main differences between the two types of specimens was the place where the crack initiated, with the as-welded specimens fracturing at the weld toe, 10/13 times, and the smooth-welded specimens fracturing in the HAZ, 6/10 times.

For this article, there is no available FEM model, however, proceeding with the conclusions from the other articles, the fatigue reassessment of the nominal stress data was possible. Figure 4.10 shows the characteristic dimensions of the specimens used by Wang

et al. [58] and, if further validation is required, the model can be created and treated in the same way as the previous ones. Finally, the nominal S-N curve defines the vertical axis as amplitude, creating the need to adjust the value according to the stress ratio of the tests and consequently only work with stress ranges.

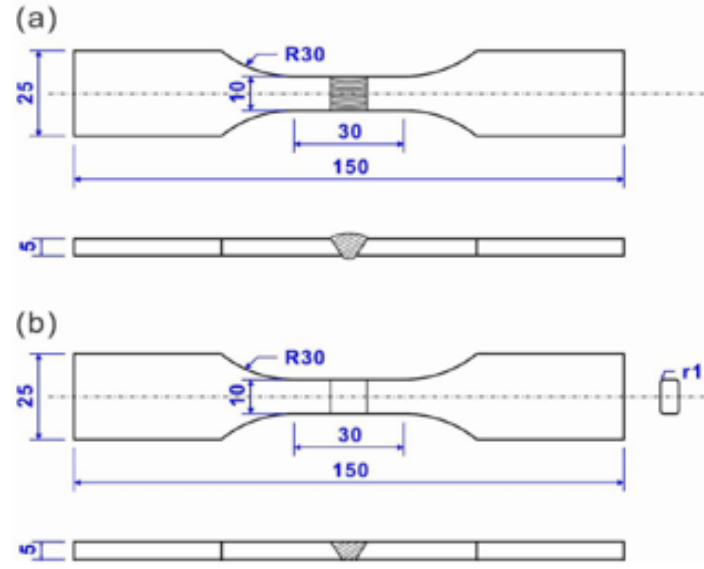


Fig. 2. Schematic diagrams of the shapes and the dimensions of fatigue specimens: (a) the as-welded specimen; (b) the smooth-welded specimen.

Figure 4.10: Butt joint dimensions in mm (adapted from Wang et al. [58]).

Table 4.7 presents the data for nominal and equivalent structural stresses obtained after the fatigue reassessment. As the treated specimens in the article are 5 mm thick and the already addressed cases do not cover this value of thickness, the transformation between nominal and equivalent structural stresses needs to be done using the equation that defines ΔS_S . For that matter, the ΔS_S expression is presented in the next equation with the values of the specimens.

$$\Delta S_S = \frac{\Delta \sigma_s}{5^{\frac{2-m}{2m}} \cdot I(r_b)^{\frac{1}{m}}} \quad (4.1)$$

As observed for the previous case, the imposed load-controlled boundary conditions will result in a Mode I-dominated result, meaning that the value for the bending stresses is much lower than the value of the membrane stresses. Therefore, the value of the polynomial will tend to the order 0 term, in this case, 1.2223. Moreover, the value of the structural stress range for butt welds in the analyzed weld line is always approximately equal to the nominal stress range, with this being a conservative approximation as the values of $\Delta \sigma_S$ tend to be higher than $\Delta \sigma_n$. The structural stress has two components, the bending and membrane stresses, and the bending stresses were already determined to be

approximately null; therefore, it is possible to consider only the membrane stresses in the equation. Consequently, it is possible to write:

$$\Delta S_S = \frac{\Delta \sigma_m}{5^{\frac{2-m}{2m}} \cdot 1.2223} \quad (4.2)$$

Table 4.7: Results for the fatigue reassessment of Wang et al.

N	$\Delta \sigma_n$ [MPa]	ΔS_S [MPa]
37852	400.00	467.96
66140	293.13	342.93
112362	291.44	340.95
207943	291.44	340.95
102486	269.51	315.31
152482	270.30	316.22
171386	270.30	316.22
157572	247.80	289.91
182724	247.09	289.07
197531	247.09	289.07
153034	223.92	261.96
210645	223.27	261.20
395954	223.27	261.20
521135	201.16	235.34
1975441	199.42	233.30
1979844	177.61	207.78

4.2.3 Case Study 2 - T-joints

A T-joint connection was considered whenever a welded joint is formed when the edge of one component is connected to the surface of another component at approximately 90 degrees, producing a T-shaped configuration. For the sake of presenting all the case studies, longitudinal and transversal attachments were considered in this section, condensing the analysis. For T-joints, 2 articles were considered: Zhao et al. [56] and Boris Fuštar et al. [54], where the first article was already broadly presented in the previous case study. In this case study, only the first article has a FEM model and the same methodology was followed.

Zhao et al. [56]

The characteristic dimensions for the specimens are shown in Figure 4.11 and the analysis contemplated 2 values for the thickness of the plates, 6 mm and 8 mm, respectively. The model for this detail is presented in Figure 4.13, alongside the respective auxiliary coordinate system, created for the implementation of the macro script. For this geometry, the boundary conditions are equal to those applied to the butt welds, with a load applied in one of the faces of the base plate and the opposite face under a zero displacement condition. The analysis of the stresses on the weld line has to be carried out on a weld line that connects two faces that are perpendicular to the load. If the chosen weld lines are those that connect the perpendicular plate to the weld profile, the returned loads will be much lower than those present on the face perpendicular to the load. In the next images, the right side face is subjected to the load and the weld line is also analyzed on that size. Therefore, the named selections and the von Mises equivalent stress plot are shown in Figures 4.13 and 4.14. The stress distribution along the weld line is presented in Figure 4.12.

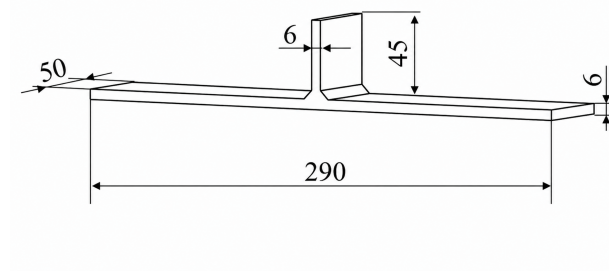


Figure 4.11: T-joint dimensions in mm (adapted from Zhao et al. [56]).

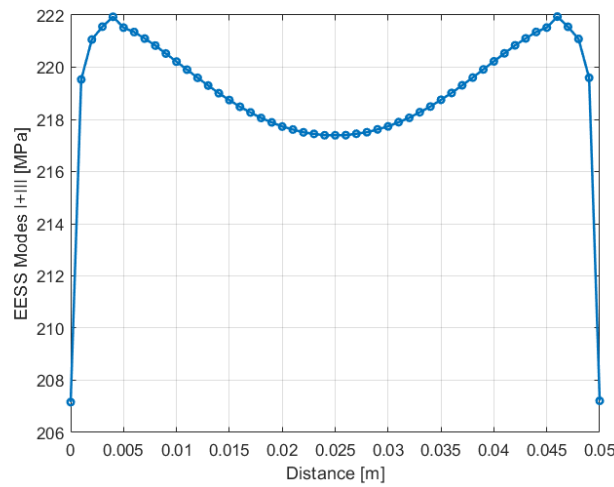


Figure 4.12: Equivalent structural stresses plot along the weld line for a T-joint 6 mm thick.

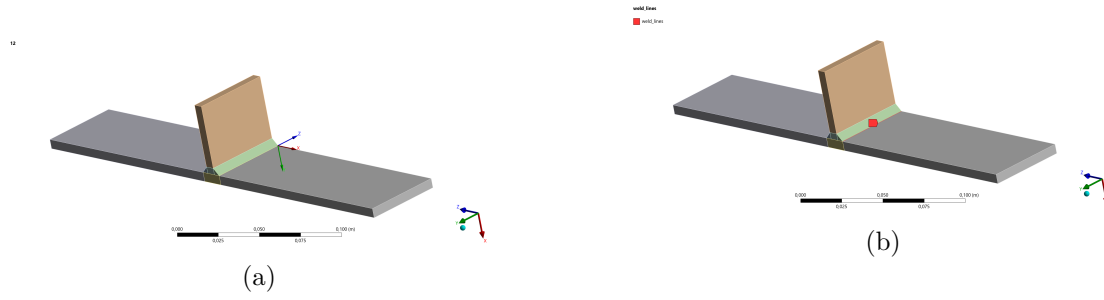


Figure 4.13: T-joint model with 6 mm thickness and auxiliary coordinate system (a) and named selection: weld lines (b).

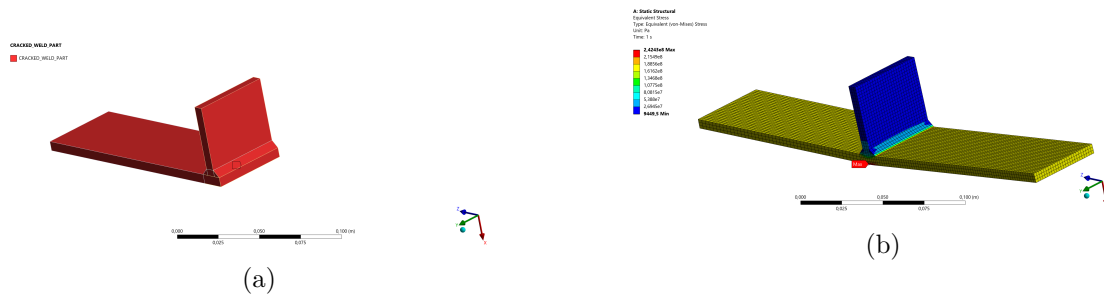


Figure 4.14: T-joint named selection: cracked weld part (a) and von Mises stress plot for a load of 54 kN (b).

Table 4.8: Fatigue reassessment results for a thickness of 6 mm.

Thickness [mm]	N	$\Delta\sigma_n$ [MPa]	$\Delta\sigma_S$ [MPa]	ΔS_S [MPa]	Ratio $\Delta S_S/\Delta\sigma_n$
6	237410	219.00	221.63	266.67	1.218
	357004				
	374824				
	441800				
	523924				
	540120				
	617542				
	339492	200.00	202.40	243.53	
	516763				
	573120				
	938590				
	1003608				
	1099587				
	571783	180.00	182.16	219.18	
	698971				
	913793				
	959407				
	1609675				
	1731836				
	1955951				

Following the same framework as before, the fatigue reassessment results are presented in Tables 4.8 and 4.9 for thickness values of 6 and 8 millimeters, respectively. The macro outputs verified the same trend that happened in case study 1 where Mode I is the dominant opening mode and presenting results very close to the ones of the combined mode, due to the low order of magnitude of Mode III.

Table 4.9: Fatigue reassessment results for a thickness of 8 mm.

Thickness [mm]	N	$\Delta\sigma_n$ [MPa]	$\Delta\sigma_S$ [MPa]	ΔS_S [MPa]	Ratio $\Delta S_S/\Delta\sigma_n$
8	191726	150.00	150.92	195.15	1.301
	220595				
	258534				
	297483				
	377401				
	450449				
	596396				
	458176	140.00	140.86	182.14	
	469496				
	517640				
	677136				
	698112				
	1808122				
	480505	130.00	130.79	169.13	
	655941				
	705768				
	879130				
	1095002				
	1306946				
1751406					

Fuštar et al. [54]

The study by Fuštar et al. provides critical high-cycle fatigue data on S355 structural steel details featuring both longitudinal and transverse non-load-carrying attachments. Special attention is given to their implementation of High-Frequency Mechanical Impact (HFMI) treatment and the subsequent quality control protocols, which are benchmarked against the baseline as-welded (AW) condition to quantify the life-extension factors under distinct stress ranges.

The article considered 2 different details, with a thickness of 8.4 millimeter thickness each, with longitudinal and transverse attachments, respectively. The characteristic dimensions for both specimens are shown in Figure 4.15 and it was used the MAG welding process in accordance with EN ISO 14175: M21 [60]. For both details, only the as-welded data was considered.

Fig. 4 Geometry of test specimens: **a** detail with longitudinal attachment, **b** detail with transverse attachment

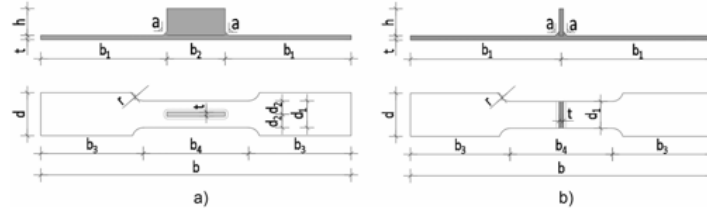


Table 5 Nominal geometrical parameters for laboratory specimens

Parameter	a (mm)	t (mm)	b (mm)	b_1 (mm)	b_2 (mm)	b_3 (mm)	b_4 (mm)	d (mm)	d_1 (mm)	d_2 (mm)	r (mm)
Specimens with longitudinal attachment	5.0	8.4	560	228	190	185	190	80	50	25	20
Specimens with transverse attachment	5.0	8.4	560	280	-	185	190	80	50	-	20

Figure 4.15: Longitudinal and transverse attachments dimensions (adapted from Fuštar et al. [54]).

As this article does not have a FEM model to implement the fatigue reassessment methodology through the macro scripts, a similar workflow relative to the one employed for the Wang et al. article in case study 1 was carried out. Moreover, the same assumptions of tensile stresses being dominant over shear stresses on the weld lines and the proximity between the values of the nominal stress ranges and structural stress ranges were made. Consequently, the equation for the calculation of the equivalent structural stresses for this article is:

$$\Delta S_S = \frac{\Delta \sigma_m}{8.4^{\frac{2-m}{2m}} \cdot 1.2223} \quad (4.3)$$

Applying this equation to all available data for as-welded specimens, the transformation from nominal stress to equivalent structural stresses data is complete. The results from the fatigue reassessment are addressed in Table 4.10. The specimen's name denotes the attachment type of the details, with *LA* and *TA* referring to longitudinal and transverse attachment, respectively. Furthermore, the article also evidences the material of the base plates and the state of the welded connections under analysis, with *AW* and *HFMI* differentiating the as-welded and high frequency mechanical impact treated specimens.

These results are obtained by taking into account the previously mentioned considerations, however, developing a FEM model to implement the macro script methodology allows for a validation of this approximation. In fact, modeling both of the specimens in Ansys resulted in approximately the same ratios between the equivalent structural stress and nominal stress ranges. For the transverse attachment detail, the ratio was equal to 1.314, presenting a similar value when compared with the T-joint from Zhao et al. [56]. This fact is due to the very low bending stress values in both model analyses and due to the similar values of thickness, of 8 mm and 8.4 mm, respectively. For the longitudinal

Table 4.10: Fatigue reassessment results for the as-welded (AW) S355 steel specimens from Fuštar et al.

Specimen	N	$\Delta\sigma_n$ [MPa]	ΔS_S [MPa]	Ratio $\Delta S_S/\Delta\sigma_n$
S355-LA-AW-01	172014	197.14	258.82	1.313
S355-LA-AW-02	185091	197.14	258.82	
S355-LA-AW-03	256144	197.14	258.82	
S355-LA-AW-04	204328	197.14	258.82	
S355-TA-AW-01	249997	197.14	258.82	1.313
S355-TA-AW-02	136918	240.00	315.09	
S355-TA-AW-03	107516	240.00	315.09	
S355-TA-AW-04	143237	240.00	315.09	

attachment, the ratio was equal to 1.332, slightly higher when compared with the transverse attachment specimens. Although the model analyses revealed that the values for the equivalent structural stresses are slightly higher when compared with the approximation approach, it is possible to conclude that the difference between the results validates the considerations and respective approximation.

4.2.4 Case Study 3 - Cruciform joints

A cruciform joint is a welded joint in which two members are attached to a third member at right angles and along the same centerline, forming a cruciform or cross-shaped geometry. For this case study, the following articles were considered for the assessment of cruciform joints: Zhao et al. [56], Kassner et al. [57], Dong et al. [13], Qi Dong et al. [61], Leitner et al. [62], Brunnhofer et al. [63], Knysh et al. [64] and Ghahremani et al. [65].

One already mentioned concern about welded connections is whether they are load-carrying or non-load-carrying, therefore influencing the FEM models. In this case study, only the first 2 sources have finite element models prepared for the implementation of the macro script framework. These models were addressed in such a way that the welded connection was always load-carrying. The main difference between these two types of considerations is shown in Figure 4.16 through a schematic representation of the welded connection and it should be noted the existence or absence of material continuity in the direction of the applied load.



Figure 4.16: Schematic of a load-carrying (a) and a non-load carrying (b) welded connection (adapted from Dong et al. [13]).

This aspect is the main consideration because the presence of continuity of the parent material will tend to allow for the load to be transferred through the plate instead of through the weld profiles. The material continuity is usually one of the considerations for the welding process, as a total penetration welded joint will ensure that the load is transferred through itself, when a partial penetration welded joint cannot ensure that. Consequently, as the numerical models do not consider the difference between welded and parent materials in the weld root and consider only a generic solid that represents the presence of the weld profiles, the plates of the welded specimen need to be addressed as the differential component of the geometry. Moreover, the original numerical models had all the adjacent faces constrained as *shared* faces, meaning that the material continuity is considered. However, if all of the material continuity is assured, then the plate that has the applied load will be modeled as a monolithic solid, leading to the load transfer mechanism to occur through the parent material and classifying the welded connection as non-load-carrying. Therefore, recalling Figure 4.16, the adjacent faces need to be shared only in interfaces where it is desirable to have the load transfer path, unsharing the faces where the horizontal plate touches the perpendicular welded attachments, in order to resemble the schematic of load-carrying connections. Figure 4.17 shows the shared and unshared faces and edges, with the latter represented in green.

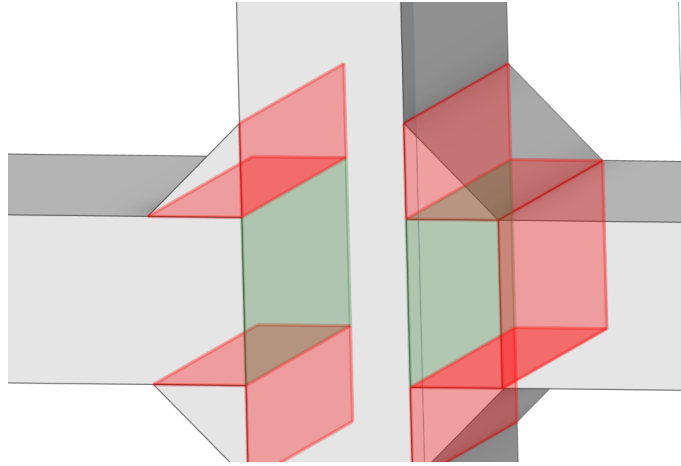


Figure 4.17: Shared and unshared features around the weld toe for a cruciform joint.

Regarding this topic, the numerical models were updated and the methodology of the macro script implementation was employed. Furthermore, for some articles the results justified the comparison with other approaches under the equivalent structural stress method. This was accomplished through the adaptation of the FEM models to consider different aspects and considerations of the same approach. All these analyses are explained under the respective sources, in the following sections.

Zhao et al. [56]

The characteristic dimensions for the cruciform details in this article are shown in Figure 4.18, with the available nominal S-N data related to one value of thickness, 10 mm. The applied boundary conditions are the same as the other models, with a force applied in one face of the base plate and a zero displacement condition on the opposite end. The analyzed weld line is once again the line between the profile and the upper face of the base plate, with the named selections defined in a similar fashion to the others already chosen. Furthermore, the FEM model used for this analysis is shown in Figure 4.19. The addressed model is the original model, with the shared faces in the weld roots that categorize the welded connections as non-load-carrying, however, those changes do not affect the overall appearance of the geometry. One of the added changes, that is also present in previously presented models, is the separation of the original solid into small solids with shared faces, and therefore numerical continuity, near the weld profile. This process allows for a better definition of the mesh around the weld profile, resulting in a more structured and uniform mesh, contrary to what happens far from this zone, where the analysis is less demanding. Finally, the model had a mesh composed of quadratic solid elements, for each the following results were obtained.

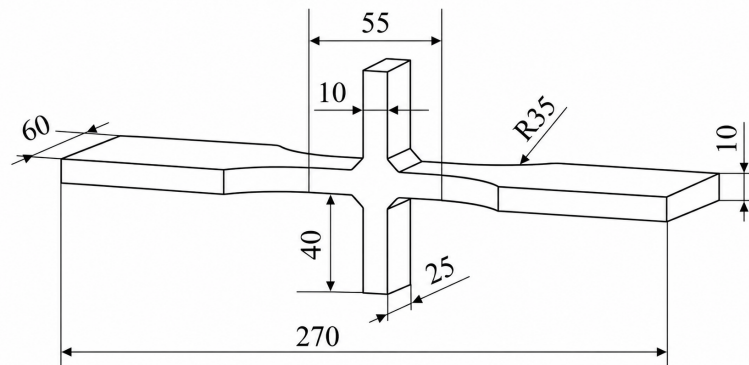


Figure 4.18: Cruciform joint dimensions in mm (adapted from Zhao et al. [56]).

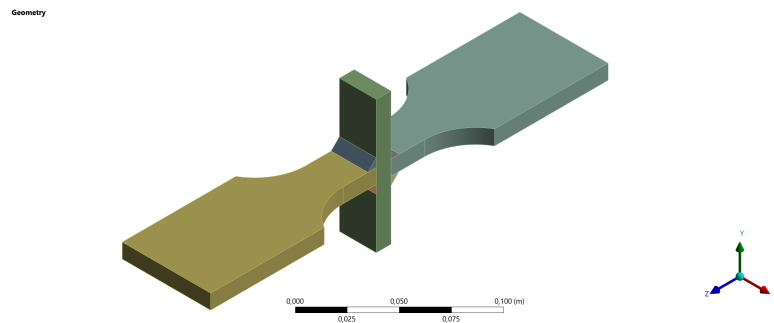


Figure 4.19: Cruciform joint model with 10 mm thickness.

From the implementation of the macro script, evaluating the failures at the weld toe, the distribution of the equilibrium equivalent structural stresses along the weld line for Modes I+III is obtained and is presented in Figure 4.20. It is possible to observe the numerical instability near the start and end of the weld line, similar to what occurred for the other case studies. Therefore, it is possible to conclude that the values from the start and end of the weld line must not be considered under the numerical analysis for the mean value of equivalent structural stresses for a certain load. However, some specimens were reported to fail at the weld root, possibly due to incomplete weld penetration. Zhao et al. [56] do not differentiate between the different failure modes when presenting the experimental results.

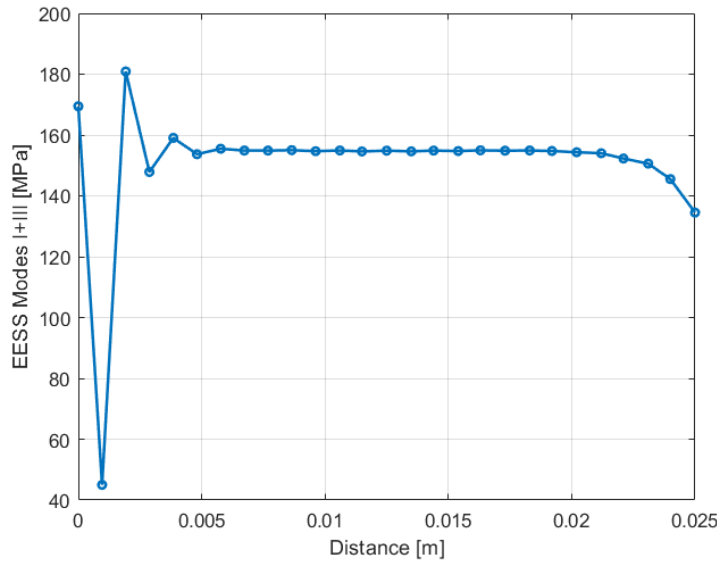


Figure 4.20: Equivalent structural stresses plot along the weld line for a cruciform joint with 10 mm thickness submitted to a 22 kN load.

Following the same framework as before, the fatigue reassessment results are presented in Table 4.11 for a thickness value of 10 millimeters. The macro outputs verified the same trend that happened in case study 1 and 2, where Mode I is the dominant opening mode and presenting results very close to the ones of the combined mode, due to the low order of magnitude of Mode III. Moreover, Table 4.12 presents the values of the equilibrium equivalent structural stresses for Mode I, Mode III and Mode I+III along the weld line for a load of 22 kN, the same analysis shown in Figure 4.20, in order to compare the order of magnitude and values of each mode. This data was then compared with the ASME [14] standard Master S-N curve and the values for this article were lower than expected. This happened for all the details of Zhao et al. [56], however, this is a matter to discuss in 4.3. For the cruciform joints, other methodologies were considered following the equivalent structural stress approach, such as: the consideration of half the thickness in the macro script inputs, analyzing only until half the depth of the thickness, and a new FEM model

that considers only one half of the thickness with imposed symmetry through boundary conditions. This considerations allow for an analysis of the welded detail where the effects of the bending stresses are considered. By analyzing half the thickness through the macro script or having a model with only half of the joint, it is possible to disregard all the symmetric effects that the bending stresses are subjected and, consequently, the values of this structural stress component will be higher. In the original approach, the $\Delta\sigma_b$ values were significantly lower when compared with the membrane stresses, where the former had maximum values in the order of 10 MPa and the latter showed values in the hundreds of MPa.

Table 4.11: Fatigue reassessment results for a thickness of 10 mm.

N	$\Delta\sigma_n$ [MPa]	$\Delta\sigma_S$ [MPa]	ΔS_S [MPa]	Ratio $\Delta S_S/\Delta\sigma_n$
247860 263453 283465 292246 304997 334227 377603 411274	112.00	132.86	154.95	1.383
215793 259152 412009 491697 454257 432618 639271 667165 944694	100.00	118.62	138.35	1.383
439119 540120 584611 602683 652328 736806 783064 754972	88.00	104.39	121.75	1.383

For the application of the two new approaches, the detail fatigue reassessment is presented in detail in this chapter. However, there is a third way to implement these con-

siderations in a FEM model. Theoretically, the equivalent structural stress approach is applied in the path of the crack that occurs in the weld toe of the detail. The cruciform joints have two effects of the weld profiles, one in each end of the thickness. Therefore, the crack will have similar probability of initiating in either of the sides of the plate, in theory. In order to implement this approach, the nodal forces and moments need to be obtained and, consequently, it is possible to calculate each node's structural stress value. Therefore, the third approach is to consider these nodal values and apply manually the definition of this method. This could be done by modifying the macro script in order to output each nodal value through the thickness, but due to time constraints this was not adopted.

Table 4.12: Macro outputs for ESSP Mode I, ESSP Mode III and EESS Modes I+III as a function of distance along the weld line.

Distance [m]	Mode I [MPa]	Mode III [MPa]	Modes I+III [MPa]
0.0000	169.43	-1.03	169.44
0.0010	44.99	-0.13	44.99
0.0019	180.89	-3.24	180.89
0.0029	147.85	-3.43	147.85
0.0038	159.04	-4.00	159.04
0.0048	153.69	-3.61	153.69
0.0058	155.49	-3.51	155.49
0.0067	154.89	-2.98	154.89
0.0077	154.89	-2.52	154.89
0.0087	155.01	-1.90	155.01
0.0096	154.69	-1.28	154.69
0.0106	154.93	-0.60	154.93
0.0115	154.60	0.02	154.60
0.0125	154.87	0.71	154.87
0.0135	154.62	1.41	154.62
0.0144	154.89	2.09	154.89
0.0154	154.73	2.77	154.73
0.0163	154.96	3.38	154.96
0.0173	154.85	4.02	154.85
0.0183	154.91	4.46	154.91
0.0192	154.75	5.03	154.75
0.0202	154.33	5.11	154.33
0.0212	154.01	5.55	154.01
0.0221	152.28	4.97	152.28
0.0231	150.60	5.30	150.60
0.0240	145.50	3.03	145.50
0.0250	134.52	2.42	134.52

Firstly, considering half the thickness in the macro script inputs only changes one step

of the whole implementation in Ansys Workbench, therefore, the rest of the methodology was followed in the same way as before. This new consideration takes into account the symmetry aspects of the stress distribution along the thickness, mainly regarding bending stresses. Considering only half the thickness in the analysis, the symmetry effects over the local bending stresses, both tensile and shear related, will theoretically be null, therefore resulting in higher values.

The equivalent structural stress approach states that the analysis of the stresses must be done through the thickness of the specimen, along the crack path. Considering half the thickness is equivalent to analyzing the stresses along the crack length until the point where the effects near the start of the crack have dominant influence. This is a possible analysis as the opposing weld line characteristics, that would correspond to the end of the analysis through the whole thickness, are dominant up until half the thickness, following the opposing direction. Consequently, the analyses of one weld line will only be carried out until the point where its local characteristic dominates the fatigue behavior of the welded connection. These considerations will result in a creation of a new model that corresponds to half of the original model, with symmetry regarding the half thickness plane imposed through boundary conditions. This last process is the second approach considered for the treatment of this detail.

The results for the first approach are shown in Figure 4.21, in equivalent structural stress, and compared with the original results. Analyzing the data, it is possible to conclude that the approach that considers only half the thickness in the macro inputs has generated higher equivalent structural stress values, that fall closer to the Master S-N curve established by ASME [14].

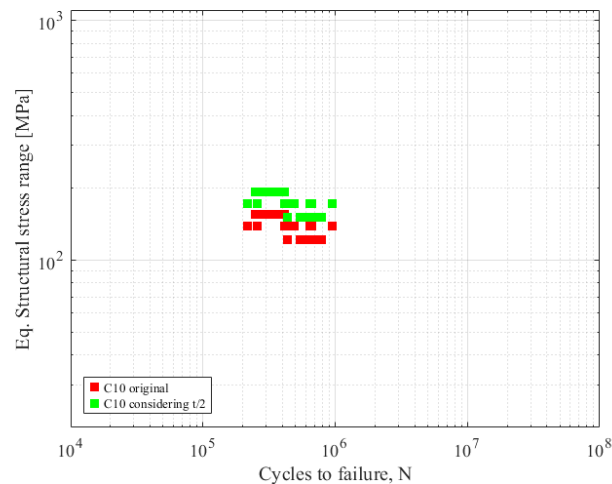


Figure 4.21: Comparison between the original fatigue reassessment data and the first new approach considering half the thickness in the macro script inputs.

The observed increase is due to the higher values of bending stresses that will influence the value of $I(r_b)^{\frac{1}{m}}$, that itself influences the values of Mode I, Mode III and, consequently, Mode I+III. The observed increase in equivalent structural stresses was of 23.8% relative to the original approach.

For the symmetric model approach, Figures 4.22 and 4.23 present the auxiliary coordinate system and the named selections. The boundary conditions that ensure the symmetry properties are simple supports along the base plate, that has the load applied at one end, in the face corresponding to the symmetry plane, except in the weld profiles and vertical attachment zones, where the boundary condition is a hinged support. The difference between the two boundary conditions resides in the degree of freedom in the direction parallel to the load, whereas the vertical displacement, perpendicular to the symmetry plane and the load, is always restrained and equal to zero.

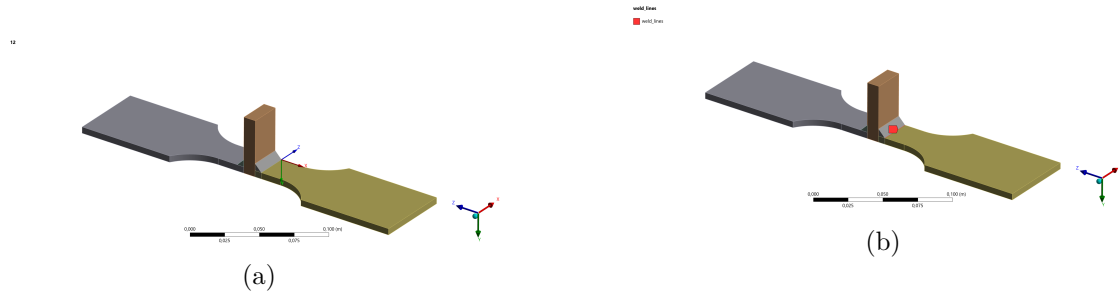


Figure 4.22: Cruciform joint model with auxiliary coordinate system and named selection: weld line (b).

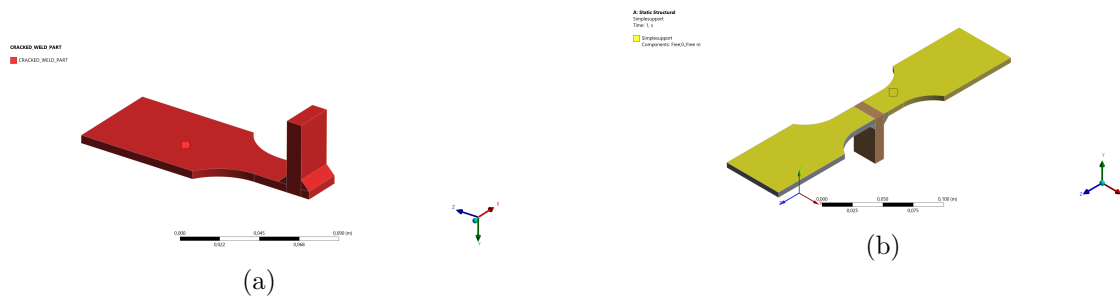


Figure 4.23: Cruciform joint named selection: cracked weld part (a) and simple support boundary conditions applied to the faces of the symmetry plane (b).

The results of the fatigue reassessment following this approach are almost equal when compared with the original approach. As the symmetry is still applied in the model, the effects over the bending stresses are still numerically applied, leading to results almost identical to the original methodology, with a maximum of 2 MPa of deviation between the data set points.

Kassner et al. [57]

This article considers the same conditions for the butt welds, presented in case study 1, and for the cruciform joints. Therefore, the methodology employed was the same in a first approach. As for the previous article the value of the cruciform welds was different depending on the considered thickness, and given that the new approach with an input equal to half the thickness, for these details the two methodologies were employed and compared. Furthermore, the work on the FEM models and considerations was the same as for Zhao et al. [56], with the addition of more load values in the simulation and different contact conditions on the weld toe, for analysis of behavior of results and further possible validation of the approach. Regarding the contact conditions employed in the models, the results presented no relevant deviations around the original values obtained without the definition of contact surfaces in Ansys. The characteristic dimensions for the cruciform joint are shown in Figure 4.24.

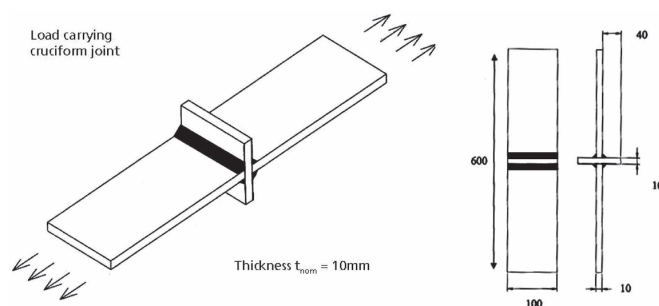


Figure 4.24: Cruciform joint dimensions (adapted from Kassner et al. [57]).

The first analysis was carried out using quadratic solid elements and the results along the weld line are presented in Figure 4.25 for a load of 106.7 kN. The numerical divergence near the extreme points is once again observed in this plot. The nominal stress data from this article accounts for 3 different manufacturers, however, those differences are not taken into account. Finally, as the article differentiates the failures by the place where the specimen cracked, only the data referring to toe failures was considered. This step clears the data from the throat fractures that are not included in the equivalent structural stress approach of this dissertation.

Furthermore, this analysis led to the results presented in Table 4.15. The ratio observed between the equivalent structural stress and the nominal stress is close to the one from the previous article when using the original methodology. For the same detail, material and thickness the ratio is proven to be approximately equal, with data from two articles and numerical models. Similar to what happened for the previous article, this approach generated points that fall a little under the lower bound usually employed for the Master

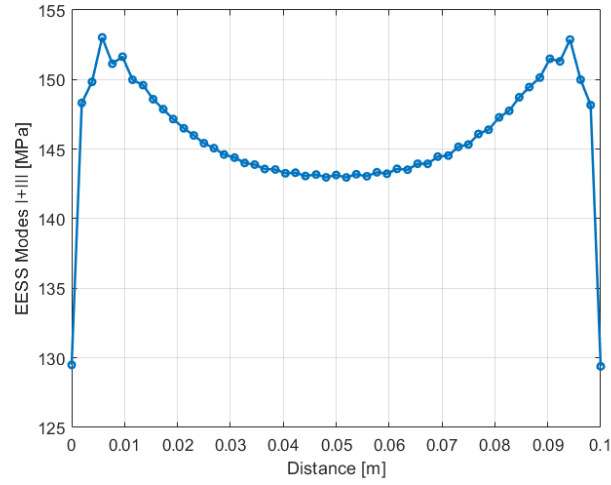


Figure 4.25: Equivalent structural stresses plot along the weld line for a cruciform joint with 10 mm thickness submitted to a 106.7 kN load.

curve, corresponding to the mean curve minus 2 standard deviations from the cataloged data.

Table 4.13: Fatigue reassessment results for a thickness of 10 mm.

N	$\Delta\sigma_n$ [MPa]	$\Delta\sigma_S$ [MPa]	ΔS_S [MPa]	Ratio $\Delta S_S/\Delta\sigma_n$
10000000	51.20	53.31	70.04	1.369
4729756	65.53	68.23	89.64	1.369
1855184	89.01	92.68	121.76	1.369
9823315	41.80	43.53	57.18	1.369
8074149	41.80	43.53	57.18	1.369
4646188	51.20	53.31	70.04	1.369
3056056	55.67	57.96	76.15	1.369
1253327	69.93	72.81	95.66	1.369
740760	89.01	92.68	121.76	1.369
108998	144.22	150.16	197.28	1.369
9736146	45.65	47.54	62.45	1.369
4064735	58.95	61.38	80.64	1.369
3166979	58.95	61.38	80.64	1.369
1471438	78.89	82.15	107.92	1.369
747392	106.67	111.07	145.92	1.369
587533	106.67	111.07	145.92	1.369
224372	133.90	139.42	183.17	1.369

For that matter, the analysis consisting of the change of input in the thickness value in the macro script was followed. Upon this framework, Ansys could not simulate the model

using quadratic solid elements, as the mesh was connecting the quadratic nodes of the base plate and the attachment. Consequently, the stress plot along the weld line was also influenced and the distribution was composed of alternating peaks and valleys of positive and negative stress values, of the same order of magnitude. Therefore, a simple adjustment was made by changing the mesh to a linear solid mesh. As mentioned previously, the equivalent structural stress approach is mesh insensitive for size and element type, with the contemplated change not changing the outcome. With this method, the obtained results are presented in Tables 4.14 and 4.15, for the structural stress components along the weld line for the same load as the original analysis and for the fatigue reassessment data to input in the Master S-N curve, respectively.

Table 4.14: ESSP Mode I, ESSP Mode III, and EESS Modes I+III in MPa along the weld line.

Distance [m]	ESSP Mode I	ESSP Mode III	EESS Modes I+III
0	204.35	-6.16	204.63
0.004	222.53	-6.23	222.79
0.008	218.89	-10.31	219.62
0.012	215.75	-10.15	216.47
0.015	212.86	-9.66	213.52
0.019	210.72	-8.62	211.25
0.023	209.17	-7.48	209.57
0.027	208.08	-6.30	208.36
0.031	207.31	-5.16	207.50
0.035	206.77	-4.06	206.89
0.039	206.41	-3.01	206.48
0.042	206.18	-1.98	206.21
0.046	206.05	-0.99	206.06
0.050	206.02	-0.00	206.02
0.054	206.06	0.98	206.07
0.058	206.20	1.98	206.23
0.061	206.44	3.00	206.50
0.065	206.81	4.05	206.93
0.069	207.35	5.15	207.54
0.073	208.13	6.29	208.41
0.077	209.23	7.47	209.63
0.081	210.79	8.61	211.31
0.085	212.93	9.65	213.59
0.089	215.83	10.14	216.55
0.092	218.98	10.30	219.71
0.096	222.60	6.23	222.86
0.100	204.48	6.14	204.75

Analyzing the results on the tables, there is no relevant change in terms of the order of magnitude of Mode III. However, if the components of the structural stresses are taken into

account along the weld line, as the macro script outputs, the change in order of magnitude is considerable, mainly in the bending stresses. This comparison is briefly shown in Table 4.16 presenting the values of the same nodes for both methodologies, considering full or half thickness. Finally, it is notable that the use of the half thickness analysis results in a ratio of 1.97 when compared with the previous 1.37, representing an increase of 43.8%, almost double when compared with the previous article.

Table 4.15: Fatigue reassessment results considering half the thickness and linear solid elements.

N	$\Delta\sigma_n$ [MPa]	$\Delta\sigma_S$ [MPa]	ΔS_S [MPa]	Ratio $\Delta S_S/\Delta\sigma_n$
10000000	51.20	92.50	100.61	1.965
4729756	65.53	118.39	128.78	1.965
1855184	89.01	160.82	174.92	1.965
9823315	41.80	75.53	82.15	1.965
8074149	41.80	75.53	82.15	1.965
4646188	51.20	92.50	100.61	1.965
3056056	55.67	100.58	109.40	1.965
1253327	69.93	126.34	137.42	1.965
740760	89.01	160.82	174.92	1.965
108998	144.22	260.56	283.42	1.965
9736146	45.65	82.49	89.72	1.965
4064735	58.95	106.51	115.85	1.965
3166979	58.95	106.51	115.85	1.965
1471438	78.89	142.54	155.04	1.965
747392	106.67	192.72	209.62	1.965
587533	106.67	192.72	209.62	1.965
224372	133.90	241.92	263.14	1.965

Table 4.16: Results of bending stresses for some nodes using both approaches.

Distance [m]	Half thickness [MPa]	Full thickness [MPa]
0.0192	75.41	9.45×10^{-3}
0.0231	74.82	1.05×10^{-2}
0.0269	74.41	1.08×10^{-2}
0.0308	74.12	1.05×10^{-2}
0.0346	73.93	9.73×10^{-3}
0.0385	73.79	8.71×10^{-3}
0.0423	73.71	7.51×10^{-3}
0.0462	73.66	6.19×10^{-3}
0.0500	73.65	4.85×10^{-3}

Dong et al. [13]

This article represents one of the baselines of the equivalent structural stress approach, with relevant data for different details. The assessment of the data is purely numerical, without the use of FEM models or other types of simulation. It compares fatigue resistance data with nominal S-N curves and with the Master S-N curve, where as expected the dispersion of the data was reduced. For this dissertation, joint C and joint B were chosen because they both represent cruciform joints, with one being a load-carrying and the other a non-load-carrying welded connection, respectively. The schematics of this details are presented in Figure 4.16 that was used for the definition of these nomenclatures.

The equivalent structural stress data was then acquired to be used in this dissertation's Master curve as points that were reviewed according to every postulate of the approach. Furthermore, the referred data is presented in Table 4.17.

Table 4.17: Fatigue data for joint C and joint B.

Joint C		Joint B	
N	ΔS_S [MPa]	N	ΔS_S [MPa]
109150	427.39	85420	420.39
150906	362.36	210472	333.65
576045	258.34	416651	264.81
1191367	200.02	450809	230.15
1289039	226.38	680299	201.68
3376892	149.84	1346721	176.73

Qi Dong et al. [61]

This source considers two types of loading: constant amplitude loading (CAL) and variable amplitude loading (VAL). Regarding the latter, a VAL spectrum is almost always the case in railway engineering as the different types of loads are dependent of different sources during the operation of the rolling stock. Therefore, this were the points that were chosen to be featured in this case study. Furthermore, only the S355 related points were taken into account, as there were other materials in the study. Finally, this article also uses an equivalent structural stress approach and compares it to nominal stress and effective notch stress approaches.

Therefore, the obtained data is already in equivalent structural stress, not needing any type of reassessment. Consequently, the final data from this article is presented in Table 4.18 without any need of calculation. This article also establishes that, for the assessed points, VAL fatigue life is lower than CAL fatigue life, once more proving the importance of the consideration of a variable amplitude load spectrum in the analysis.

Table 4.18: ΔS_S vs. N

N	ΔS_S [MPa]
83778	522.87
96533	604.80
155443	604.80
163897	608.81
175882	608.81
125721	533.35
144781	533.35
96457	492.64
123397	409.33
181798	342.36
202249	414.78
202382	492.64
217138	467.24
222848	409.33
232804	368.21
258808	370.65
331454	409.33
303250	340.10
331219	340.10
374676	320.44
395043	320.44
405741	342.36
446897	305.94
519023	278.87
670855	340.10
829221	358.59
2474395	299.92

Leitner et al. [62]

This article also considers two types of cruciform joints, one load-carrying and the other non-load-carrying. The specimens were welded using the GMAW process and all are made out of S355. The failure criterion was the complete rupture of the specimen through the thickness and the stress ratio applied was $R = 0.1$. Furthermore, the article also considers data from details that underwent post welded treatment, in this case, HFMI, resulting in a better fatigue resistance scenario when compared with the as-welded specimens. However, the scope of this dissertation is to analyze the as-welded state of the welded connections, therefore, only that data was considered in this fatigue reassessment.

There is no FEM model for this article, therefore the methodology used will be the same as in other case studies, with the approximation of the nominal stress to the structural stress values and the consideration of the term of order 0 of the characteristic polynomial $I(r_b)^{\frac{1}{m}}$ for load-controlled conditions due to the uniaxial nature of the tests. Moreover, Figure 4.26 presents the characteristic dimensions of the analyzed details in this article. This information may later be used for a fatigue reassessment study comparing the results from the approximation carried out in this dissertation and the numerical results from a developed FEM model.

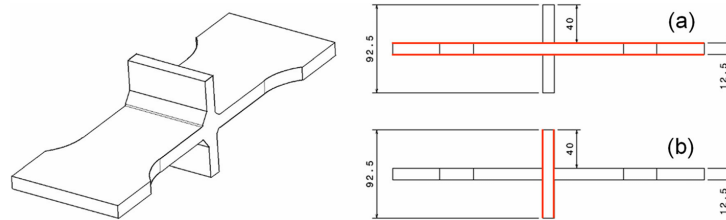


Fig. 1 Investigated joint types: transverse non-load-carrying attachment (a) and load-carrying cruciform joint (b)

Figure 4.26: Cruciform joint dimensions (adapted from Leitner et al. [62]).

The resulting equation from all the considerations stated above is:

$$\Delta S_S = \frac{\Delta \sigma_m}{12.5^{\frac{2-m}{2m}} \cdot 1.2223} \quad (4.4)$$

From this equation, the nominal stress data can be transformed into equivalent structural stress data and then be used in the Master curve with all the case studies. The results from the fatigue reassessment are presented in Table 4.19. Finally, it is worth noting that the ratio between ΔS_S and $\Delta \sigma_n$ is equal to 1.434, a value that is close to the ones of the original FEM model reassessment. This proximity can be mainly due to the approximation of the value of the bending radius, as only the order 0 term was considered. Therefore, if the analysis was made with a FEM model through half the thickness, the

importance of the bending stresses would be noted in the bending ratio, increasing the final value for the equivalent structural stress. Consequently, this means that the obtained data from the approximations will lead to a conservative reassessment for this detail. A further analysis of the validity of this approximations may be interesting to test the applicability on different details that, unlike butt welds and T-joints, do not have an expected low value for the bending stresses.

Table 4.19: Fatigue reassessment results for a cruciform joint with a thickness of 12.5 mm.

N	$\Delta\sigma_n$ [MPa]	ΔS_S [MPa]
39175	347.57	498.45
48579	321.71	461.36
58812	305.55	438.18
67847	321.71	461.36
70350	295.22	423.38
88269	282.81	405.58
100706	252.92	362.71
168300	228.14	327.18
240802	214.83	308.08
305790	202.29	290.10
813923	177.83	255.02
854505	148.47	212.92
928560	160.41	230.04
1297710	141.01	202.22
1444935	138.61	198.78
1648154	129.40	185.58
2066110	147.20	211.10
3453203	130.52	187.18
19701779	142.23	203.97
19478527	128.30	183.99
19720206	118.75	170.30

Brunnhofer et al. [63]

This article addresses similar data to Leitner et al. [62], with the same conditions and considerations. The main difference is that this article focuses specifically in cruciform joints, of S355 and S700, in as-welded and HFMI conditions. Therefore, as the thickness and geometry is the same, considering the nominal S-N curve data, the fatigue reassessment is identical to the previous article. Therefore, it is possible to write directly the fatigue reassessed results, that are presented in Table 4.20.

Table 4.20: Fatigue reassessment results for cruciform joint with thickness of 12.5 mm.

N	$\Delta\sigma_n$ [MPa]	ΔS_S [MPa]
40506	354.88	508.94
50836	334.63	479.89
62205	309.32	443.59
70590	328.37	470.92
72372	291.60	418.18
59860	253.71	363.84
101711	230.13	330.04
104335	249.13	357.27
91990	283.26	406.22
177286	228.23	327.31
243075	219.56	314.88
312858	205.01	294.00
827013	182.54	261.79
869437	152.76	219.07
961969	158.98	227.99
1352425	142.71	204.66
1458842	141.33	202.69
1740592	130.63	187.34
2104292	147.20	211.10
2744539	172.61	247.53
3530539	132.20	189.59
19921598	145.25	208.30
19917811	135.52	194.35
19915647	130.25	186.80
19910780	119.15	170.87

Knysh et al. [64]

This article focuses on the improvements of HFMI-treated specimens with lack of fusion. As mentioned before, the lack of fusion in cruciform joints may lead to the load transfer mechanism to disregard the welded material and travel through the parent material instead. This leads to the categorization of such details as non-load-carrying, and that is exactly the case study of the article. For that, S355 specimens welded by manual metal arc welding were conceived with the characteristic dimensions presented in Figure 4.27.

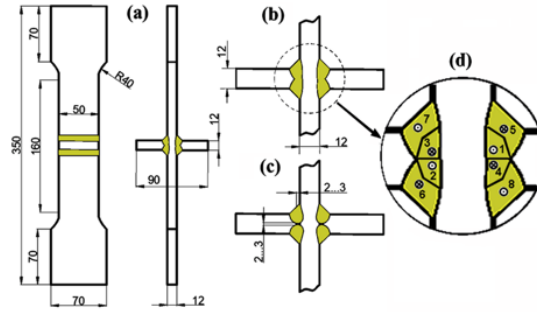


Fig. 2. The chart of fatigue test specimen with the transverse non-load-carrying attachments: (a) the main dimensions, (b) welds with full penetration and (c) welds with incomplete penetration ('lack of fusion'). (d) The sequence and directions of the welding passes used to produce the weld root (1-4) and cap welding (5-8), the pointed and crossed circles respectively indicate the toward and outward directions from the reader.

Figure 4.27: Cruciform joint dimensions (adapted from Knysh et al. [64]).

The data of the nominal S-N curves of the article come in as-welded and HFMI-treated specimens. For the scope of this dissertation, only the as-welded details were considered for the fatigue reassessment. The fatigue resistance tests were carried out under a stress ratio of $R = 0.01$. Furthermore, as there is no FEM model for this article, the same reassessment as done in previous articles was followed. For that matter, the equation that defines the main reassessment change is:

$$\Delta S_S = \frac{\Delta \sigma_m}{12^{\frac{2-m}{2m}} \cdot 1.2223} \quad (4.5)$$

Finally, the results of the fatigue reassessment are presented in Table 4.21. The ratio between the equilibrium structural stress and the nominal stress is equal to 1.4212, similar to the value for the previous articles. Once again, the approximations of disregarding the relevance of the value of the bending stresses will influence and reduce by some margin the final fatigue reassessed data. This ultimately leads to a conservative result of the engineering safety curve in excess, meaning that a detail will have better fatigue resistance life than predicted.

Table 4.21: Fatigue reassessment results for a cruciform joint with a thickness of 12 mm.

N	$\Delta \sigma_n$ [MPa]	ΔS_S [MPa]
127427	281.01	399.35
137704	244.18	347.02
191472	232.64	330.62
232443	260.39	370.05
230200	214.97	305.51
390462	230.02	326.90
615848	179.66	255.32
796256	199.69	283.79
2009823	149.59	212.59
2150956	160.72	228.41

Ghahremani et al. [65]

Finally, this article considers two specimen types, one load carrying and one non-load carrying welded connection, subjected to as-welded and treated toe conditions (HFMI). These details were fatigue tested under three load histories including two variable amplitude loading spectrums and one constant amplitude loading with the stress ratio $R = 0.1$. All of the fatigue tests were conducted in a servo-controlled testing frame with the loading frequency of 30–100 Hz, depending on the loading magnitude. The non-load carrying specimen, Type-X, is a cruciform joint and is the one that is relevant for this case study. The load-carrying joint, Type-L, will be addressed in the next case studies as it is a lap-joint. The characteristic dimensions of the welded specimens are presented in Figure 4.28.

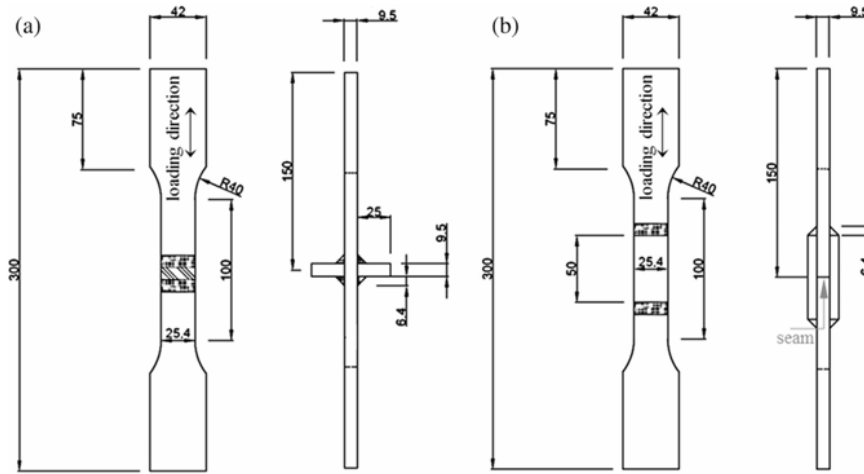


Fig. 1. Fatigue specimen geometries (all dimensions in mm): (a) transverse stiffener (Type-X); (b) lap joint (Type-L).

Figure 4.28: Cruciform joint and lap-joint dimensions (adapted from Ghahremani et al. [65]).

As this article does not possess a FEM model, the methodology used was the same as the other sources in the same position. Once again, the approximations will lead to a conservative value for the equivalent structural stress. The equation that defines that main reassessment is presented in eq. 4.6 and the fatigue reassessment results are presented in Table 4.22. The difference resides in the fact that the structural (hot-spot) stress range is presented in a S-N curve for each detail, allowing for the direct use of this value instead of approximating to the value of the nominal stress range.

$$\Delta S_S = \frac{\Delta \sigma_S}{9.5^{\frac{2-m}{2m}} \cdot 1.2223} \quad (4.6)$$

Table 4.22: Fatigue reassessment results for a cruciform joint with a thickness of 9.5 mm.

Loading case	N	$\Delta\sigma_S$ [MPa]	ΔS_S [MPa]
CA	902135	197.84	266.94
CA	1996731	150.34	202.84
VA1	407590	246.38	332.43
VA1	734201	163.06	220.01
VA1	1108481	148.15	199.89
VA1	3814781	114.22	154.11
VA2	615372	200.47	270.49
VA2	1014822	160.91	217.11
VA2	1938831	114.10	153.95
VA2	8198568	90.46	122.05

For this article and results, the ratio between $\Delta\sigma_S$ and ΔS_S , that results from the approximation of the value of $I(r_b)^{\frac{1}{m}}$, is equal to 1.349. Once again, the value is near to the ones already calculated and if all of these cases end up near the lower bound of the standardize Master curve it is possible to point out this approximations as the main culprit. However, that assessment will be made for all the case studies and data in 4.3.

4.2.5 Case Study 4 - lap-joints

In this case study, a lap-joint from the Ghahremani et al. [65] article is presented. This specimens were subjected to the same conditions as the cruciform joints presented previously and the dimensions are expressed in Figure 4.28. Furthermore, for this welded detail there are no previous results or considerations that were verified through the implementation of the macro script in FEM models. Therefore, this case study presents 3 different approaches and respective results, the first with the approximation carried out for the cruciform joints disregarding the bending stress effects and the other two implementing the model in Ansys and comparing the results for full thickness and half thickness analysis.

For the approximation approach, the respective equation is presented in 4.7 for the thickness of 9.5 mm of the base plate. The specimens will break on the weld toe or weld root, with the second case following the already existing space between the plates. This spacing is present on the specimens in order to ensure the load-carrying characteristic for the welded connection and the space corresponds to the contact interfaces between the base plates and the attachments. The two types of weld root cracks for the Type-L specimens are presented in Figure 4.29. However, this cracks were only observed for the HFMI-treated connections, therefore, for the scope of this thesis, only the as-welded connection failure modes are considered. For AW specimens, the observed failures occurred

in the weld toe and a failed specimen is presented in Figure 4.30. The obtained results from equation 4.7 are shown in Table 4.23.

$$\Delta S_S = \frac{\Delta \sigma_S}{9.5^{\frac{2-m}{2m}} \cdot 1.2223} \quad (4.7)$$

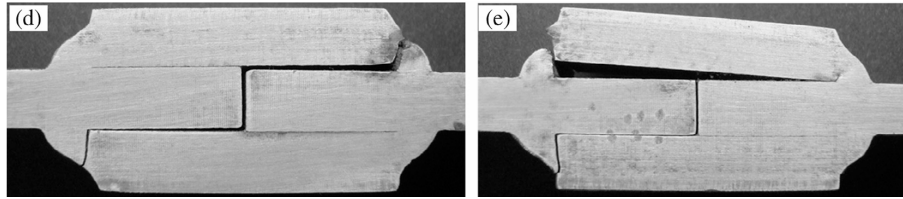


Figure 4.29: Observed root failure in treated lap specimens (d and e) (adapted from Ghahremani et al. [65]).

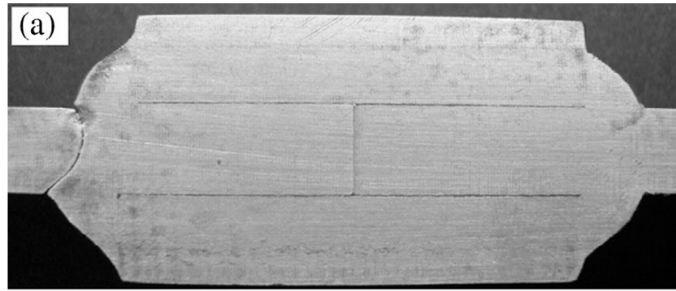


Figure 4.30: Observed toe failures in as-welded lap specimen(a) (adapted from Ghahremani et al. [65]).

Table 4.23: Fatigue reassessment results for a lap joint.

Loading	N	$\Delta \sigma_S$ [MPa]	ΔS_S [MPa]	Ratio $\Delta S_S / \Delta \sigma_S$
CA	16220	570.10	769.21	1.349
CA	26249	464.16	626.27	1.349
VA1	54048	329.50	444.58	1.349
VA1	443576	155.05	209.20	1.349
VA1	636537	129.75	175.06	1.349
VA1	1059566	138.95	187.48	1.349
VA1	1564981	138.95	187.48	1.349
VA2	111200	264.62	357.04	1.349
VA2	885888	119.50	161.24	1.349
VA2	9492762	93.38	125.99	1.349

For this case study, a FEM model was also developed in order to compare the results from the implementation of the macro script with the results considering the approximation. The consideration of the model is more relevant in this case study due to the fact that there were no results obtained by this approach for a lap-joint welded connection. This is different to the scenario observed in the previous case studies where there was always the verification of the connection through the model implementation, where only the thickness varied between articles. Moreover, Figure 4.31 presents the model that represents the Type-L specimen considered by Ghahremani et al. [65].

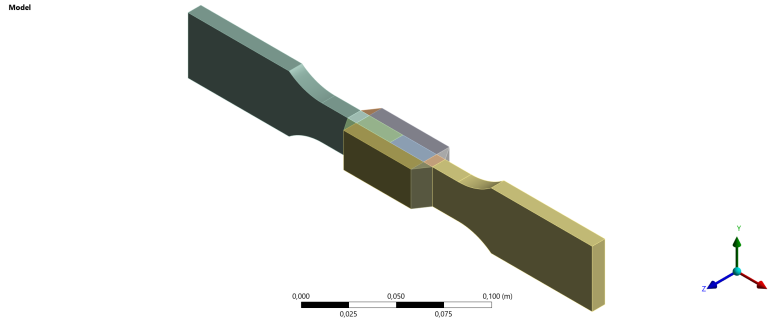


Figure 4.31: Lap-joint model.

Furthermore, the named selections were defined following the same framework as the previous case studies, with the analysis being carried out on the weld toe, where the as-welded specimens failed. The forces applied on the model were calculated based on the nominal stress ranges for the different points presented in the article. The obtained results are addressed in Table 4.24.

Table 4.24: Fatigue reassessment results for a lap-joint using a FEM model.

Loading	N	$\Delta\sigma_S$ [MPa]	ΔS_S [MPa]	Ratio $\Delta S_S/\Delta\sigma_S$
CA	16220	534.32	712.21	1.333
CA	26249	435.03	580.28	1.334
VA1	54048	308.82	413.65	1.339
VA1	443576	145.32	192.28	1.323
VA1	636537	121.60	159.48	1.311
VA1	1059566	130.23	169.74	1.303
VA1	1564981	130.23	169.74	1.303
VA2	111200	248.01	328.13	1.323
VA2	885888	112.00	151.18	1.350
VA2	9492762	87.52	112.68	1.287

From Tables 4.23 and 4.24 it is possible to observe that the values for the ratios between equivalent structural stresses and structural stresses for the approximation approach and for the numerical model implementation return similar results. However, the main difference between the two approaches results is the value of the structural stress obtained along the weld line. The results for the approximation are relative to the structural stress results presented in the article and the macro implementation results occur from the application of a force that induces the respective nominal stress range. This difference is not noted in the respective analyses ratios, but it is revealed when comparing the final equivalent structural stress ranges. In fact, the difference between these values presents differences between 6.24% and 10.57%, as addressed in Table 4.25.

Table 4.25: Fatigue reassessment results for a lap-joint using a FEM model.

Loading	N	ΔS_S [MPa]	Macro ΔS_S	Ratio $\Delta S_S / \text{Macro} \Delta S_S$
CA	16220	769.21	712.21	1.080
CA	26249	626.27	580.28	1.079
VA1	54048	444.58	413.64	1.075
VA1	443576	209.20	192.28	1.088
VA1	636537	175.06	159.48	1.098
VA1	1059566	187.48	169.74	1.105
VA1	1564981	187.48	169.74	1.105
VA2	111200	357.04	328.13	1.088
VA2	885888	161.24	151.18	1.067
VA2	9492762	125.99	112.67	1.118

4.2.6 Case study 5 - square tube-to-plate

This case study addresses the results from an equilibrium equivalent structural stress approach for welded square tube-to-plate connections in bogie frames. Moreover, the article from Yang et al. [66] compares the results with the cataloged reference FAT curves for nominal stresses and with the Master curve for equivalent structural stresses. Finally, the article presents the comparison between the standard values for the Master S-N curve and the derived values from the linear regression of the obtained results.

As the article presents the data in ESS, the results can be directly employed in this dissertation's Master S-N curve alongside the reassessed results from other sources, already presented. Therefore, the results for this case study are presented in Table 4.26.

Table 4.26: Fatigue data for a square tube-to-plate welded connection.

N	ΔS_S [MPa]
166686	487.69
445409	249.41
530865	239.26
642665	258.45
532940	261.53
676091	249.41
765966	229.52
1131354	239.26

4.3 Validation according to existing Master S-N curves

As addressed before, all of the numerical fatigue reassessment data from the case studies would be combined in order to validate the obtained results with the cataloged Master S-N curve according to ASME [14]. Firstly, this standard's curve is often presented with the mean curve and two boundary curves that are distanced from the mean by certain number of standard deviations (SD). The characteristic values for the definition of this curves are derived from fatigue resistance data correspondent to a series of details evaluated according to the equivalent structural stress approach. Consequently, the resultant characteristic values from this assessment are presented in Table 4.27.

Table 4.27: Master S-N curve parameters according to ASME BPVC Sec. VIII Div. 2 [14].

Statistical basis	C	h
Mean	19,930.2	-0.3195
+2 SD	28,626.5	
-2 SD	13,875.8	
+3 SD	34,308.1	
-3 SD	11,577.9	

From this values, each of the reference curves can be traced around the mean Master curve. Applying the selected reference curves and the fatigue reassessment data, the Master S-N curve with the points from the presented case studies is shown in Figure 4.32. Moreover, the labeling of each data set represents the case study's number followed by the main author's name from the respective source and the characterization of the detail. The last one is done by a letter, *B*, *T* or *C* for butt, transverse and cruciform joints respectively, followed by the thickness of the specimens in millimeters. This redundant

information relative to the case study number has the sole purpose of simplifying the interpretation of the Master curve following up on the presentation of the case studies.

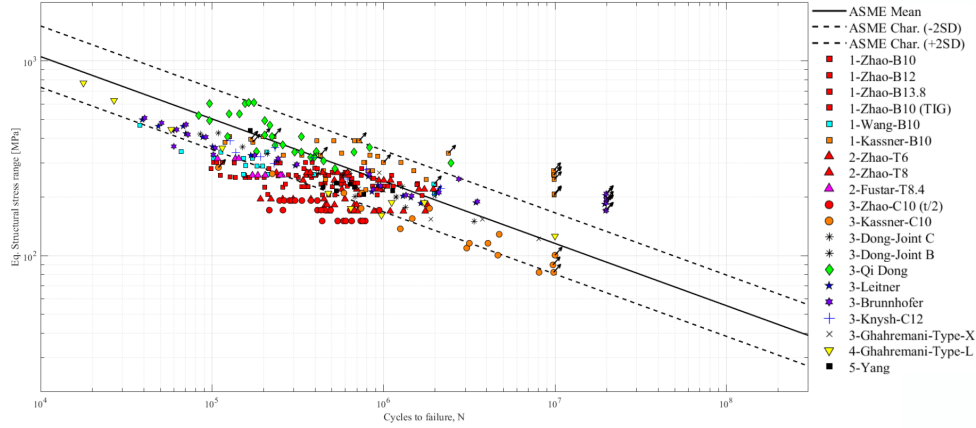


Figure 4.32: Comparison between the numerical fatigue reassessment data and the cataloged Master S-N curve.

From the analysis of the previous figure, it is possible to observe that some data sets fall below the inferior boundary established by the mean curve minus 2 times the standard deviation of the cataloged data in ASME [14]. The data considered by the standard included different types of welding techniques, different detail categories and different thicknesses inside each detail category. However, not all the articles presented in this dissertation present a thorough analysis of the weld quality. In fact, the article Zhao et al. [56], which refers to welded connections in metro bogies, does not include any mention of the weld quality of the specimens. On the other hand, articles like Kassner et al. [57] state that the welding quality is mostly level B, with some details falling in quality level C. Therefore, taking the data from this latter article as the benchmark for comparison, the quality level of the welded connections presented in Zhao et al. [56] must be equal to level C or lower, considering the presented results. Another aspect that supports this conclusion is the fact that the nominal stress data for this article is lower when compared with other sources, for the same details.

Furthermore, in the span of 3 different case studies, for different details, the data from this article falls lower on the S-N curve when compared to the others, suggesting a differentiation factor. As the weld quality is not addressed in any way and considering that this factor plays an important part in the fatigue resistance of the joint, it is possible to infer that it might be the reason for the deviation of the results. Finally, this is one of the

articles that have a FEM model for each detail, allowing for the fatigue reassessment to be made through the implementation of the macro script and, consequently, confirming that the reassessment was carried out according to the established methodology. Therefore, this only leaves the original nominal stress data as an analysis focus for the deviation, which is dependent on the tested specimens.

In order to account for this inconsistency across all the reassessed results, a design curve was considered for two scenarios: considering the data from Zhao et al. [56] and considering only the other sources. These two engineering design curves were traced considering that all the failure data fall on or above the design curve. This approach was chosen over a statistical linear regression due to time constraints in the development of this dissertation. Furthermore, the obtained curves had a value of equivalent structural stress range of 82 MPa and 109 MPa at 2 million cycles for the global and restricted design curves, respectively. The design curves have slopes of $m = 3$ until 10 million cycles and $m = 5$ from that point onward. The relation between the two slopes is $m_2 = 2m_1 - 1$ due to the low number of points in very high cycle fatigue domain and, therefore, not enough data to conclude that there is no plateau after the 10 million cycles mark.

Comparing the obtained values with the lower bound of the ASME Master curve for $2SD$, it is possible to state that the fatigue reassessment results in this dissertation fall below this bound, which presents a value of 133 MPa at 2 million cycles. Finally, Figure 4.33 presents the Master curve obtained in the scope of this dissertation, with the considered design curves and the cataloged ASME reference curves. In Figure 4.34, the same data is presented considering all the sources except Zhao et al. [56].

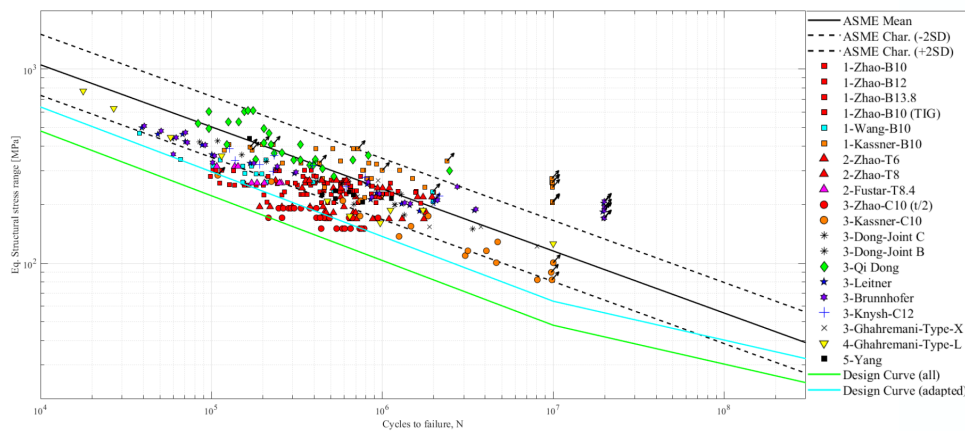


Figure 4.33: Master S-N curve for all the fatigue reassessed data.

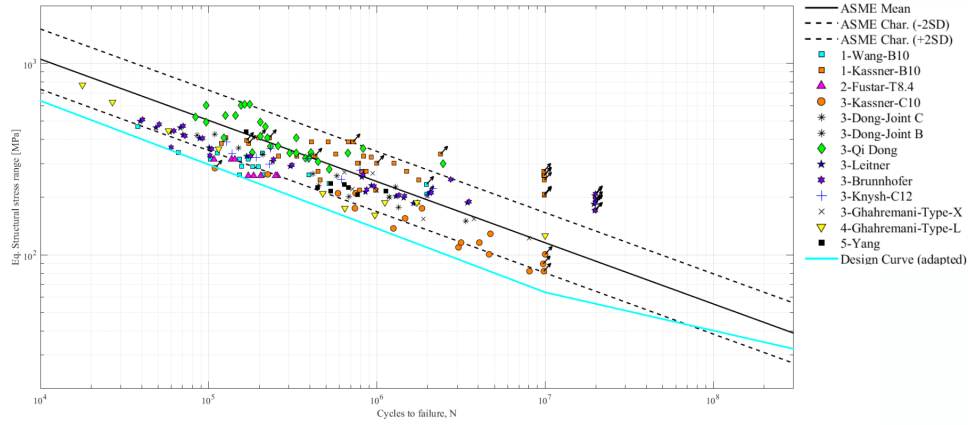


Figure 4.34: Master S-N curve disregarding the results from Zhao et al. [56].

In terms of the model validation of the approximations carried out in this chapter, all of the results from the implementation of the macro script in models developed for each relevant article reveal that the approximations obtained similar results to the model implementation reassessment results. Therefore, it is possible to conclude that these approximations can be considered for welded connections where the membrane stresses are significantly higher than the bending stresses and the latter are approximately null. Furthermore, the implementation of the models also revealed that the values of the structural (hot-spot) stress along the weld line are lower than the ones obtained and available in the literature for the same detail [65], hinting that the determined design curves in this dissertation may present lower reference fatigue resistance values at 2 million cycles than expected.

Finally, the presented results hint that the welding quality level of the specimens may influence significantly the Master S-N curve based fatigue reassessment. As observed by the disregarded article, specimens that do not have quality level specification lead to results that are distanced from the rest of the Master curve in relevant amounts, for 3 different details and several different thickness values in total. Furthermore, one of the main advantages of the use of the Master curve is its ability to collapse all the data from different details into one single S-N curve instead of using one curve for each category of details. Therefore, as the welding quality level of the referred article is not addressed, it is possible to infer that it was the factor that led to lower nominal S-N data and, consequently, resulted in points that did not collapse near the Master curve. In conclusion, the welding quality level influences the Master S-N curve based approach in a significant way.

4.4 Concluding remarks

In this chapter, the fatigue reassessment results from the literature were addressed. The results were compiled into 5 different case studies, with each one presenting a different type of welded connection. For the numerical reassessment, two different strategies were adopted. The first one consisted in implementing a macro script into a finite element model in Ansys Workbench and comparing the results with the analytical calculations, from the structural stress components obtained from the script. The second approach consisted in the approximation of the polynomial expression for $I(r_b)$ and the similar values for the nominal and structural stresses for the same welded connection with small thickness value variations. The two approaches were presented in detail and the second was validated comparing the results with developed FEM models.

In the first case study, butt welds were presented with nominal fatigue resistance data from the literature. It is relevant to emphasize that the results presented in this case study are all referent to welded connections in railway applications. Furthermore, it was observed that the uniaxial nature of the loading conditions resulted in similar values for Mode I and Mode I+III. For that matter, it was possible to address data for butt welds with different thickness without the implementation of the FEM model methodology.

For the second case study, T-joints were addressed. The results also verified that Mode III had lower order of magnitude relative to Mode I, consequently leading to Mode I and Mode I+III to be of similar value.

Case study 3 addressed cruciform joints from the literature considering 3 types of analysis through the implementation of the macro script. The first approach was the one employed for the other case studies, considering the full thickness of the specimen. The second approach consisted in the consideration of half the thickness in the input arguments of the script. Finally, the third approach consisted in the development of a new model that represented a symmetric representation of the joint, with a symmetry plane in the middle of the original thickness. The results from the first and third approaches led to similar values, with the bending stress component much lower than the membrane stresses, for the calculated structural stress. The second approach proved to consider the bending stresses, with these values presenting a similar order of magnitude when compared with the membrane and structural stresses. This was possible because analyzing only half the thickness suppresses the symmetry effects over the bending stresses results and, therefore, increases its value in the analysis, culminating in a higher equivalent structural stress value.

Case study 4 and 5 presented a lap-joint and a square tube-to-plate connection, respectively. For the former, a model was developed considering the as-welded state of the

material and respective failure mode at the weld toe. These results were compared with a similar approach to the one made in the previous case studies where an approximation that disregards the bending stress values were carried out. For the latter case study, the results from the literature were already presented in equilibrium equivalent structural stresses, therefore, further treatment was not needed.

A model verification was also conducted for some relevant details, where it was observed that the approximations considered were close to the numerical reassessment results from the implementation of the macro script in FEM models. Some structural stress results from this validation revealed to be lower than the ones presented in the respective source for the same detail and loading conditions, revealing that the reassessed data from the model implementation may be higher and, consequently, the design curves may present closer reference values to the ASME [14] standard Master S-N curve. For some articles, mainly in case study 3 where cruciform joints were addressed, only the approximation was carried out as the validation through the FEM model revealed that the approximations were conservative for both load-carrying and non-load-carrying joints. This happens as the bending stresses were considered equal to 0 in the approximations and from the application of the analysis for half thickness in FEM models it was verified that this values were relevant, therefore increasing the equivalent structural stresses of the datasets.

Finally, the results were compiled into one Master S-N curve and two engineering design curves were considered, encompassing all the fatigue data. These curves were not determined through statistical linear regression, therefore, according to the applicable standards, the value of the design curves at 2 million cycles can be higher. Furthermore, it was observed that the data fell below the lower bound of the Master S-N curve cataloged in ASME [14], with a reference value at 2 million cycles of 133 MPa. Considering all the data, the design curve had a reference value of 82 MPa. However, disregarding the article Zhao et al. [56] due to a possible lower weld quality, the reference value was then 109 MPa at 2 million cycles. This improvement suggests that a weld quality level equal or under level C may present challenges when applying a Master S-N curve based approach.

Chapter 5

Analysis of a Full-Scale Freight Railway Wagon

This chapter presents the SmartWagons case study and the fatigue reassessment of a critical welded joint located within the Sggrs(s) 80 wagon platform, made out of S355J2+N structural steel. This case study is based on the work presented in [67], implementing the Master S-N curve derived in Chapter 4. The wagon under analysis has 3 bogies, located at each end of the platform and at the Talbot articulation. The total length of each wagon is 12.220 m, with a width of 2.970 m, as presented in Figure 5.1.

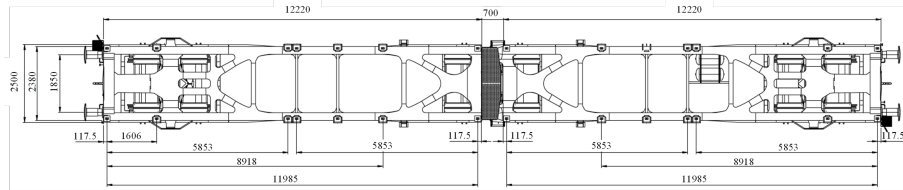


Figure 5.1: Technical drawings of the Sggrs(s) 80' freight wagon top view, dimensions in mm (adapted from Silva et al. [67]).

The FEM model shown in Figure 5.2 represents half of the Sggrs(s) 80 wagon platform that was addressed in this article. It should be noted that Figure 5.2 serves as an illustrative geometric representation of the wagon platform model used in the analysis.

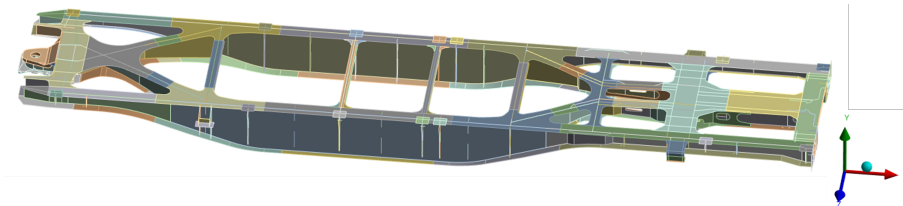


Figure 5.2: Global FE model of the Sggrs(s) 80 wagon platform in Ansys.

The analysis is developed within a global-local multiscale framework, in which a global shell finite element model is first employed to identify the most fatigue-sensitive location, according to EN 12663-2 [21] loading conditions. Considering the global analysis results, a local solid submodel is then used to reassess the selected welded connection with better precision, enforcing the global results as boundary conditions. This approach enables the evaluation of the structural response at different length scales, linking the global load transfer mechanism to the local stress concentration phenomena that govern fatigue damage in welded joints.

For that matter, the reassessment of the critical welded joint is addressed in this chapter, alongside the employed FEM models and respective results. Furthermore, the results are analyzed by comparing the obtained values with cataloged reference values, allowing for the definition of the utilization factor according to ASME [14] and DVS 1612 [18]. Finally, the Master S-N curve obtained from the fatigue-reassessed data presented in Chapter 4 is compared with these standards and with the results from the freight wagon.

5.1 Fatigue assessment of a critical welded joint

The fatigue assessment carried out in this article focuses on the welded detail that exhibited the highest utilization factor in the global nominal stress-based verification. The selected location is particularly relevant as it lies in a region of pronounced load transfer, where the combination of vertical and transverse actions produces a local stress state that is not fully captured by nominal stress values alone. For this reason, the detail is reassessed through a local stress-based methodology, with special attention given to the weld toe region, the structural stress distribution and the fatigue-related mechanisms associated with the joint geometry, through structural hot-spot stresses (SHSS) and effective equivalent structural stresses (EESS) [67].

Firstly, the global FE model is used to assess the behavior of all the welded details and to determine the governing load case, according to the loading scenarios contemplated in EN 12663-2 [21]. Subsequently, it was possible to observe where the critical joint lay, which was then isolated and represented in a local solid model. This model allowed for a more refined mesh density in the vicinity of the weld toe and the use of quadratic solid elements "consistent with a fine-mesh strategy for SHSS extraction according to IIW Recommendations" [17]. This is a relevant aspect to consider for the SHSS approach, whereas for the Master S-N curve-based approach there is a relative mesh insensitivity, where the main advantage of the latter method lies. The boundary displacement conditions were transferred from the global analysis solution to the local submodel through the *ANSYS Imported Cut Boundary Constraint* functionality, which imposes the global displacement

field result as boundary conditions in the local submodel. This methodology ensures kinematic compatibility and preserves the load transfer mechanism observed at the global scale, while still presenting an effective trade-off between the fidelity of the model and the computational cost. Finally, the local model was used to compute structural (hot-spot) stress and equivalent structural stress, alongside the corresponding fatigue utilization factors, which are compared with the derived Master S-N curve.

In terms of numerical assessment, considering that the EN 12663-2 [21] standard only contemplates fatigue loading as a quasi-static load case, the utilization factor for the local assessment was defined within a safe-life framework, with respective expression presented in:

$$U_f = \frac{\Delta\sigma_{ED}}{\Delta\sigma_{RD}} \quad (5.1)$$

where $\Delta\sigma_{ED}$ represents the numerical evaluated stress range and $\Delta\sigma_{RD}$ represents the corresponding strength stress range. In terms of the utilization factor, $U_f \leq 1$ indicates that the utilization factor is in compliance with the resistance curve.

5.2 Results

After applying the already addressed methodology in the FE models and considering the maximum stress range obtained for the SHSS and for EESS, Silva et al. [67] presents each utilization factor for different considerations. In fact, the analysis comprised the calculation of the utilization factor for nominal stresses, for both shell and solid models, structural hot-spot stresses and equivalent structural stresses, both for the solid model. The results from the different analyses are presented in Table 5.1. For all calculations, a value of $\gamma = 1.15$ was considered as a strength reduction factor, which takes into consideration the safe-life approach, the inspection levels and the consequences of failure.

Table 5.1: Maximum global and local structural stress range values, and corresponding utilization factors (adapted from Silva et al. [67]).

Approach	$\Delta\sigma_n$ (Shell)	$\Delta\sigma_n$ (Solid)	$\Delta\sigma_S$	ΔS_S Modes I+III
$\Delta\sigma_{max}$ [MPa]	70.24	64.18	91.06	92.97
U_f	0.93	0.84	1.05	0.80 / 0.70

In the previous table, it is possible to observe that the utilization factor's value is lower for the EESS based approach. The values for this approach are compared with the standardized Master S-N curve and the different results for the utilization factor represent the comparison with the Master S-N curve from ASME [14] and the Master S-N curve derived by Alencar et al. [15] for bridge details. Moreover, Silva et al. [67] also presents the respective survival probability for each of the methods considered.

This dissertation did not consider a statistical linear regression of the data points in the Master S-N curve, where the distribution and survival probability would be taken into account. Instead, the design curves were determined considering a conservative approach that encompassed all the data from the fatigue reassessment methodology through a lower-bound envelope approach. Therefore, the obtained results have a conservative framework already accounted for, allowing for the possibility of considering the strength reduction factor $\gamma = 1.0$. Consequently, the design fatigue resistance curves of this dissertation were directly compared with the maximum stress range value of the equivalent structural stress for Mode I and Modes I+III. As shown in Table 5.2, where the utilization factors are presented, it is possible to observe that in the critical welded connection of the freight wagon, Mode I and Modes I+III have similar values. This behavior has been observed over the majority of the addressed welded connections, as expressed in the previous chapter.

Table 5.2: Maximum EESS range values and respective utilization factor for each of the considered design curves.

Design Curves	All data		Without Zhao et al. [56]	
	Mode I	Modes I+III	Mode I	Modes I+III
$\Delta\sigma$ [MPa]	92.838	92.967	92.838	92.967
$\Delta\sigma_c$ [MPa]	82	82	109	109
γ	1.0	1.0	1.0	1.0
U_f	1.132	1.132	0.852	0.853

From these results, it is possible to conclude that the critical welded joint analyzed in this case study would not be in compliance with the design curve that considers all the data from the previous case studies. However, when disregarding the fatigue reassessment data from Zhao et al. [56], the adapted design curve returns a utilization factor equal to 0.853 for the combined Modes I+III. This value is still higher than the one achieved by the source when comparing with the ASME [14] reference curve, but the project conclusion would now be that the critical welded connection in the freight wagon is in compliance with the standards. Furthermore, the strength reduction factor $\gamma = 1.0$ combined with the methodology employed in this dissertation results in a value for the utilization factor

6.62% higher than the one achieved by Silva et al. [67]. Therefore, it is possible to conclude that the adapted design curve derived from the fatigue reassessment data from the previously addressed case studies returns a similar value for the utilization factor of the critical welded connection in a freight wagon when compared with the ASME standard [14].

Overall, based on the results of the presented case studies, it is possible to state that, even when disregarding data from a railway fatigue resistance article that presents a deviation from the majority, the achieved design curve still presents lower fatigue resistance values when compared with the standards that employ a Master S-N curve-based approach. Although the design curves of this dissertation did not consider a statistical linear regression of the data, the utilization factors for this latter case study show that they are more conservative than the ASME [14] and Alencar et al. [15] Master curves. Nevertheless, in order to account for the engineering approach of the design curves, a strength reduction factor $\gamma = 1.0$ was applied and the results presented a low difference when compared with the cataloged Master S-N curves. However, for the ASME [14] standard, the Master S-N curve is derived from different welded details employed in boilers and pressure vessels and for Alencar et al. [15] the Master curve was derived from welded bridge details. This aspect, allied to the different survival probabilities considered in each assessment, 97.5% and 95% respectively, may justify certain deviations from the final results.

For this dissertation, the anchor points of the design curve, defined as the lowest data points that would anchor the final design curve, are from literature that considers railway vehicles welded connections. This may hint at a possibility of railway welded joints presenting a different fatigue behavior, when compared to other welded joints, that may influence the final Master S-N curve-based analyses. In contrast, the final reference value for the considered case studies, at 2 million cycles, was 109 MPa, and the ASME [14] and Alencar et al. [15] reference values are 133 MPa and 152.9 MPa, respectively. This difference may not be significantly large as to consider that railway vehicle welded connections diverge from the Master S-N curves already addressed for pressure vessels and bridges.

However, considering the factor $\gamma = 1.15$ and comparing the results with the cataloged lower bound from the ASME standard [14], the result is similar to the result obtained for the adapted design curve that disregards Zhao et al. [56]. In fact, at 2 million cycles:

$$\frac{133}{1.15} = 115.65 \text{ MPa} \quad (5.2)$$

This value is similar to the 109 MPa reference value obtained from the fatigue reassessment in this dissertation. Therefore, these results hint that if a statistical treatment was carried out for the considered data, disregarding the Zhao et al. [56] dataset, the fatigue resistance of the design curve may be close to the standards.

5.3 Concluding Remarks

This chapter addressed the multiscale fatigue reassessment of a critical welded joint within an Sggrs(s) 80 railway freight wagon by transitioning from a macro-scale structural model to a refined local submodel [67]. This global-local multiscale framework proved to be a computationally efficient and high-fidelity methodology, capturing the complex local stress fields that govern fatigue damage, which are typically underestimated by conventional nominal stress methods. Specifically, the global shell model successfully isolated the critical fatigue-sensitive details under EN 12663-2 [21] loading conditions, while the local solid submodel accurately resolved the steep spatial stress gradients at the weld toe, ensuring a robust representation of the structural response without prohibitive computational costs.

Regarding the numerical assessment, the structural hot-spot stress (SHSS) and effective equivalent structural stress (EESS) approaches presented distinct utilization factors, U_f . Notably, when applying the conservative design curves derived in this dissertation based on the complete literature dataset, the critical joint exhibited a utilization factor greater than unity, $U_f = 1.132$, indicating non-compliance with the safe-life design criteria. However, when the statistical outlier data from Zhao et al. [56] were excluded from the master dataset, the adapted design curve presented a compliant utilization factor of $U_f = 0.853$ for combined Modes I+III. This updated result is highly consistent with the commercial ASME BPVC standard code framework [14], exhibiting a deviation of only 6.62% and demonstrating the high sensitivity of the structural verification route to the experimental data filtering process.

Finally, the comparative analysis indicated that the lower reference fatigue strength at 2 million cycles, 109 MPa, derived from railway-specific welded joints data, lies below the reference values of the Master S-N curves proposed by ASME [14] and Alencar et al. [15], which are 133 MPa and 152.9 MPa, respectively. Nevertheless, after accounting for the strength reduction factor, namely $\gamma = 1.15$ for the cataloged curves and $\gamma = 1.0$ in the present work, the adjusted lower reference strengths become 115.65 MPa and 132.97 MPa, respectively. Therefore, these corrected values are similar to the result obtained in this dissertation.

Chapter 6

Conclusions and Future Work

This final chapter synthesizes the primary contributions, methodologies and findings established throughout this dissertation, which addressed the advanced fatigue life analysis of welded details in steel railway freight wagons. This overview provides a reflection on the structural, normative and numerical insights gained across the entire research work, concluding with strategic recommendations for future research developments in the field.

6.1 Conclusions

A comprehensive review of the current state of the art and regulatory frameworks highlighted a significant fragmentation in industrial practices. Standard design codes, such as EN 12663-2 [21] for wagon bodies and load cases, EN 1993-1-9 [30] Eurocode framework and the specialized German code DVS 1612 [18], rely heavily on nominal or hot-spot stress concepts. While these conventional routes ensure standard safety margins, they introduce high subjectivity during joint classification and also present strong dependence on finite element mesh density. This limitation can lead to complications in the maintenance and management of the fleet for the weight reduction of the freight wagons. In contrast, the equilibrium equivalent structural stress approach, driven by the extraction of nodal forces and moments along the weld line, demonstrated mesh insensitivity and a superior capability to collapse scattered fatigue data from distinct welded details into a single fatigue resistance curve, i.e., the Master S-N curve.

This methodology was applied through a customized post-processor, developed by Alencar et al. [55], integrated within Ansys Workbench to extract local structural stresses across various weld configurations in S355 steel, including butt, T-joints, cruciform, lap and square tube-to-plate connections. The subsequent reassessment of literature datasets containing experimental fatigue tests run until complete specimen failure revealed critical insights into experimental data filtering. Within the analyzed data, an outlier dataset,

from Zhao et al. [56], lacked explicit welding quality definitions, such as those prescribed by ISO 5817 [59], causing a major drop in the compiled fatigue envelope. Therefore, these results present the necessity of further and more advanced analyses. Disregarding this dataset allowed for the derivation of a refined design curve with a reference strength of 109 MPa at 2 million cycles. This threshold is noticeably lower than the general curves from ASME [14], 133 MPa, and Alencar et al. [15], 152.9 MPa, which are based on pressure vessels and bridges, respectively. Considering the fact that this dissertation did not consider a statistical assessment of the data and the strength reduction factor γ was considered equal to 1.0, instead of the 1.15 addressed in the standards, the obtained reference value at 2 million cycles of this thesis is similar to the ASME standard [14]. In fact, 109 MPa is close to $133/1.15 = 115.65$ MPa and still presents a considerable difference when compared with the Master S-N curve developed by Alencar et al. [15], $152.9/1.15 = 132.96$ MPa. Although the results of this thesis align closely with pressure vessel standards rather than welded bridge details, it is crucial to emphasize that the underlying analyses strictly targeted welded connections within railway vehicle applications. Consequently, providing validation for railway applications represents the primary scientific contribution of this work.

Finally, the global-local multiscale finite element framework was analyzed through the structural reassessment of a critical welded joint on the Sggrs(s)80 articulated freight wagon, from the SmartWagons project, under the quasi-static load combinations enforced by EN 12663-2 [21]. The global shell model acted as an efficient tool to identify structural vulnerabilities, employing a classic nominal stress-based approach. Then, the local solid submodel resolved the stress gradients at the weld toe through boundary conditions that complied with the global analysis results. The fatigue utilization factors proved to be highly sensitive to the chosen design curve, as the unrefined master dataset resulted in non-compliance, $U_f = 1.132$, whereas the adapted design curve showed a safe and fully compliant value, $U_f = 0.853$. This compliant result demonstrates excellent alignment with the established commercial ASME [14] framework, with a deviation of only 6.62%, confirming that the EESS methodology coupled with rigorous data filtering provides an accurate, robust and industry-ready verification route for structural railway engineering.

In summary, the equivalent structural stress-based approaches and the global-local multiscale finite element framework allow for a better representation of the load transfer mechanism in large-scale models and enable the subsequent detailed analysis of a critical welded connection. The correct representation of these mechanisms and the highly detailed local analysis lead to more realistic predictions, which consequently result in more adequate maintenance management of the rolling stock. Finally, these aspects contribute to the optimization of the life cycles of the railway vehicles, which is an important sustainability goal.

6.2 Future Work

Based on the limitations identified and the achievements reached in this dissertation, several strategic paths for future research are proposed. Firstly, the deterministic envelope approach used to construct the custom design S-N curves should be advanced towards a full probabilistic framework. This can be achieved by applying a formalized statistical linear regression analysis to the filtered literature dataset. This methodology will allow for the explicit calculation of survival probabilities, perfectly matching the statistical safety formats of international structural codes. Furthermore, targeted experimental fatigue testing of standard S355 welded specimens and high-strength steel alternatives should be performed locally to expand the current database and directly calibrate the Master curve slope parameters.

Secondly, it is relevant to compile a welded joints database for railway vehicles, where the welding quality level and other relevant considerations should be presented. This step will ultimately allow for a better assessment of the behavior of welded connections in railway vehicles by applying the Master S-N curve approach and validating the hypothesis of different behaviors for these types of joints. Moreover, the database should also be increased with more cases of high multiaxiality loading.

Lastly, an evolution should be made regarding the nature of operational loading cases. While the quasi-static load scenarios defined in EN 12663-2 [21] are ideal for standard compliance checks, they cannot capture the transient and high-frequency dynamic interactions that take place in the vehicle during regular operations. The resulting transient time-history force vectors at the body-to-bogie interfaces can be mapped directly into the global-local finite element submodel. This allows for the extraction of dynamic structural stress histories to be processed using a rainflow cycle counting algorithm and Palmgren-Miner cumulative damage rules, consequently shifting the current safe-life approach into a highly realistic damage prognosis framework. These analyses should be carried out considering S355 as it is one of the most commonly used structural steels for freight wagons and repeated for a high strength steel, such as S690, in order to assess its applicability and advantages.

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