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A Composite Multi-Body Dynamics and Finite Element Framework for Offshore Wind Turbine Fatigue Assessment

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Abstract

The race to replace fossil fuels with renewable energy sources has intensified in response to the escalating threats of climate change. This urgency has driven unprecedented growth in the wind energy sector, leading to significant advancements in wind turbine technology. Over the past two decades, turbines have dramatically increased in both size and capacity, as the industry strives to maximise energy capture and position wind energy as a viable competitor to conventional fossil fuel-based generation. The development of offshore wind turbines has enabled access to the rich aeolian resources of the open sea, where higher wind speeds and improved airflow quality enhance power production. However, the increased scale and offshore location of modern, multi-megawatt wind turbines introduce complex structural and operational challenges. The growing length of turbine blades and corresponding tower heights has led to increasingly slender and flexible structures. These large-scale systems experience highly dynamic responses under the elevated aerodynamic and hydrodynamic loads characteristic of offshore environments. The rotating mass of the turbine blades, in an atmospheric boundary layer characterised by wind shear, further contributes to the operational demands of these systems, imposing substantial cyclic, gravitational forces on key structural components. Consequently, modern offshore wind turbines are subjected to intensified cyclic loading, often exceeding 100 million load cycles over a 20-year design life, while operating in the corrosive conditions of the marine environment. The combination of these factors makes wind turbines and their components highly susceptible to fatigue damage, directly impacting their operational lifespan. Accurately quantifying this damage is therefore essential for the effective design and analysis of offshore wind turbines. It enables the optimisation of maintenance schedules and end-of-life strategies, while also improving the accuracy of Levelised Cost of Energy (LCOE) calculations, ensuring both structural safety and economic viability.

Recognising the critical importance of accurately quantifying fatigue damage, in this work a composite modelling framework is developed to perform an efficient and high-fidelity fatigue analysis of offshore wind turbines. The composite modelling framework employs a blended approach, integrating a multibody dynamic (MBD) model and a detailed finite element (FE) model, leveraging the strengths of both approaches to deliver an efficient and accurate fatigue analysis of the IEA 15-MW monopile-based offshore wind turbine. The nonlinear MBD model, developed previously within the research group, is fully coupled with 22 degrees of freedom (DOFs) and formulated using Kane's dynamics. This model captures the global dynamic response of the wind turbine under operational conditions, considering the effects of both hydrodynamic and aerodynamic loading. However, while the MBD model provides accurate system-level dynamics, its reliance on simplified geometric representations limits its ability to resolve localised stresses in critical turbine components such as bolts, welds, or flanges. To address this limitation, the resulting nominal stress time histories from the MBD model are enhanced by incorporating stress concentration factors (SCFs) derived from a high-fidelity FE model, enabling accurate consideration of local geometric effects.

Employing this composite model framework, a fatigue analysis was performed considering real metocean conditions defined by the National Renewable Energy Laboratory (NREL) for a site off the east coast of the United States. The analysis focuses on a butt weld located at the base of the tower, a known fatigue-critical detail, with industry failures documenting its susceptibility to crack initiation. This butt weld connects the tower shell to an internally bolted ring flange which serves as the interface between the tower and the transition piece. By considering the stiffening effect of the internally bolted ring flange connection through the use of SCFs, the fatigue life of the butt weld was observed to decrease by 19.3% in comparison with traditional approaches which neglect such effects. This highlights the significant impact of local structural features on fatigue performance. Failing to account for such features can lead to non-conservative fatigue life estimates, increasing the risk of premature failures in wind turbine towers giving rise to both structural safety concerns and substantial financial losses.

The fatigue analysis conducted was further extended to include the effects of the corrosive marine environment, reflecting material degradation over time. The parametric capabilities of the developed FE model were employed to capture the

effects of material degradation caused by the corrosive marine environment, generating time-varying SCFs. By combining these SCFs with the nominal stress time histories and appropriate S–N curves, the model enhances fatigue damage predictions at the butt weld, accounting for scenarios in which corrosion protection measures have failed. When assuming a typical corrosion protection system that remains effective for the first five years of operation, the predicted fatigue life of the butt weld was reduced to 13.39 years. This fatigue life falls significantly short of the intended 20-year lifespan illustrating the vital importance of considering the effects of corrosion wastage on the lifespan of such critical components. Given that inspection and repair of these welds on offshore structures are both prohibitively expensive and operationally complex, accurately predicting their fatigue performance is essential to avoid unplanned maintenance, reduce operational risk, and ensure long-term structural reliability. The vulnerability of these welds to the corrosive marine environment was further demonstrated using the composite modelling framework to perform fatigue analyses for two sites off the coast of Ireland using environmental conditions obtained from the Irish Marine Data Buoy Observation Network (IMD-BON). Under the same corrosion protection system, with failure assumed after five years, the fatigue lives were estimated to be 24.75 and 13.38 years for the east and west coast sites respectively demonstrating significant reductions as a result of the material wastage induced by corrosion. Additionally, these results reveal substantial variations in structural performance between the two sites, emphasising the necessity of site-specific design considerations.

These findings collectively demonstrate the turbine’s vulnerability to fatigue loading and underscore the need to effectively reduce fatigue demands on structural components, particularly in harsh offshore environments. To address the limited lifespans observed, vibration control devices, including a Tuned Mass Damper (TMD) and Tuned Mass Damper Inerter (TMDI), were installed at the top of the wind turbine tower. While both devices demonstrate significant vibration mitigation capabilities, their overall effectiveness in reducing fatigue damage is limited due to adverse interactions with non-targeted structural modes. As a result, neither the TMD nor the TMDI provided a significant improvement with regard to the fatigue performance for the offshore wind turbine tower.

The findings of this thesis highlights the importance of integrated modelling approaches that account for both global dynamics and local stress concentrations,

while also incorporating environmental degradation effects. The results emphasise the critical role of detailed fatigue assessment in offshore wind turbine design and the limitations of conventional mitigation strategies, reinforcing the need for more resilient solutions to ensure structural reliability and long-term economic viability.

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Acronyms

3D-PTMD Three-Dimensional Pendulum Tuned Mass Damper.

API American Petroleum Institute.

ATMD Active Tuned Mass Damper.

BEM Blade Element Momentum.

BModes Blade Modes (modal analysis tool from NREL).

BSI British Standards Institution.

C-F Corrosion-Fatigue.

CAPEX Capital Expenditure.

CCATMD Cable-Connected Active Tuned Mass Damper.

CFD Computational Fluid Dynamics.

CHS Circular Hollow Section.

DEn Department of Energy.

DLC Design Load Case.

DNV Det Norske Veritas.

DOE Design of Experiments.

DOF Degree of Freedom.

FE Finite Element.

FEA Finite Element Analysis.

FEM Finite Element Method.

FFT Fast Fourier Transform.

FT Fourier Transform.

Acronyms

GPD Generalised Pareto Distribution.

HAWC2 Horizontal Axis Wind turbine Code 2.

Hs Significant Wave Height.

IDVA Inerter-Based Dynamic Vibration Absorber.

IEA International Energy Agency.

IEC International Electrotechnical Commission.

IIW International Institute of Welding.

IMDBON Irish Marine Data Buoy Observation Network.

IRENA International Renewable Energy Agency.

ISO International Organization for Standardization.

ITTC International Towing Tank Conference.

JPD Joint Probability Distribution.

LCOE Levelised Cost of Energy.

LHS Latin Hypercube Sampling.

LQR Linear Quadratic Regulator.

LRF Load Range Factor.

LUlt Ultimate Design Load.

MBD Multibody Dynamics.

MSL Mean Sea Level.

NOAA National Oceanic and Atmospheric Administration.

NREL National Renewable Energy Laboratory.

PDE Partial Differential Equations.

PDF Probability Density Function.

PSD Power Spectral Density.

RAO Response Amplitude Operator.

RIDTMD Rotationally Inertial Damped Tuned Mass Damper.

RMS Root Mean Square.

ROSCO Reference OpenSource Controller.

RWT Reference Wind Turbine.

S-N Curve Stress vs Number of Cycles Curve.

SCF Stress Concentration Factor.

TI Turbulence Intensity.

TLCD Tuned Liquid Column Damper.

TLP Tension Leg Platform.

TMD Tuned Mass Damper.

TMDI Tuned Mass Damper Inerter.

TMIS Tuned Mass Inerter System.

T_p Peak Spectral Wave Period.

TVMD Tuned Viscous Mass Damper.

Chapter 1

Introduction

1.1 Motivation for Research

It is widely recognised that the burning of fossil fuels has led to several severe environmental crises and will continue to do so until they are replaced with clean sources of energy production. The burning of fossil fuels is responsible for many of the critical issues facing our world today, including climate change, rising sea levels, air and water pollution, deforestation, and acid rain. These issues have already had significant impacts on human health and global ecosystems, contributing to the extinction or endangerment of numerous species. If left unaddressed, these consequences will continue to intensify, contributing to irreversible ecological damage and widespread public health crises. Beyond the devastating environmental consequences that arise from the burning of fossil fuels, the natural reserves of these finite resources are rapidly depleting due to the continuously growing demand for energy. The World Coal Association stated in 2016 that at current extraction rates, there are only 110 years worth of coal remaining, while global oil reserves are only expected to survive until 2070. Given both the severe environmental impacts and the accelerating depletion of fossil fuel reserves, it is imperative that the world transitions to sustainable energy sources such as solar, wave and wind energy as quickly as possible [1].

At its core, wind energy harnesses the kinetic energy of moving air to generate electricity. Its key advantages include the virtually inexhaustible potential of wind and its minimal lifecycle greenhouse gas emissions, making it one of the most sustainable options for large-scale energy generation. Although often perceived as a

modern technology put in place in an effort to reduce and eventually replace fossil fuels, wind energy has been battling its way to the forefront of energy production for quite some time now. The first electricity-generating wind turbine was constructed by James Blyth in 1887 [2], marking the beginning of a slow but steady journey toward mainstream energy production. Over the past 20 years, this momentum has accelerated considerably, driven by a strong global push toward the adoption of renewable energies (see Figure 1.1).

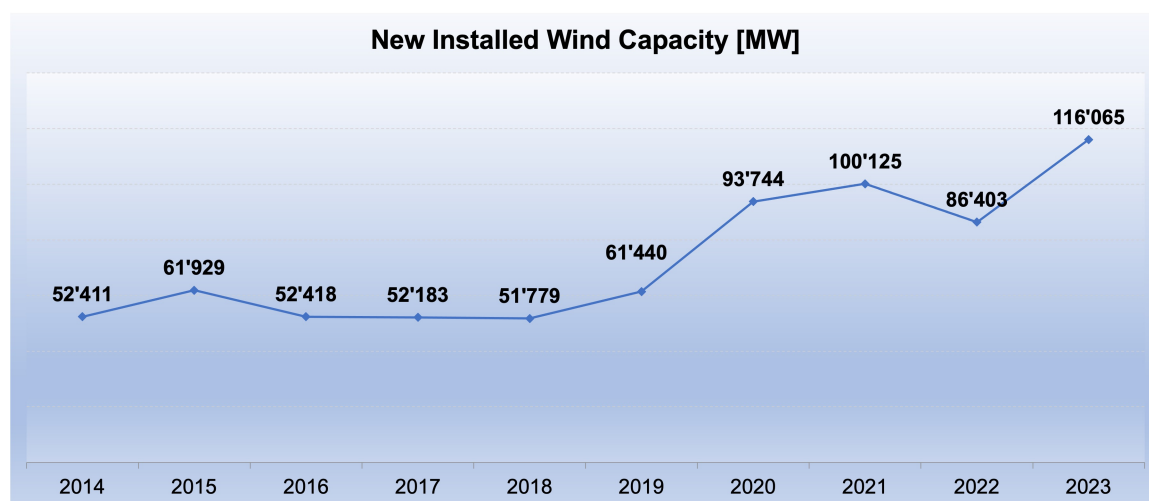


Figure 1.1: Annual newly installed wind energy capacity world wide [3]

To date, the implementation of wind energy has contributed to a reduction of over 1.1 billion tonnes of CO₂ emissions globally and with increased investment, it has the potential to play an even greater role in mitigating climate change. However, the urgency for rapid action is evident. The 2010-2019 period marked the hottest decade on record, and despite a temporary decrease in CO₂ emissions due to COVID-19 travel restrictions in 2020, the years 2023 and 2024 set new records as the hottest years ever recorded. This alarming trend is further emphasised by 2024 marking the first year in which global temperatures averaged more than 1.5°C above pre-industrial levels [4] (see Figure 1.2). To counteract this, the levels of CO₂ emissions being released into the atmosphere must be dramatically reduced. However, at the current rate of renewable energy deployment, carbon neutrality will not be achieved until approximately 2050. To have a realistic chance of meeting the goals set out in the Paris Agreement in 2015, the total global renewable power generation capacity will need to triple by 2030, reaching more than 11,000 GW [5]. For wind energy specifically, installation rates must at least double with an annual increase of 180

GW in wind energy capacity representing the very minimum that must be achieved if the world is to avoid the most detrimental impacts that climate change could bring [6].

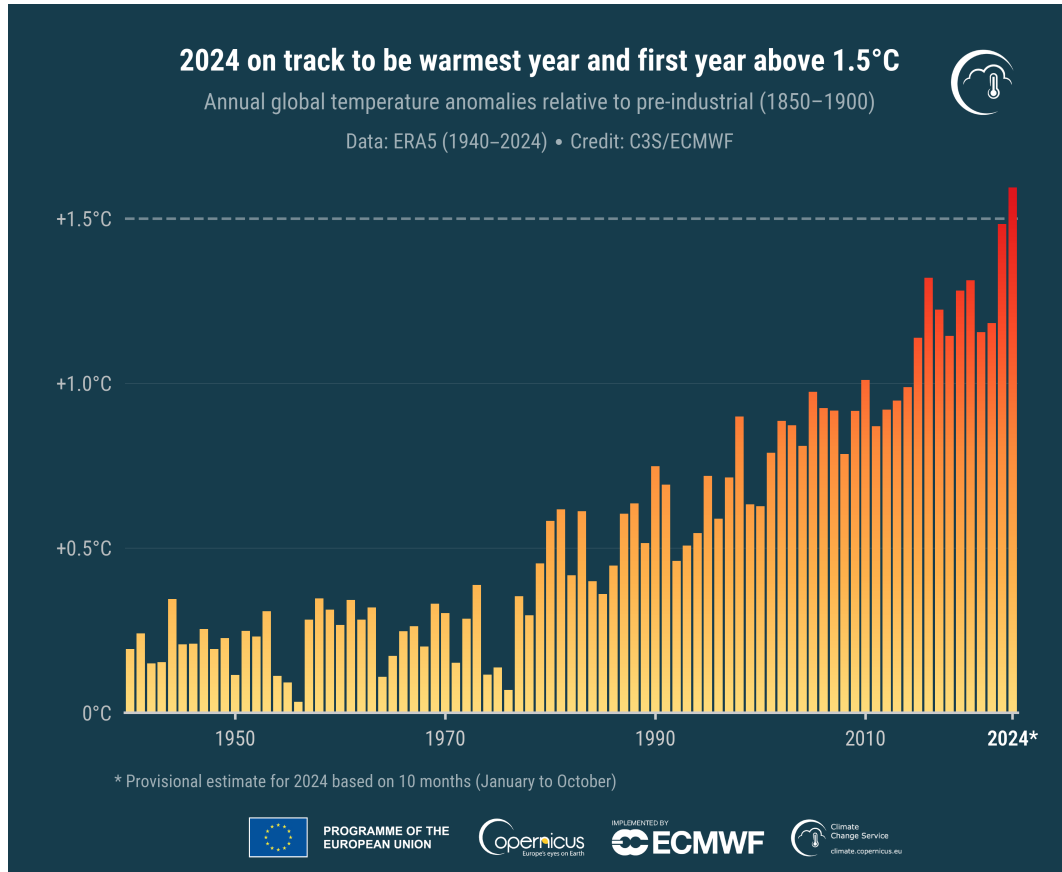


Figure 1.2: Annual global temperature with respect to pre-industrial levels [7]

Despite the devastating impacts that climate change has already caused in intensifying natural disasters - including catastrophic wildfires, unprecedented heat-waves and flooding- the Paris Agreement alone will not compel the world to make this monumental shift from fossil fuels to renewable energy. To effectively compete with traditional fossil fuels, wind energy must not only be sustainable, but it must also become more economically viable. This burden, however, is being eased by recent geopolitical instability and the volatility of fossil fuel prices (see Figure 1.3), both of which are playing a decisive role in accelerating the transition to renewable energy. Between January 2020 and December 2021, the World Bank's energy price index surged by 50%, placing significant strain on global energy markets. This crisis was further intensified by the conflict in Ukraine that began in February 2022, further exacerbating already rising energy prices. Within two weeks of the conflict

starting, oil prices increased worldwide by 25% and by March 2022, gas prices had increased by a staggering 580% compared to March 2021 [8]. The severe impact of geopolitical events leading to such energy crises occurring in recent years has highlighted the importance of energy security worldwide. This has further driven demand for local renewable energy sources, such as wind, to ensure supply security and provide greater protection against volatile global energy markets.

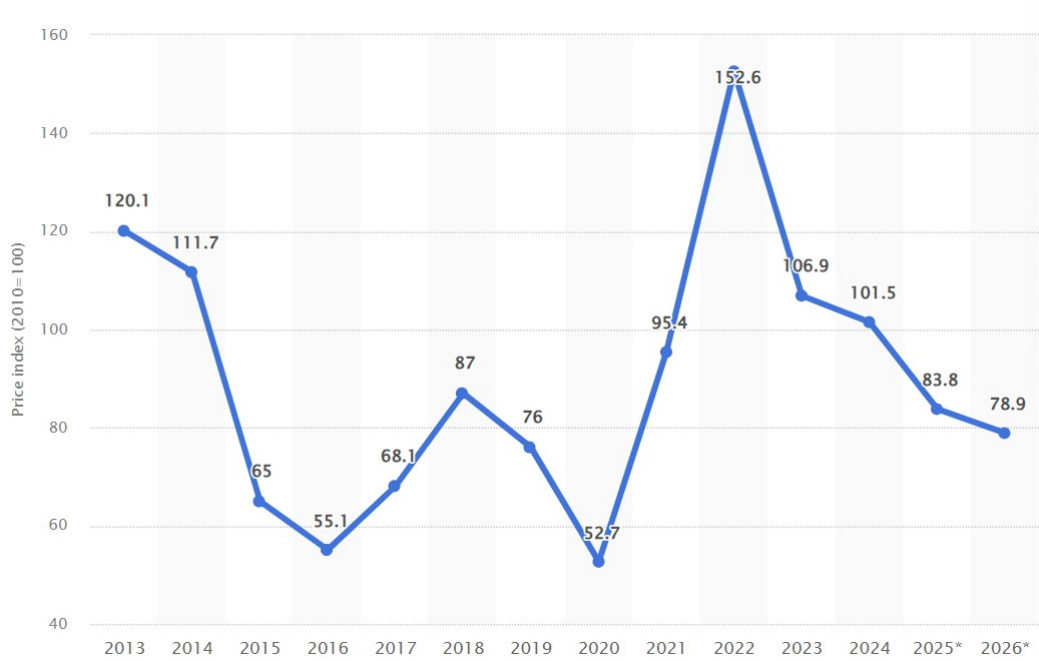


Figure 1.3: Global energy commodity price index from 2013 to 2023 [9]

Conversely, the cost of renewable energy is steadily decreasing. In 2020, it was reported that over 62% of newly installed renewable energy sources had a Levelised Cost Of Energy (LCOE) lower than the cheapest fossil fuels available [10]. This decline has been driven by advancements in construction techniques, economies of scale, and improvements in the performance and reliability of generation systems. The LCOE of offshore wind energy in particular has seen significant reductions since its introduction, with reports suggesting that over the past decade, the LCOE of offshore energy has fallen by as much as 50% [11]. In 2020 alone, the International Renewable Energy Agency (IRENA) reported that the average cost of onshore wind production fell by 13% while offshore production costs fell by 9% [10].

Although these reductions represent significant progress, further cost decreases remain essential to enable the large-scale transformations required to meet

global climate and energy targets. Consequently, in an effort to reduce wind turbine capital expenditure (CAPEX) wind turbine designers are increasingly implementing strategies focused on optimising material use, extending turbine lifespans, and maximising the energy output of individual units. As larger turbine ratings tend to produce higher returns on investment [12], modern wind turbines have undergone a substantial increase in scale, enabling them to capture significantly increased quantities of energy (see Figure 1.4). The power production formula for wind turbines reveals a twofold benefit of scaling up. First, taller towers enable wind turbines to take advantage of Hellmann’s power law [13], allowing the turbines to capture wind with greater velocity, which has a cubic relation to the power produced. Secondly, larger hub heights accommodate greater rotor diameters, which have a squared relation with power production. This has resulted in significant growth in the length of the rotor blades and the heights of turbine towers as depicted in Figure 1.5. Enevoldsen and Xydis [14] detailed the extent of this development, reporting that hub heights for onshore wind turbines doubled from 82m in 1998 to 164m in 2016. In a similar time frame, wind turbine rotor diameters have almost tripled, increasing from 54m in 1998 to 158m in 2017 [14]. By increasing both the hub height and rotor diameter, the energy production capacity of individual wind turbines has significantly risen, allowing modern wind turbines to be constructed with greatly increased capacities, such as Dongfang’s 26-MW wind turbine. This turbine, located in China, with a hub height of 185 meters and a rotor diameter exceeding 310 meters, stands as the largest wind turbine in the world [15]. Wind turbines of similar scale to this are now being deployed globally, making today’s modern multi-megawatt wind turbines the largest rotating structures in the world [16].

Chapter 1. Introduction

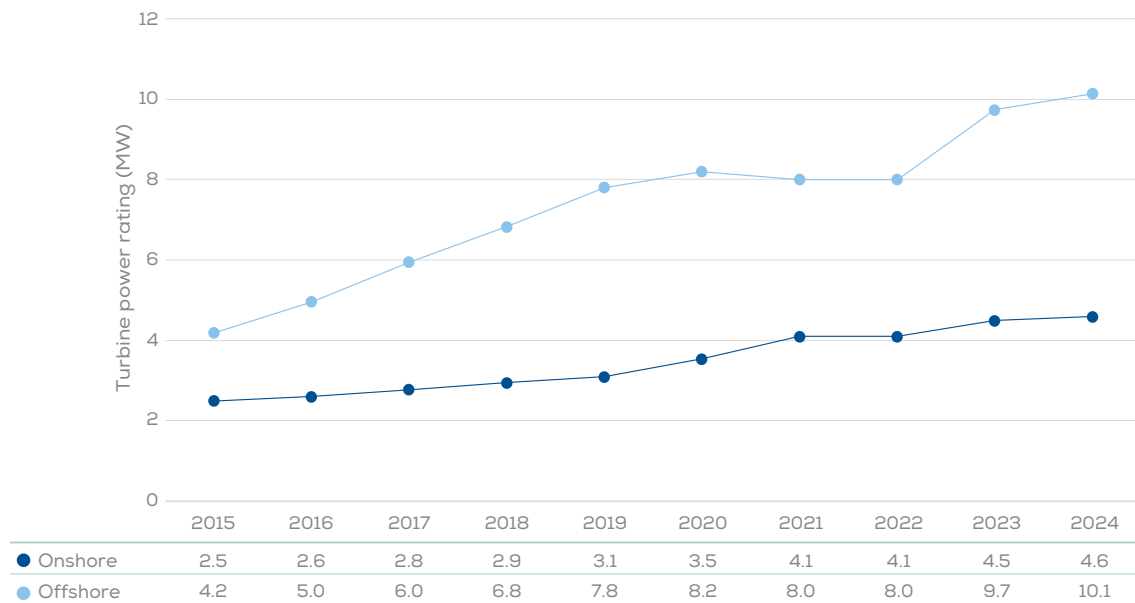


Figure 1.4: Average power rating of onshore and offshore wind turbines installed in Europe [17]

Trend of onshore and offshore turbine size, 1980-2030

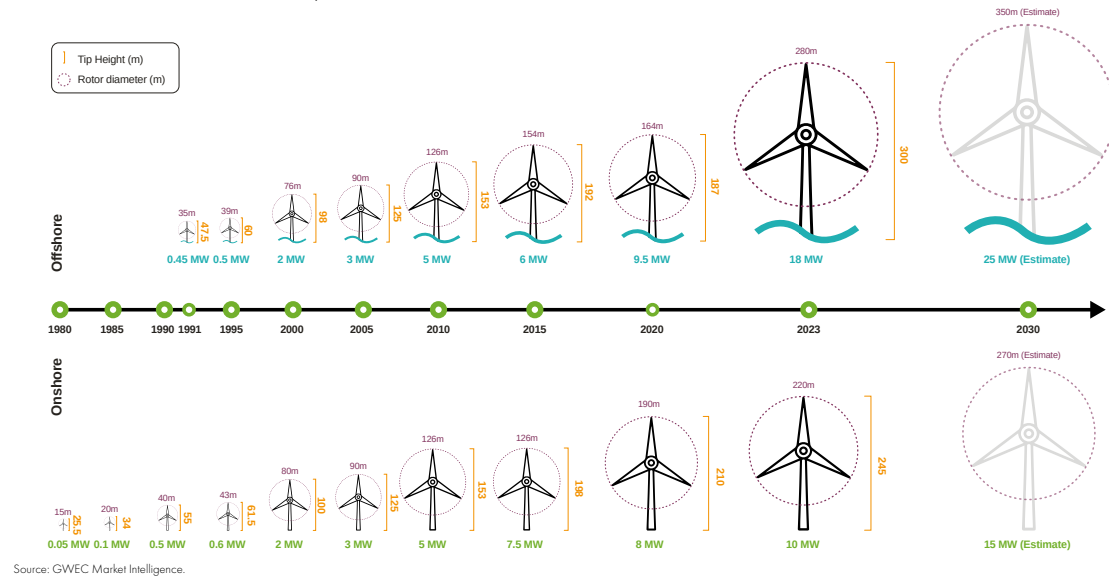


Figure 1.5: Increasing scale of onshore and offshore wind turbines [18]

In addition to the increasing scale of modern wind turbines, the deployment of turbines in offshore locations has become a major strategy for boosting power generation. This relocation enables wind turbines to benefit from the rich aeolian resources of open seas, which boast stronger and less turbulent wind resources compared to onshore locations. Offshore environments offer access to winds with greater velocities and airflow quality stemming from the reduction in surface turbulence arising as air travels across land features, inducing flow distortion [19]. By harnessing these superior wind conditions, offshore wind turbines can achieve significantly higher energy yields. A further benefit of relocating wind turbines offshore is the reduction in construction restrictions that are often imposed to mitigate the disturbances that can arise from wind turbine rotor wakes to communities in the surrounding areas [20].

The advantages of offshore wind deployment have been particularly transformative in countries that have embraced the technology. Denmark, for instance, leads Europe in wind energy integration, with wind power accounting for 56% of its electricity mix [17]. The significant role of offshore wind in countries such as Denmark, the UK, and the Netherlands is illustrated in Figure 1.6, where it has substantially increased the share of electricity generated from wind technologies. This strategic investment in offshore wind marks a key distinction between Ireland and Denmark, highlighting the disparity in wind energy utilisation between the two nations.

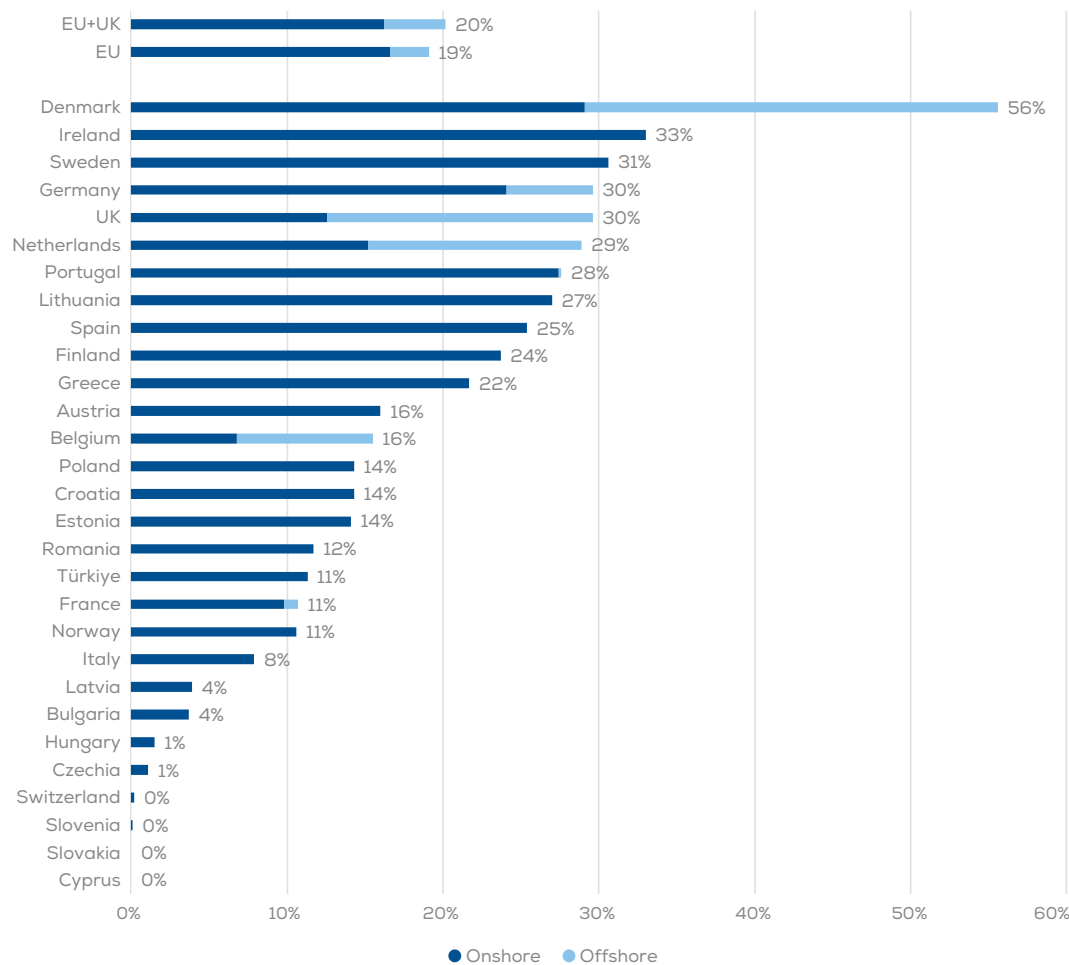


Figure 1.6: Percentage of electricity generated by wind energy in European countries [17]

As of early 2024, the Republic of Ireland has installed 4.74 GW of wind power capacity, generated from over 3000 wind farms. However, all of this energy is generated from onshore wind turbines. Despite the lack of offshore development, wind power remains the dominant source of renewable electricity in Ireland, accounting for 84% of total renewable energy generation and producing more than three times the electricity generated from coal, peat, and oil combined. The environmental and economic benefits of Ireland's wind energy sector have been substantial. In 2023, the 4.74 GW capacity contributed to the avoidance of 4.55 MtCO₂eq emissions by displacing fossil fuel-based electricity generation. Financially, the energy produced by the wind farms avoided the importation of €748 million worth of gas in 2024 and saved the taxpayer a further €268 million in carbon credits [21]. Furthermore,

in 2020, the sector supported over 5,130 jobs across the wind energy industry and its associated supply chain [22], highlighting its growing role in Ireland's economy.

While Ireland's total renewable energy mix accounts for 33.7% of electricity usage across the country [23], ranking it second in the world (see Figure 1.7), there is still a large scope for increasing wind power capacity in Ireland. Renewable energy only accounts for 14.1% of Ireland's total energy consumption, while 82.7% is still being generated via fossil fuels (see Figure 1.8). With a 55.6% reliance on oil [23], Ireland remains one of the most import-dependent countries in Europe, leaving it vulnerable to international geopolitical and economic events.

VRE share of power systems by country, 2022

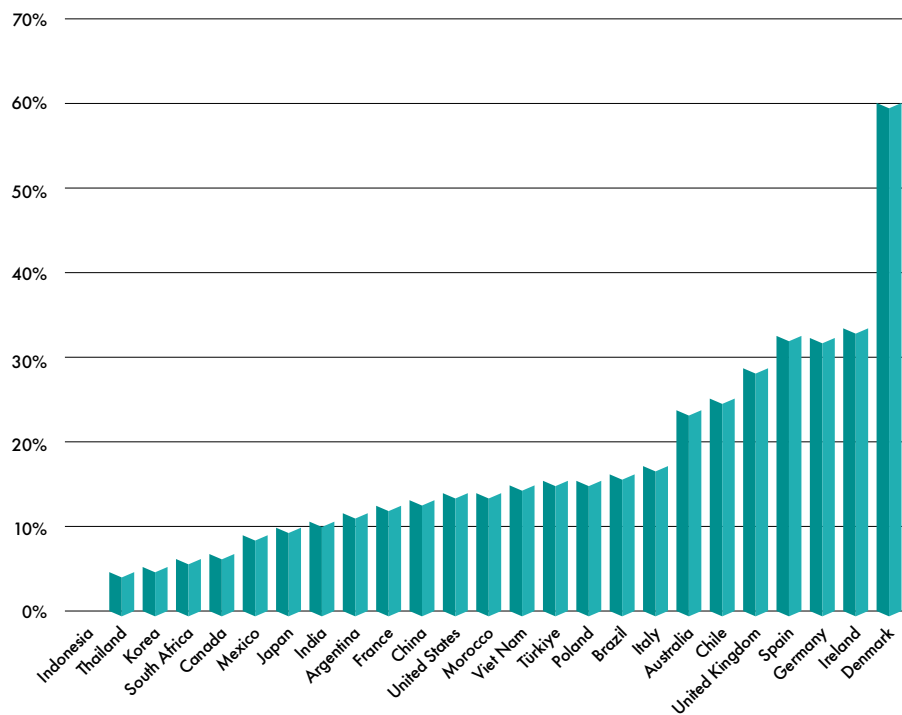


Figure 1.7: Percentage share of electricity generated by renewable resources world wide [18]

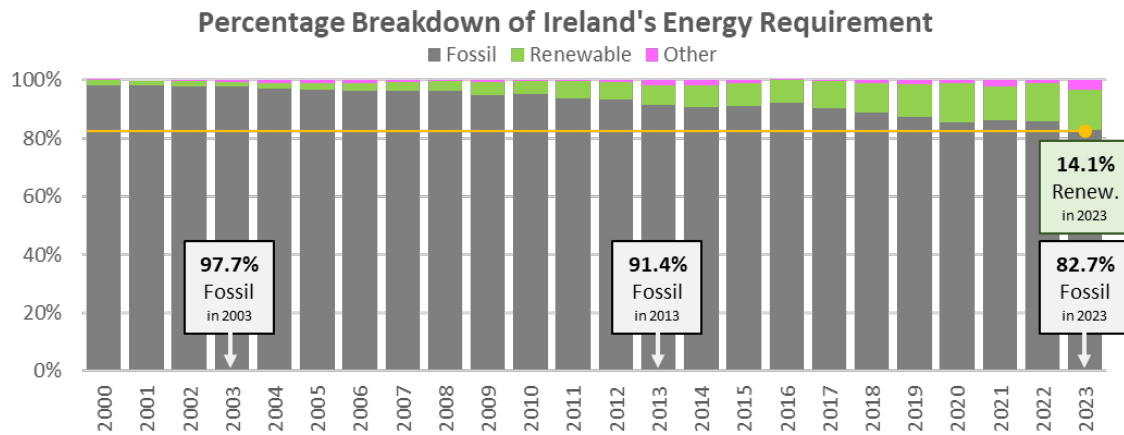


Figure 1.8: Ireland's percentage share of total energy demand supplied by fossil fuels vs. renewable energy [23]

A key opportunity to enhance Ireland's energy security and increase its renewable energy mix lies in offshore wind development. Despite having some of the strongest offshore wind resources in Europe (see Figure 1.9), Ireland's offshore wind industry remains severely underdeveloped. Ireland's abundant offshore wind resources stem from its geographical location in the Atlantic Ocean, where it is subjected to consistent and powerful southwesterly winds. With the nearest landmass to the west being the United States, these prevailing winds travel uninterrupted over the Atlantic Ocean, providing strong, steady wind flows, ideally suited for large-scale wind energy generation. Beyond its natural wind potential, Ireland has excellent infrastructure, such as ports and roads, required for the installation and maintenance of wind turbines. Furthermore, Ireland benefits from a highly educated and skilled workforce, making it an highly attractive location for investment in wind energy generation [24].

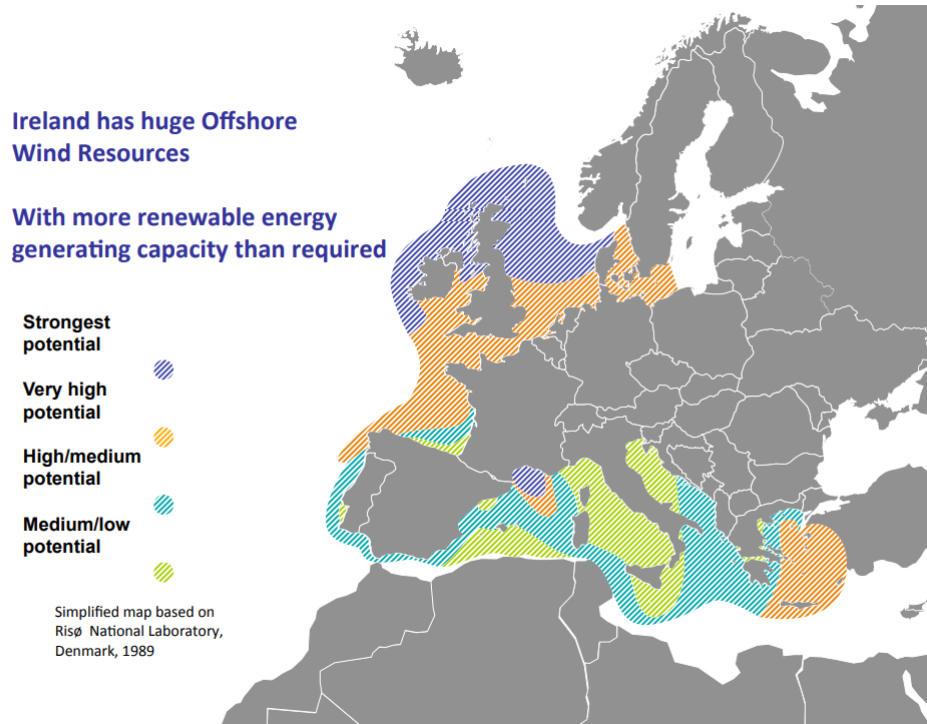


Figure 1.9: European offshore energy production potential [25]

1.2 The Role of Fatigue in Modern Multi-megawatt Wind Turbines

Ireland’s abundant wind resources, combined with advancements in wind turbine technology, make it well-positioned to generate vast amounts of renewable energy. However, the deployment of modern offshore wind turbines in locations like Ireland also presents significant structural and operational challenges. These challenges result from both the increasingly hostile environments in which offshore turbines are installed and the ever-growing size of the turbines themselves.

The offshore locations of these modern large-scale turbines expose them to a hostile and corrosive marine environment, characterised by highly fluctuating wind speeds, volatile sea states, and persistent saltwater exposure. As wind turbine dimensions continue to grow, modern multi-megawatt wind turbines are becoming increasingly slender and flexible structures, making them dynamically sensitive to external forces. This increased flexibility leads to larger structural displacements and accelerations, especially under the greater aerodynamic and hydrodynamic load-

ing experienced in offshore environments. These external forces are further compounded by the turbine’s considerable self-weight, with its continuous operation inducing large cyclic loads within key components. Over a typical lifetime of 20 years [26], offshore wind turbines experience more than 100 million load cycles [27], while enduring a range of weather conditions including variations in temperature, humidity, precipitation, and sunlight. This relentless exposure makes wind turbine components particularly prone to fatigue damage.

The vulnerability of wind turbines to fatigue damage has long been recognised. In 1999, Herbert J. Sutherland identified wind turbines as fatigue-critical machines, highlighting the inherent vulnerability of these structures to cyclic loading [28]. The phenomenon of fatigue has been studied for nearly two centuries, with Wilhelm Albert publishing the first documented work on fatigue damage in 1837 [29]. Two years later, in 1839, the term “fatiguing of engineering materials” was used by a French scientist, I. V. Poncelet [30], who described metals as being “tired”, which links the term to its Latin origin “fatigare” meaning to tire out. In the decades that followed, knowledge regarding the phenomenon of fatigue was progressively uncovered. In Donald J. Wolpi’s book [31], “Understanding how components fail”, the author refers to a definition of fatigue from the ASM Handbook from 1986. The handbook defines fatigue as “The phenomenon leading to fracture under repeated or fluctuating stresses having a maximum value less than the tensile strength of the material”. The author goes on to describe how fatigue fractures begin as microscopic cracks which increase in size over time, progressing into critical fractures resulting in failures. This process of microscopic crack growth makes predicting accurate stress time histories in wind turbine components critical for wind turbine design.

As wind turbines continue to increase in size and capacity, they are subjected to larger and increasingly fluctuating loads. Their dynamic responses, governed by these inherently stochastic environmental loads, are becoming increasingly complex to capture accurately. Performing simulations which accurately capture the resulting stress histories in their intricate components is becoming increasingly challenging and computationally demanding. This challenge is further compounded by the high number of uncertain variables governing the dynamic loads, which necessitates a large number of simulations to obtain statistically significant results. Consequently, fatigue has become not only more prevalent but also more complex to analyse and

mitigate [32]. These factors have therefore motivated a growing focus on advanced simulation tools capable of accurately predicting stress time histories of wind turbine components with low computational costs. Such models are essential not only for improving fatigue life estimation and ensuring structural reliability but also for enabling design optimisation at the structural level.

The physical scaling of turbines has placed additional emphasis on the performance and cost-effectiveness of these structures. Wind turbine monopoles, for instance, have significantly increased in both diameter and shell thickness to withstand the additional loads induced by the increased mass of the superstructure and the resulting increase in environmental loading [33]. However, this scaling has come at a cost, the support structure, including foundations, manufacturing, and installation, can account for up to 25% of wind project budgets [12]. With recent increases in interest rates, material costs, and supply chain disruptions creating economic challenges for the wind energy sector [34], optimising the design and manufacturing of turbine support structures has become a critical strategy for enhancing the economic viability of wind energy projects. However, such optimisation efforts should be undertaken with careful consideration, as overly aggressive optimisation to reduce expenditures can increase the susceptibility of support structures to fatigue damage, which has become a dominant failure mechanism in support structures [32]. Failures in these critical components can have severe consequences, including the catastrophic collapse of the turbine, resulting in significant financial losses. Numerous studies have identified fatigue as a primary cause of failure of modern wind turbines [19, 32, 33, 35] with welded connections in the support structures emerging as particularly exposed components [36, 37, 38, 39, 40, 41, 42].

Given the large CAPEX costs associated with modern wind turbines, ensuring the structural integrity and longevity of these systems is paramount, particularly in offshore environments where maintenance and repair operations are more challenging and costly. By accurately predicting component damage, operators and designers can safely optimise structural designs, maintenance schedules and end-of-life strategies. This not only minimises unplanned downtime but also improves LCOE calculations, helping to ensure both the structural safety and economic viability of wind turbines. Such fatigue life calculations are becoming increasingly important not only for the latest generation of large-scale turbines, but also as many early-generation wind turbines approach the end of their design life (see Figure 1.10).

This raises critical questions about the feasibility, safety, and cost-effectiveness of potential life extension strategies. These considerations are particularly vital as the wind energy sector strives to achieve cost parity with conventional energy sources while meeting ambitious renewable energy targets.

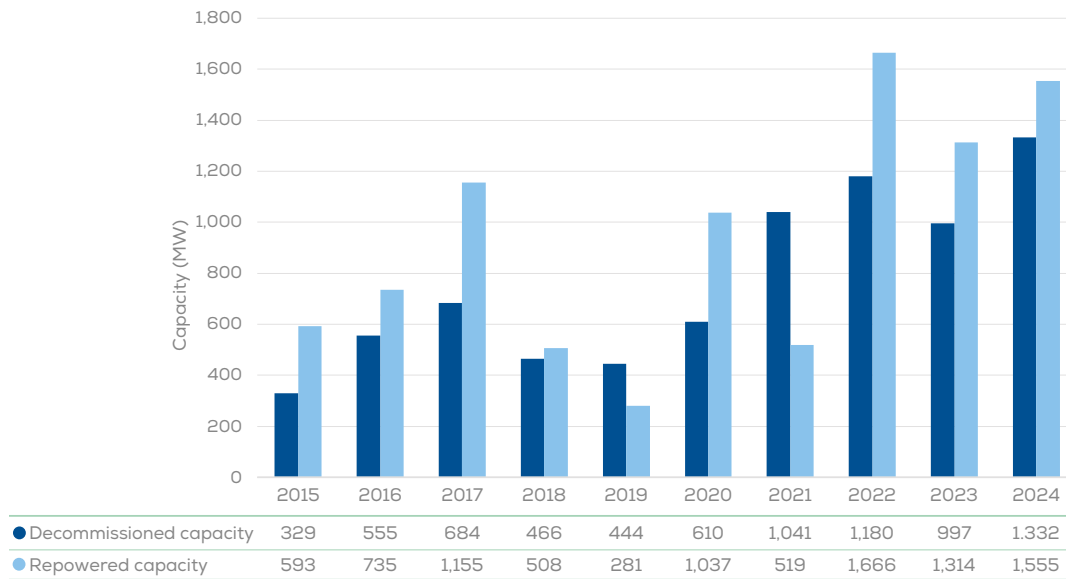


Figure 1.10: Decommissioned and repowered wind energy capacity in Europe [17]

1.3 Research Aims and Objectives

Building on the critical importance of accurate fatigue assessments for ensuring the structural integrity and financial viability of offshore wind turbines, the overarching aim of this research is to develop an enhanced methodology for fatigue assessment. The ultimate goal is to create an approach that provides improved accuracy while keeping computational effort to a minimum. Once developed, this modelling framework can subsequently be implemented to assess the life span of detailed components under environmental loading, including the effects of corrosion. Using the proposed framework, the work will additionally evaluate the effectiveness of damping devices in reducing fatigue loads in modern multi-megawatt offshore wind turbines. Finally, to address the lack of investment in the offshore wind turbine industry in Ireland, the suitability of two sites off the east and west coasts of Ireland will be evaluated.

To achieve the overarching aim of this thesis, the following objectives will be pursued:

- Development and validation of a high-fidelity, parametric FE model of an offshore wind turbine capable of calculating *stress concentration factors* and accounting for material degradation due to corrosion, with a specific focus on an internally bolted ring flange connection at the base of the turbine tower.
- Construction of a composite model framework by integrating an MBD model with the developed FE model, facilitating a comprehensive and detailed fatigue analysis beyond the current state-of-the-art.
- Execution of a complete fatigue assessment for prospective offshore wind turbine sites off the east and west coasts of Ireland. This will include the acquisition of environmental data, development of environmental probability distribution models, and evaluation of site suitability through fatigue analysis, utilising the composite model framework.
- Integration of auxiliary damping devices such as the tuned mass damper and tuned mass damper inerter into the composite model via the MBD model, to evaluate their effectiveness in mitigating fatigue loads.

These aims are addressed in the subsequent chapters of this thesis. A brief

overview of the structure and content of each chapter is presented in the following section.

1.4 Thesis Outline

Chapter 2 presents a detailed literature review, providing a broad discussion of the background theory of the key topics discussed in this thesis. This section discusses areas of research such as wind turbine modelling, wind turbine fatigue, corrosion in offshore structures and load reduction techniques through the use of control devices, highlighting the current state-of-the-art practices and areas requiring further investigation.

Chapter 3 details a state-of-the-art fatigue assessment for a modern offshore wind turbine. This section describes the development and validation of an MBD model used to perform a fatigue analysis, highlighting its holistic modelling capabilities under operational conditions. To ensure reproducibility and to facilitate comparison within the research community, benchmark conditions are applied. This chapter lays the groundwork for the advanced fatigue analysis techniques explored in subsequent chapters.

Chapter 4 introduces a novel integrated modelling approach for fatigue analysis, combining a FE model with an MBD model to capture both global system behaviour and local stress distributions in critical components. The chapter presents the design and resulting dimensions of a proposed ring flange connection at the base of the wind turbine tower, which is used throughout this thesis. It outlines the construction, meshing, and validation of the FE model, and concludes with an enhanced fatigue analysis conducted under benchmark conditions.

Chapter 5 investigates the impact of corrosion on the fatigue lifespan of offshore wind turbines by incorporating time-varying SCFs into the composite model. This approach allows for an accurate representation of material degradation over time, addressing a limitation in current design standards. Additionally, probability distribution models for key environmental parameters are developed for two sites off the coast of Ireland, enabling a detailed, site-specific fatigue analysis. The findings from this chapter offer valuable insights into the effects of corrosion and environmental variability, contributing to reliable lifespan predictions.

Chapter 6 explores the potential of fatigue life enhancement through the integration of both a TMD and TMDI within the composite modelling framework. This chapter evaluates the vibration mitigation capabilities of these damping devices and quantifies their impact on the fatigue life of offshore wind turbines.

Chapter 7 concludes the thesis by summarising the key findings and contributions of the research, while outlining potential avenues for future research. It highlights the advancements made in fatigue assessment methodologies, the effectiveness of damping devices, and their implications for the design and operation of offshore wind turbines.

Chapter 2

Literature Review

2.1 Introduction

The rapid growth of the wind energy sector, driven by advancements in turbine technology, has enabled the deployment of increasingly large and complex wind turbines, both onshore and offshore. These technological innovations are a product of extensive research across multiple engineering disciplines, including turbine design, materials, and control technologies. Despite the significant advancements, the wind turbine industry continues to face numerous challenges that hinder further evolution and the widespread deployment of next-generation systems. Among these challenges, fatigue damage has emerged as a particularly critical area of concern.

As discussed in Chapter 1, fatigue has been recognised as a primary cause of structural failure in modern wind turbines [19, 32, 33, 35]. In particular, welded connections within the support structure have consistently been identified as critical locations for fatigue damage accumulation [36, 37, 38, 39, 40, 41, 42]. The accurate fatigue assessment of such components is essential, not only for ensuring structural integrity and reducing maintenance costs, but also for enabling more efficient and sustainable turbine designs through an improved understanding of their fatigue behaviour. Within this broader context, this thesis investigates the fatigue performance of the circumferential butt weld connections located between turbine towers and transition pieces (depicted in figure 2.1), which have been implicated in several costly failures [43, 44].

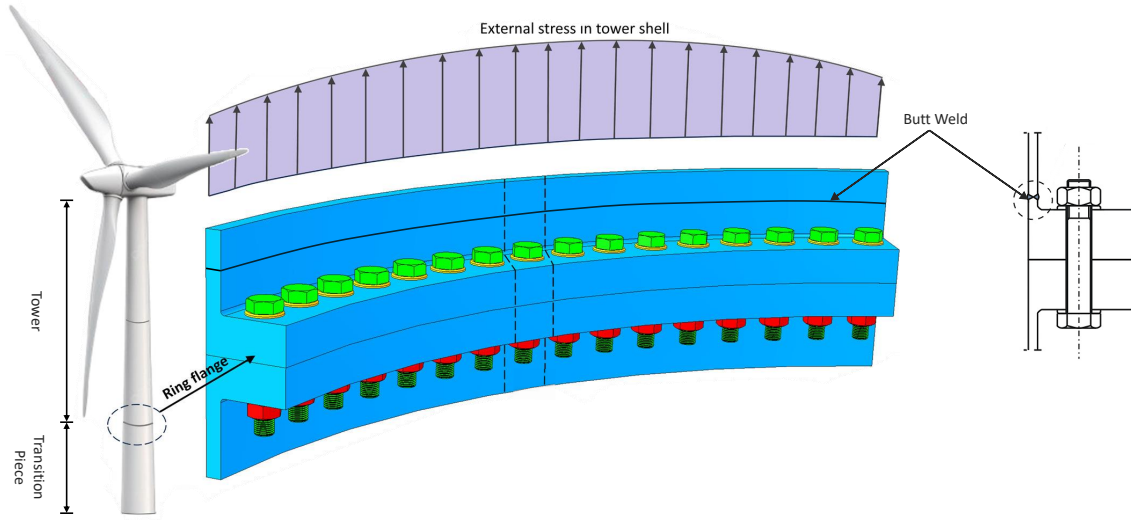


Figure 2.1: Tower-to-transition piece ring flange connection with butt weld detail adapted from [45] and [46]

With respect to this, this chapter examines recent research to assess key methodologies and ongoing challenges in wind turbine fatigue analysis. It begins with an overview of fundamental fatigue concepts, including accuracy requirements and relevant international standards. Next, appropriate modelling techniques are evaluated, highlighting their advantages and limitations, particularly in capturing the complex behaviour of large-scale turbines under real-world operating conditions. The review also addresses the role of corrosion in offshore environments as a contributing factor to accelerated fatigue degradation and concludes by exploring the emerging use of auxiliary control devices to mitigate fatigue damage and extend turbine lifespans.

2.2 Fatigue Damage Assessment

There are two principal approaches to fatigue analysis, namely the total-life approach and the defect-tolerant approach. The total-life approach utilises laboratory experiment results to estimate the number of cycles under which a carefully manufactured and smooth (unnotched) specimen can survive under cyclic loads of a specified amplitude. The total life approach assumes that the fatigue life of a component is primarily governed by the initiation and propagation of microcracks from the material's surface or inherent defects. By testing smooth specimens, the

approach focuses on the initiation phase of fatigue damage, which is critical for high-cycle fatigue scenarios where components are subjected to a large number of stress cycles at relatively low stress levels. This approach is particularly useful for real-world applications where loading conditions are more complex with varying stress amplitudes and mean stresses.

The Defect-Tolerance approach [47], in contrast, is based the principles of fracture mechanics. This method of analysis assumes that there is an imperfection or flaw present in the component and calculates the number of loading cycles required for the crack to grow to a critical crack length, at which point failure occurs. The crack growth behaviour can be determined through the use of empirical formulas such as the Paris Law [48] which relates the crack growth rate to the stress intensity factor of the component. The Defect-Tolerant approach is typically applied in low-cycle fatigue scenarios or in cases where crack propagation is the primary concern, such as in components with known defects or those operating under high-stress conditions.

The International Electrotechnical Commission’s (IEC) Standard 61400-1 [49] for wind energy generation systems specifies that fatigue damage in offshore wind turbines “shall be estimated using an appropriate fatigue damage calculation” and cites the use of Palmgren-Miner’s rule [50] in combination with S-N curves [51] as an example of an appropriate fatigue damage calculation. This total-life approach to fatigue analysis is recognised as the gold standard and has been implemented by a significant volume of literature concerning the fatigue analysis of wind turbines [20, 37, 52, 53, 54, 55, 56, 57, 58].

2.3 Wind Turbine Structural Modelling

Fatigue analysis is a critical aspect of offshore wind turbine design, as these structures are subjected to complex and variable loading conditions throughout their operational lifetime. Accurately assessing the accumulation of fatigue damage under such stochastic conditions requires simulation methodologies that can effectively capture the dynamic response of the system. Two primary approaches are commonly used for this purpose: time domain methods and frequency domain methods. Each approach offers distinct advantages and limitations in terms of accuracy, computational cost, and suitability for various applications.

2.3.1 Time Domain Methods

Time domain simulations are widely used in fatigue load assessments for offshore wind turbines due to their ability to accurately capture the complex behaviour of these systems [54, 55, 56, 57, 58, 59]. Wind turbines operate in highly stochastic and dynamic environments, where fluctuating wind speeds, turbulence, and changes in wind direction introduce significant variability in the applied loads. The increasing flexibility of modern turbines, combined with their sophisticated control algorithms for pitch and yaw regulation, creates strong interactions between environmental loads and structural dynamics, resulting in complex, highly nonlinear system responses. Quast and Sain [60] demonstrated the importance of including these control-structure interactions and actuator dynamics in offshore wind turbine design, demonstrating that neglecting such interactions can lead to misrepresentative system responses in simulation models.

Time-domain analysis has the unique capability to capture the coupled dynamics, incorporating the nonlinear effects inherent in wind turbine systems. This enables a comprehensive and realistic representation of turbine response under operational conditions. However, accurately modelling these dynamic effects requires high temporal resolution in time-domain simulations, which can incur significant computational costs. According to IEC International Standard 61400-1 [49], when performing fatigue analyses, simulations must be conducted across a sufficiently broad range of environmental conditions to reflect the long-term distribution experienced by a wind turbine. Furthermore, for each environmental condition considered, at least six independent 10-minute simulations must be performed to ensure statistically reliable results. This significantly increases the required number of simulations and consequently the computational burden, making fatigue assessment one of the most resource-intensive tasks in offshore wind turbine design [61], particularly when high-fidelity time-domain simulations are required.

Owing to the considerable computational cost associated with time domain simulations, a number of approaches have been developed to bypass the need for generating detailed stress time histories required for rainflow counting. One such approach is performing the analysis of wind turbines in the frequency domain.

2.3.2 Spectral Methods

Frequency-domain analysis or spectral analysis characterises the turbine’s dynamic behaviour using transfer functions that describe its frequency-dependent response. The aerodynamic and hydrodynamic loading on the turbine are represented using power spectral density (PSD) functions, which describe how the load’s energy is distributed across different frequencies. By combining the transfer functions with the PSDs, the system’s spectral response can be computed, bypassing the need for computationally expensive time-series simulations.

This methodology has been widely implemented for the fatigue analysis of wind turbines [52, 62, 63, 64, 65, 66, 67, 68, 69]. However, frequency domain approaches rely on linear system assumptions, which limit their ability to capture the nonlinear behaviour inherent in wind turbine dynamics, inevitably reducing their accuracy in comparison with time domain approaches [70]. Furthermore, spectral methods used for fatigue estimation are generally based on empirical formulas that employ the spectral moments of the input signal. Although these methods offer significant computational efficiency, they face a number of limitations. These shortcomings stem from both their limited theoretical backgrounds [71] and systematic biases, as wide-band solutions tend to underestimate the fatigue damage while narrow-band solutions tend to overestimate the damage [69]. These shortcomings make time-domain simulations the preferred approach for high-fidelity fatigue analysis in wind turbine applications.

2.3.3 Multibody Dynamic Models

Multibody dynamics is a key computational approach for simulating the time-domain behaviour of wind turbines. It enables the detailed modelling of flexible, interconnected components subjected to complex environmental loading. This approach is central to state-of-the-art wind turbine simulation tools such as OpenFAST [72], Bladed [73], and HAWC2 [74], which leverage MBD principles to capture the intricate interactions among aerodynamic and hydrodynamic forces, structural flexibility and control systems. Among these, OpenFAST distinguishes itself as a powerful and fully open-source wind turbine simulator. It builds on the legacy of FAST v8 [75], providing the research community with a physics-based representation of modern wind turbines. OpenFAST has been validated through a series of

benchmarking efforts and collaborative projects, ensuring strong agreement with experimental measures across a wide range of operating conditions [76]. It facilitates system-level modelling of wind turbines through the coupled simulation of aerodynamic forces, hydrodynamic interactions, system dynamics, control strategies and electrical generation systems. The flexibility of OpenFAST’s modular framework enables detailed investigation into a variety of research areas, including control system development, structural reliability, load reduction techniques, operational optimisation, and integration with electrical grids.

The ability of MBD models to generate accurate system-level simulations of wind turbine responses makes them particularly well-suited for fatigue analysis. Since fatigue assessments depend on detailed stress histories under realistic, time-varying operating conditions, MBD models provide a valuable framework for capturing the coupled effects of aero and hydrodynamic loading with control system interactions over extended simulation periods. As a result, MBD models have been widely adopted in fatigue research to evaluate the long-term structural performance of wind turbines.

Early investigations into the fatigue phenomenon in wind turbines were conducted by Larsen, Hansen, and Buhl [77], who used HAWC simulations to assess the impact of large blade deflections and their resulting stresses. The findings revealed that considering large deflections resulted in only minor differences in fatigue loading. However, it is important to note that this study was conducted in 2004, when wind turbine scales were significantly smaller compared to the multi-megawatt turbines in use today. Jonkman and Matha [78] [79], in their respective studies, highlighted the flexibility of MBD models by evaluating fatigue loads for three floating wind turbine support structure concepts for NREL’s 5-MW turbine. The three platforms, namely the MIT/NREL tension leg platform, the OC3-Hywind spar buoy (depicted in figure 2.2), and the ITI Energy barge, were compared to the land-based model. Using NREL’s FAST software, the authors analysed the dynamic response of the turbines under various environmental conditions. To quantify fatigue damage, the authors employed Miner’s rule in conjunction with the rainflow counting algorithm, enabling a systematic comparison of fatigue loads across the four turbine configurations. This approach underscored the capability of MBD models to efficiently accommodate modifications of wind turbine configurations, making them a powerful tool for evaluating both existing and emerging turbine designs.

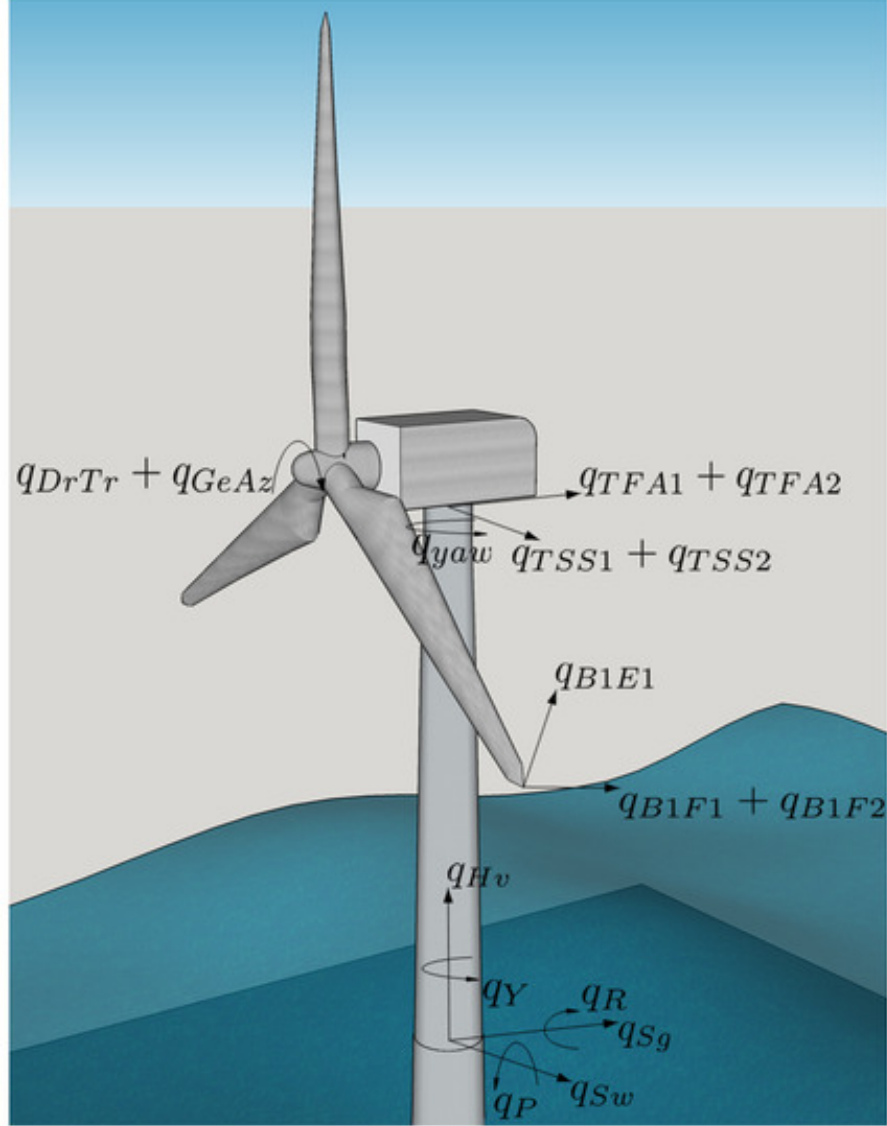


Figure 2.2: Multibody dynamic model of spar-type floating offshore wind turbine [80]

Toft et al. [55] demonstrated the efficiency of MBD models, performing over 34,500 simulations via FAST to investigate the effect of incorporating recorded wind climate parameters on wind turbine loads. Wind climate parameters were calculated from wind measurements from Boulder, CO, in the United States and Høvsøre in Denmark. NREL's 5-MW wind turbine was assessed under the environmental parameters at both sites using recommendations from the IEC [81] and the data recorded at the sites. Damage Equivalent Loads (DEL) were then calculated using rainflow counting with Palmgren-Miners rule, and a response surface was gener-

ated based on a central composite design. The results highlighted that the IEC 61400-1 standard [81], which uses a 90% quantile for turbulence, leads to a conservative assessment of the tower base fore-aft bending moments. Li et al. [20] further highlighted the importance of computational efficiency in fatigue analysis by assessing the short-term fatigue loads on NREL’s 5-MW floating offshore wind turbine with a spar-type platform. Using NREL’s openFAST simulator, they performed a large number of simulations across a broad range of environmental conditions, with simulation durations varying from 10 minutes to 2 hours. Fatigue damage was calculated using the rainflow counting algorithm and Palmgren-Miner’s rule, with an S–N curve selected from the DNV standard [82]. The study found that wave loading produced slightly higher fatigue damage than wind loading alone, and concluded that a minimum simulation length of one hour is necessary to achieve reliable fatigue estimates.

Leveraging the modular framework of MBD models, Marino, Giusti, and Manuel [59] compared fatigue damage in NREL’s 5-MW monopile-based wind turbine using nonlinear wave load modelling against traditional linear models. The nonlinear wave loads were generated using a domain-decomposition strategy [83] combined with potential flow theory, while linear wave loads were modelled using linear wave theory [84] and the JONSWAP spectrum, which describes the energy distribution of ocean waves in developing sea states [85]. Simulations were carried out using NREL’s FAST software to produce time histories, and fatigue damage was calculated using the rainflow counting algorithm with Palmgren-Miner’s rule. The research concluded that by implementing nonlinear wave loads, the fatigue damage was estimated to increase by 10% at high wind speeds and up to 17% at lower wind speeds.

Saenz-Aguirre et al. [57] harnessed the system-level capabilities of MBD models to assess the influence of oceanic waves on both fatigue damage accumulation and power output of NREL’s 5-MW OC3-Hywind turbine. Based on 30 years of wind and wave data from Hywind-Scotland, the study employed Ward’s minimum variance method [86] to cluster environmental conditions into 20 representative cases based on wind speed, wind direction, significant wave height, wave direction, and peak wave period. The simulations were carried out via OpenFAST, performing six 10-minute simulations for each case investigated in accordance with IEC specifications. The fatigue loads were then calculated using the rainflow algorithm with the

Palmgren-Miner rule. The study found that the DEL moments at the tower base, particularly in the side-to-side were heavily affected by climate-period-scaled waves, while there was little relation between the wave interaction and power production.

These papers collectively demonstrate the capability of MBD models to accurately and efficiently simulate the global response of wind turbine structures under a wide range of environmental and operational conditions. However, stemming from their reliance on simplified representations of complex geometries, MBD models often lack the granularity required to precisely predict the true stresses experienced by specific detailed components of the turbine, such as bolts, welds, or flanges. To achieve this level of detail, high-fidelity structural modelling approaches, such as FE analysis, are essential. These models, grounded in fundamental governing principles, provide detailed insights into deformations, strains, and stresses at a local level, enabling precise fatigue assessment of critical components.

2.3.4 Finite Element Modelling

The Finite Element Method (FEM) is a numerical technique for solving Partial Differential Equations (PDEs) that govern physical phenomena across engineering disciplines. The FEM is extensively used for stress analysis in structural mechanics, heat transfer simulations, Computational Fluid Dynamics (CFD), and electromagnetic field modelling, enabling high-fidelity solutions to complex multiphysics problems. The FEM is the simple idea of dividing an object into extremely small elements, also known as meshing, which are linked together using a nodal system. The method solves governing physical equations across this nodal network, with an increase in the number of elements increasing the accuracy of the analysis.

The FEM provides a powerful numerical technique for obtaining accurate solutions to a wide variety of engineering problems, including the complex behaviour of wind turbines. Its ability to accurately capture material plasticity and geometric nonlinearity makes it particularly valuable for improving our understanding of wind turbine dynamics. By producing highly accurate models, finite element analysis (FEA) enables designers to capture large displacements and accelerations that are critical in modern multi-megawatt wind turbines. As a result, the FEM has proven highly effective in analysing wind turbine responses under a range of demanding loading conditions, including extreme events such as seismic loading [87, 88, 89, 90,

91]. Additionally, the FEM enables the development of high-fidelity models that facilitate detailed examination of individual wind turbine components, allowing for the accurate evaluation of local stresses and strains. This capability is crucial for identifying potential problem areas, such as regions with significant displacements or high stress concentrations, which are critical for ensuring structural integrity and reliability. These detailed stresses can subsequently be used for structural evaluations, including fatigue life prediction.

FEA has been extensively applied in both the frequency and time domains for fatigue analysis of wind turbines [92, 93, 94, 95, 96, 97]. However, the FEM suffers from a number of limitations when performing fatigue analysis. As discussed in section 2.3.2, frequency-domain methods are limited in their ability to accurately model nonlinear effects inherent in wind turbine dynamics [98]. However, performing operational simulations via the FEM in the time domain also faces several challenges. FE software is not designed to compute aerodynamic loads directly, presenting a significant constraint for wind turbine analysis given their exposure to large, stochastic wind fields. Without integrated aerodynamic calculations, simulations must rely on simplified load inputs or externally derived aerodynamic solutions. These limitations are exemplified in the work of Hsu, Wu, and Chang [99], who conducted a fatigue analysis of a fixed-base 5-MW wind turbine tower located in Penghu, Taiwan, using ANSYS. In their work, wind speed data collected from the site was statistically modelled using a two-parameter Weibull distribution. From this model, 1000 wind speed samples were generated and used to calculate the corresponding forces and moments at the top of the tower based on basic aerodynamic relationships and the assumption of constant-amplitude wind loading. These loads were applied to a FE model of the tower to evaluate the resulting stresses at the base, assuming alignment of the nacelle and blade centre of gravity with the tower's central axis. In a similar manner, Huo and Tong [39] performed a fatigue analysis of a wind turbine tower by implementing time-domain simulations using a FE model of a 1.25-MW wind turbine structure located in Zhangbei County, Hebei Province, China. The FE model was used in conjunction with aerodynamic loads calculated using the Blade Element Momentum (BEM) theory to capture the dynamics of the wind turbine. While the concrete raft foundation and wind turbine blades were constructed with low fidelity, a detailed analysis was performed at the weld of the tower base connection and at the doorframe of the turbine tower. The fatigue analysis was carried out implementing Palmgren-Miner's rule with S-N curves selected

for appropriate detail categories according to the DNV GL guideline.

Despite the valuable insights offered in these works, they demonstrate the inherent limitations of FEM in capturing the complete aerodynamic loading environment of operating wind turbines. These limitations become particularly apparent when holistic system representations are required, including control system interactions and hydrodynamic environmental loading effects. Although these FE models excel at providing detailed stress histories for individual components, they cannot independently perform such integrated response simulations incorporating wind and wave load dynamics. This fundamental constraint has motivated the coupling of MBD models with FE approaches, enabling a more accurate representation of the turbine’s overall behaviour under complex operational conditions.

This methodology was adopted by Muskulus and Schafhirt [12] who utilised a decoupled FE model with wind loads calculated via time-domain numerical simulations to perform a reliability assessment of NREL’s 5-MW wind turbine with both a monopile and jacket type foundation. In this work, the wind loads were calculated assuming a rigid model of the support structure, producing loads in six DOFs at the tower top. A linear damper was added at the tower top to account for the influence of the motion of the structure on the aerodynamic loads. The fatigue analysis was carried out using the Palmgren-Miner linear damage accumulation rule with the rainflow counting algorithm. An appropriate S-N curve was selected from the DNV standard and the damage was calculated based on the hotspot stress approach. Extending this methodology, Yang et al. [100] performed a global optimisation of a tripod-support structure using aerodynamic loads calculated using NREL’s FAST software. These loads were applied to the FE model across all six DOFs at the tower top while the drag force equation was used to calculate wind forces on the turbine tower. Morison’s equation was employed to calculate the hydrodynamic loads applied to the wind turbine tower. Using these loads, time domain simulations were performed on the FE model of the turbine tower and support structure.

While the integration of FE models and MBD models produces stress time histories with enhanced accuracy, the application of the FEM in the time domain remains extremely computationally expensive. This high computational cost stems from both the complexity and scale of the wind turbines, which involve flexible structures subjected to dynamic loads. Achieving such accurate responses necessitates high spatial and temporal resolutions, further increasing computational demands.

These demands additionally scale with the number of finite elements, simulation duration, and the presence of non-linearities, rendering high-fidelity FE wind turbine models exceptionally resource-intensive. This limitation is especially pronounced in fatigue analysis, which requires relatively long-duration simulations for a wide range of environmental conditions. As a result, significant simplifications are often required to make simulations feasible within practical time frames. For instance, in the work of Yang et al. [100], the wind turbine rotor nacelle assembly was not physically modelled in the FE model, but was represented with moments and forces acting at the tower top. While in the more simplistic work of Huo and Tong [39], the wind turbine blades were modelled as simple cantilevers with rectangular cross-sections as presented in Figure 2.3, significantly reducing the model's complexity.

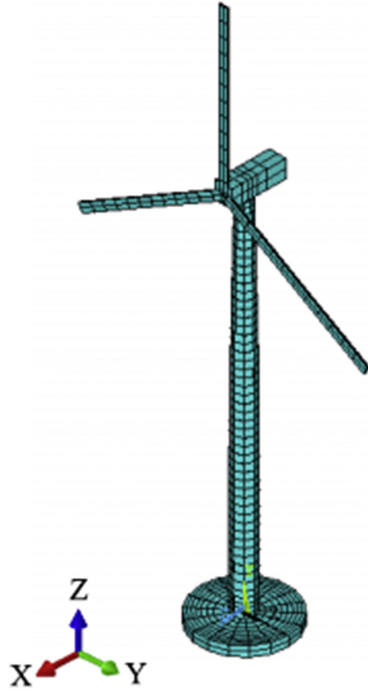


Figure 2.3: Simplified wind turbine finite element model [39]

These compromises, while necessary, can limit the accuracy of the stress predictions. To balance the high fidelity of FE methods with the efficiency of reduced-order models, one effective approach involves applying SCFs in combination with computationally efficient dynamic simulations.

2.3.5 Stress Concentration Factors

A stress concentration factor is defined in the DNV-GL [101] standard for the fatigue design of offshore steel structures as the ratio of hotspot or geometric stress range to the nominal stress range. SCFs serve as a bridge between local and global structural assessments, providing a means to account for the effects of macro geometry on stress distributions [102]. They quantify the level of stress amplification induced by geometric discontinuities such as welds, notches, or abrupt changes in cross-section. An example of this can be observed in Figure 2.4, which demonstrates the rise in stresses in a tension bar due to the presence of a hole. The stress concentration factor K_{tg} can be calculated as:

$$K_{tg} = \frac{\sigma_{\max}}{\sigma} \quad (2.1)$$

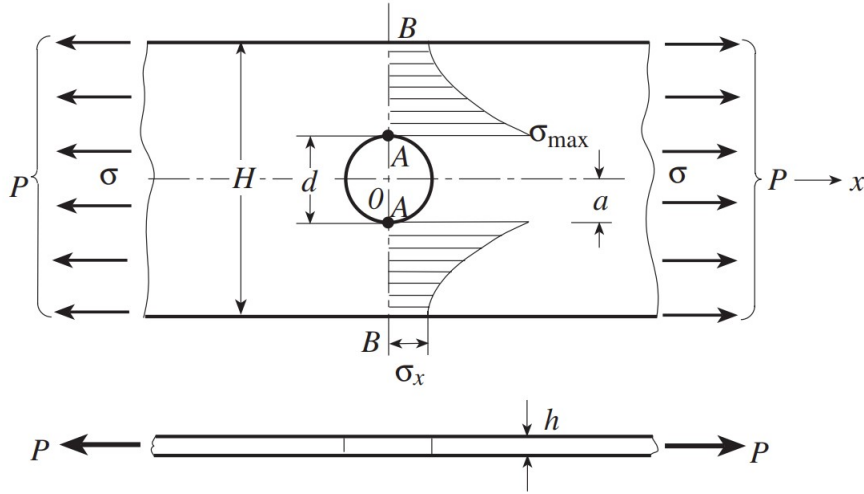


Figure 2.4: Stress distribution in tension bar with central circular opening [103]

The importance of SCFs in capturing local stress variations has long been recognised, even prior to the development of detailed structural analysis tools such as FEM. Traditionally, SCFs were determined through experimental methods. For instance, in 1962, Shea [104] performed experimental testing on a thin plate with a central circular hole to assess the impacts of dynamic loading on SCFs. It was found that for pulses which were much larger than the natural frequency of the plates, no dynamic effect was observed. Gandhi, Raghava, and Murthy [105] performed experimental tests to assess the fatigue life of internally stiffened ring tubular joints.

Initially, static loading was applied to the joints, calculating SCFs for both stiffened and unstiffened joints. Following this, fatigue tests were carried out on T and Y joints in air, seawater and seawater with cathodic protection. The experimental results were compared with American Petroleum Institute (API) and the U.K. Department of Energy (DEn) design curves.

Over time, experimental methods have largely been replaced by FEA, which has become increasingly prevalent in offshore structures due to its accuracy and versatility [106]. Moreover, SCFs have begun to be estimated using analytical and semi-empirical formulations. This is particularly well established for jacket-type substructures, which were commonly used in the offshore oil and gas industry, such as the widely adopted Efthymiou formula [107]. This formula provides SCFs of tubular joints typically used in offshore structures, including T/Y, K, and KT joints under axial, in-plane, and out-of-plane bending loads. These approaches offer efficient alternatives to detailed FE modelling, providing SCF estimates based on geometric parameters, eliminating the need for computationally intensive simulations. Many of these formulations are grounded in fundamental principles such as shell theory and are often validated against extensive FE analyses or experimental data. Hellier, Connolly, and Dover [108], for example, developed a set of semi-empirical parametric equations for axial, in-plane, and out-of-plane bending loads in Y and T joints based on over 900 thin shell FE analyses. Similarly, Lotsberg [109] implemented shell theory to produce analytical expressions for SCFs in tubular structures arising from fabrication tolerances and transitions of thickness in circumferential butt welds. These expressions demonstrated good agreement between the SCFs calculated and the DNV standard [110].

While such formulations offer significant efficiency advantages, their accuracy is not universal across all joint types and configurations as demonstrated by Hellier, Connolly, and Dover [108] who found that the semi-empirical equations were conservative when compared to experimental results and existing equations. Further highlighting this issue, Saini, Karmakar, and Ray-Chaudhuri [35] found that in general, the parametric equations used to calculate SCFs of joints tend to overestimate the SCFs in comparison with measured experimental results, while better agreement was noted between the experimental results and FE approaches.

These findings emphasise the need for careful validation and refinement of SCF estimation methods, particularly for complex joint geometries. Beyond their

questionable accuracy, such formulas are often limited in scope, lacking universal applicability across all joint configurations. Additionally, misclassification of joint types can lead to significant errors in fatigue life predictions, as noted by Kršćanski and Turkalj [111]. This limitation in applicability is particularly relevant in the context of monopile substructures, which are currently the most commonly adopted support structures for offshore wind turbines, accounting for the largest percentage of installed capacity in Europe [112]. These substructures typically employ bolted ring flange connections between the turbine tower and substructure, which are the most commonly employed connection for both monopile and jacket-type structures [113]. Despite their prevalence, research on monopile-to-tower flange connections, in particular relating to SCFs, is thus far limited. This gap was highlighted by Zou et al. [114], who additionally stated that the local stresses that arise in these connections have induced the failure of numerous wind turbines. Their study also draws attention to a broader issue in existing research that the fatigue analysis of wind turbine towers often overlooks the influence of local geometric features, which can significantly affect stress concentrations and fatigue life.

Addressing this gap, Seidel [46] developed an analytical model for calculating SCFs of bolted wind turbine flanges with the ultimate goal of investigating the potential increase in fatigue loading at welded flange necks due to local bending moments in the tower wall. Seidel's model employs shell theory to estimate the stresses resulting from the geometric discontinuity introduced by the flange, treating the flange connection as a ring beam with constrained rotational freedom. This approach allows for the prediction of local bending effects induced in the tower wall stemming from the stiffening effect of the flange. The analytical results were validated against a detailed 3D FE model, depicted in Figure 2.5, showing strong agreement. In a later paper, Seidel, Wegener, and Dijk [115] investigated the impact of flange dimensions and imperfections in the flange geometry on these SCFs. It was found that connections with greater shell thicknesses are increasingly likely to become critical fatigue hotspots, while geometric imperfections were noted to have negligible influence on the SCFs.

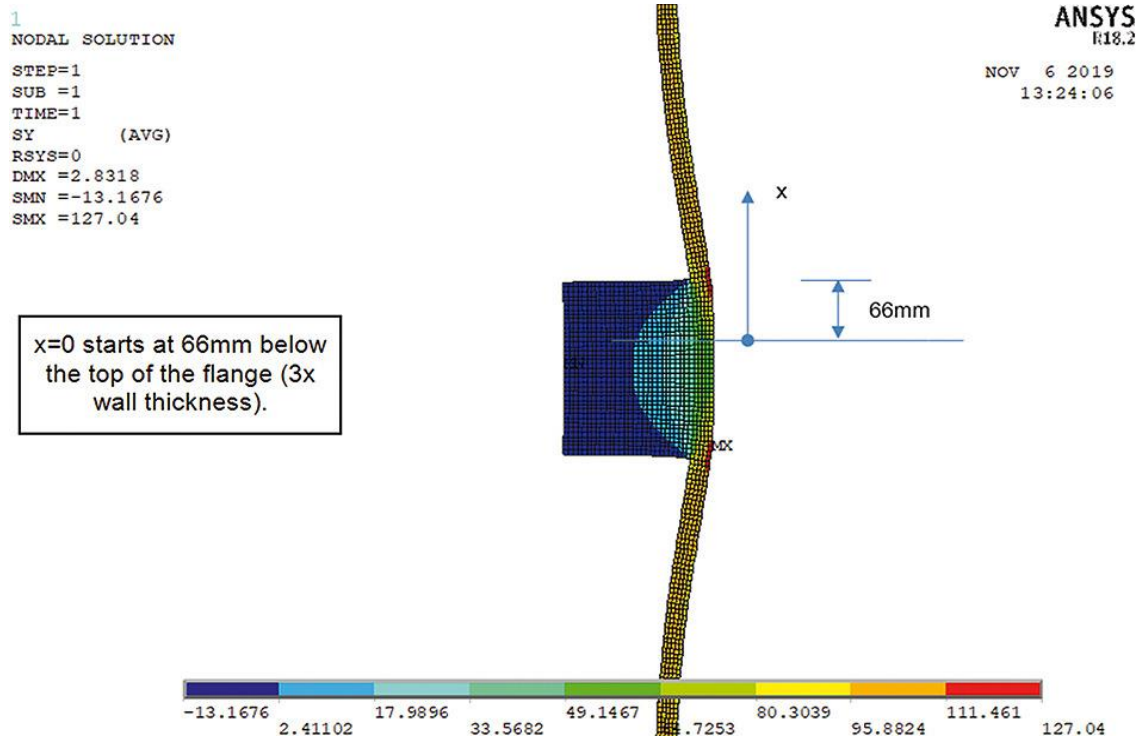


Figure 2.5: Stress distribution in ring flange connection via finite element analysis [46]

The use of SCFs such as these allows lower-fidelity models to be employed in fatigue analysis by enabling local stresses to be estimated from global structural responses. This approach significantly reduces computational cost while maintaining accuracy and has become widely adopted in FE-based fatigue assessments of offshore structures [36, 53, 54, 67, 69, 116, 117, 118, 119]. A notable contribution by Yeter, Garbatov, and Soares [120] involved a detailed fatigue analysis of a tripod substructure supporting NREL's 5-MW offshore wind turbine. In this work, a shell-based FE model was developed in ANSYS, as shown in Figure 2.6, with aerodynamic and gravitational loads from the blades and nacelle applied at the tower top as equivalent forces and moments using a decoupled approach. Morison's equation was used to calculate the wave loads on the submerged section of the support structure, while the wind and ocean current loads were calculated using the drag force equation. The dynamic analysis was performed using a global FE model, from which the resulting stresses were extracted and combined with SCFs, obtained from a finely meshed local sub-model, to calculate the local stresses expe-

rienced by structural components. Fatigue damage was then evaluated using these local stresses with S-N curves from the DNV with a thickness modification factor applied.

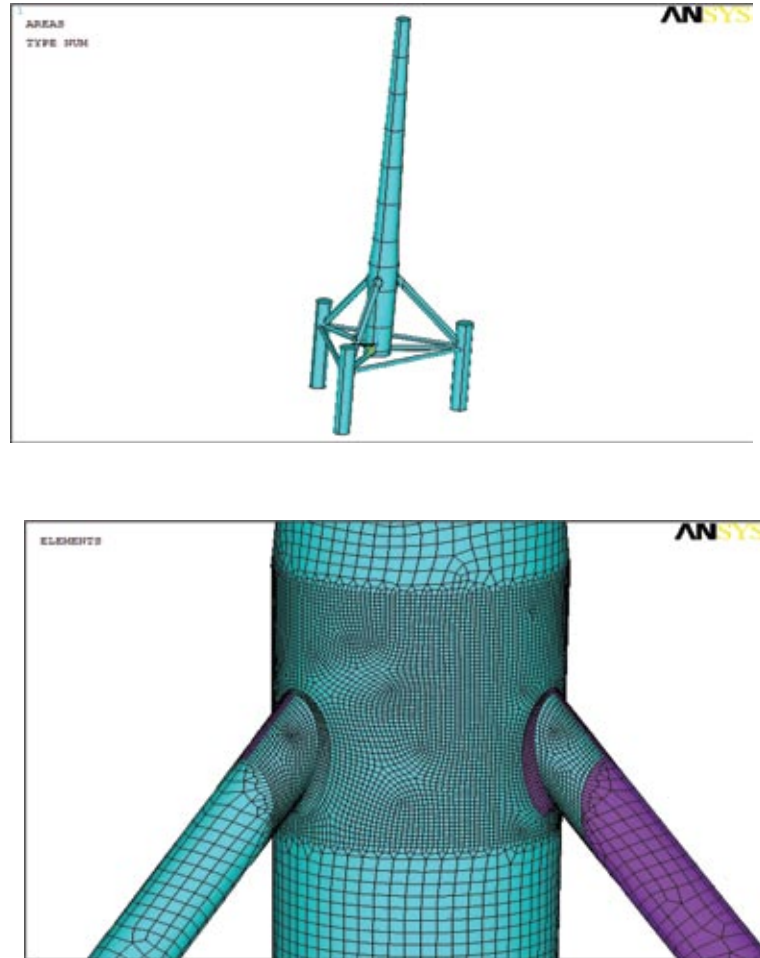


Figure 2.6: Global and local finite element model implemented in Yeter, Garbatov, and Soares [120]

However, as with general FE models, many of these approaches struggle to accurately capture the complex loading conditions experienced by wind turbines. To address this, lower-fidelity FE models incorporating SCFs have been integrated with MBD models [38, 121]. This coupling enables the simulation of the turbine's holistic response to environmental loading, including aerodynamic, hydrodynamic, and control system interactions, while still allowing for the accurate estimation of

local stresses in critical components. Although the use of SCFs allows for reduced-fidelity FE models, these approaches remain constrained by the significant computational demands associated with wind turbine analysis, often necessitating further simplifications within the FE model.

To bypass the requirement of the dynamic FE simulations while still capturing accurate local stress responses, researchers have coupled MBD models with SCFs derived from empirical or analytical formulas, or extracted from high-fidelity FE analyses. This approach enables efficient fatigue assessment without compromising the accuracy of local stress predictions. Several recent studies have highlighted the effectiveness and versatility of this method for fatigue life estimation. Velarde et al. [122] applied this methodology in a fatigue analysis, which assessed the fatigue design factors assigned in the DVNGL standard. The time domain simulations were performed using HAWC2 based on the DTU 10-MW offshore wind turbine [123] and the resulting fatigue damage was calculated using Palmgren-Miner's rule using an S-N curve for structures exposed to seawater with cathodic protection. A SCF of 1.1 was assumed at the welded connection, with additional consideration given to the plate thickness. The study concluded that a fatigue design factor of at least 3 should be recommended by standards. Fu et al. [40] emphasised the critical role of localised stress analysis in their investigation of the fatigue damage in wind turbine tower flanges and bolts. Their study employed the orthogonal expansion method [124] and the numerical-theoretical method [125] to derive 15 independent variables characterising both temporal and spatial fluctuations in wind loads. These variables were then used to assess the dynamic stresses experienced by the turbine, with results validated against experimental data. Fatigue loads in the connection flange and bolts at the turbine tower base were calculated using the rainflow counting algorithm and Palmgren-Miner's rule with a SCF of 1.12 applied to the stress histories for the connection flange. The study concluded that fatigue failure in the flange would occur significantly earlier than in any of the bolts. Nispel et al. [126] adopted the methodology to investigate the effect of soil-structure interaction on the fatigue life of NREL's 5-MW offshore wind turbine employing nonlinear springs at the mudline. Aerodynamic loads were generated using NREL's FAST software, while hydrodynamic loads were calculated via Morison's equation combined with a dynamic amplification factor. Fatigue damage over a 30-year design life at a site in Blyth, UK, was estimated using the rainflow counting algorithm and Palmgren-Miner's rule, incorporating a SCF and a bi-linear S-N curve from the DNV standard.

The analysis identified wind loading as a key contributor to fatigue damage, while other parameters such as tower stiffness and material properties were found to have less influence.

Zhu et al. [41] applied the coupled approach while additionally accounting for the influence of corrosion on the turbine tower. In this study, a novel framework for the probabilistic modelling of the long-term fatigue reliability of wind turbines was proposed. The approach combined short-term numerical simulations with a Markov model constrained by environmental conditions to capture the progressive nature of fatigue damage, while also considering uncertainty in the selected S–N curve. The framework was applied to NREL’s 5-MW monopile-supported wind turbine located in the North Sea, with time-domain simulations carried out using OpenFAST. The fatigue life was examined at the tower base, considering the axial forces and bending moments while neglecting shear forces. The fatigue damage was calculated using the rainflow counting algorithm and Palmgren-Miner damage rule, incorporating an S-N curve for seawater with cathodic protection provided by the DNV standard. A SCF and thickness correction factor were applied to the stresses obtained from OpenFAST simulations. The study concluded that in the absence of additional inspections and maintenance in corrosive environments, the failure probability increases approximately tenfold over a period of 10 service years. The emphasis on corrosion in this study and the corresponding result is particularly relevant as modern wind turbines are increasingly deployed in aggressive offshore environments, where regions of high SCFs are additionally exposed to elevated corrosion rates. Consequently, a deeper understanding of corrosion mechanisms and effective mitigation strategies is essential for ensuring long-term reliability and safety.

2.4 Corrosion in Offshore Wind Turbines

2.4.1 Corrosion Mitigation Measures

Corrosion-induced damage in offshore steel structures is primarily driven by the dissolution of metal and hydrogen embrittlement [127]. To mitigate these effects, DNV-RP-0416 [128] recommends three principal corrosion control strategies: corrosion allowance, cathodic protection, and protective coatings. Corrosion allowance is the provision of additional steel thickness in components to serve as a sacrificial layer that gradually corrodes over time [129]. This approach is passive and

requires no external power or maintenance once implemented. However, applying corrosion allowance to large structural elements, such as turbine towers, can significantly increase the total mass of steel required. This can greatly increase the price associated with the raw material and fabrication, in addition to increased expenditures associated with transportation and installation, as heavier components may require machinery with increased lifting capabilities. Furthermore, the added mass and stiffness may alter the tower's dynamic response, which must be carefully considered in design.

An alternative corrosion protection strategy is the use of protective coatings, typically metallic or polymer-based materials applied to component surfaces during fabrication. These additional layers act as a barrier between the steel and the marine environment. This prevents contact between the steel and the water and oxygen, thereby inhibiting the electrochemical reactions which drive corrosion [130]. However, these coatings experience ageing and degradation with time. In the splash zone, the protection may be expected to remain effective for up to 10 years [131], though this duration is often considerably reduced due to imperfections arising in the application of the protective layer [129]. Moreover, these protective layers can be easily damaged during transportation and installation [132], leaving the underlying steel vulnerable to corrosion. Once damaged, the repair of these coatings is challenging and expensive, with the repair of such layers costing approximately 50 times the initial installation costs [130].

The final corrosion protection strategy is the implementation of cathodic protection, which involves creating an electrical circuit between the steel component and a sacrificial anode. In this system, the anode corrodes, supplying electrons to the protected surface and thereby preventing the electrochemical reactions responsible for metal corrosion [130]. However, this approach has a number of limitations. Cathodic protection is only effective when the component is fully submerged in an electrolyte (salt water), making it unsuitable for continuous protection in the splash zone [129]. Moreover, the application of cathodic protection systems is costly, and these expenses can become significant when applied across large wind farms with many turbines [132]. Environmental concerns have also been raised regarding the emission of metal ions from sacrificial anodes into surrounding waters [133]. Furthermore, several documented failures of these systems have resulted in premature corrosion arising at offshore wind sites [129, 134, 135, 136].

2.4.2 Modelling of Corrosive Effects in Offshore Wind Turbines

Despite the limited mitigation capabilities of the protection approaches that are available, the time-dependent effects of corrosion are often overlooked in current design standards [137]. A practical approach to account for material degradation over time is to incorporate the effects of corrosion-induced material thinning directly into SCF calculations, thereby allowing more realistic fatigue life assessments of offshore structural components.

Addressing the impact of corrosion on welded joints, Dong, Moan, and Gao [36] performed a fatigue analysis considering the effects of corrosion on the welded connections of a jacket-type offshore wind turbine support structure using Efthymiou's formulas [107] to calculate the hotspot stresses. The study found that if no maintenance is carried out following the failure of the protection system, the probability of failure of the support structure increases significantly. Building on this work, Li et al. [138] implemented the Efthymiou empirical formula to calculate SCFs in the tubular joints of a monopile-based tripod substructure. These SCFs were applied in combination with S-N curves and Palmgren-Miner's rule to assess the fatigue damage for a fixed-base offshore wind turbine support structure. This paper found that the reduced thickness of joints caused by corrosion resulted in a significant decrease in the fatigue capabilities of the support structure. Extending the effects of corrosion thinning to a structural scale, Okenyi et al. [139] performed a stress analysis of the IEA 15-MW offshore wind turbine subjected to corrosion via Euler-Bernoulli beam theory and FEA. The results of the two analyses showed good agreement, illustrating the increase in stresses observed in the turbine support structure and tower that arise due to the corrosion. The paper demonstrated the importance of considering material wastage due to corrosion in stress analysis to provide safe and accurate designs. Taking a more detailed approach at the component level, Zhang et al. [42] proposed a probabilistic Corrosion-Fatigue (C-F) deterioration model for high-strength bolts of wind turbine flange connections. A multi-physics simulation was used to calculate the time-history response of the system, from which stress ranges in the bolts were derived using SCFs and a load transfer function based on the Schmidt and Neuper approach [140]. These stress ranges were then applied within the C-F model to assess fatigue performance. The study found that, due to the combined effects of corrosion and fatigue, the bolts'

service life could be reduced to as little as 30.4% of their original expected life span.

Given the documented failures of cathodic protection systems below the mean sea level, the limited durability of protective coatings, and the high costs associated with corrosion allowance techniques, the consideration of corrosion effects in offshore wind turbines becomes particularly relevant. Moreover, the significant reductions in service life reported in the reviewed studies highlight that corrosion-induced degradation is a critical factor that must be incorporated into fatigue analyses to ensure accurate lifespan predictions for offshore wind turbines operating in corrosive marine environments.

2.5 Mitigation of Wind-Induced Vibration

As previously discussed, in an effort to maximise energy capture, modern wind turbines have dramatically increased in scale and are increasingly being deployed in offshore environments. While these trends enhance power production capabilities, they also result in greater dynamic responses and heightened exposure to corrosive conditions, both of which contribute to elevated fatigue damage. To address these challenges, engineers have developed various vibration control strategies aimed at mitigating excessive oscillations, reducing dynamic loads and improving structural reliability.

Several studies have explored active aerodynamic control through innovative pitch or generator torque control strategies [141] such as Namik and Stol [142] who utilised a periodic state-space controller with individual blade pitching to reduce power fluctuations, fatigue loads and platform rolling. Expanding on this Fitzgerald, Staino, and Basu [143] developed a novel individual pitch control strategy to reduce structural vibrations of turbine blades whilst also maintaining power output within an allowable range. Despite the reductions that can be achieved by adjusting pitch and generator torque control algorithms, modern wind turbines continue to exhibit considerable dynamic behaviour. This has led to designers increasingly adopting auxiliary structural control devices, with the classical TMD emerging as a particularly effective solution.

2.5.1 Tuned Mass Damper

A tuned mass damper, also known as a harmonic absorber, is a sophisticated vibration control device widely used in engineering to mitigate unwanted oscillations in structures and mechanical systems. The device consists of three key components, a mass, springs, and viscous dampers. The mass is typically a large block of steel or concrete, which is connected to the main structure via springs and dampers in parallel, as depicted in Figure 2.7. The device works by tuning the stiffness of the springs and the damping coefficient of the viscous dampers such that the mass oscillates out of phase with the main structure and by doing so absorbs and dissipates kinetic energy in the main structure. The concept of TMDs dates back to the early 1900s when they were first developed to address the rolling motion of ships. Since then, they have been adopted across a wide range of industries, from providing damping in high-rise buildings to improving suspension systems of Formula 1 cars.

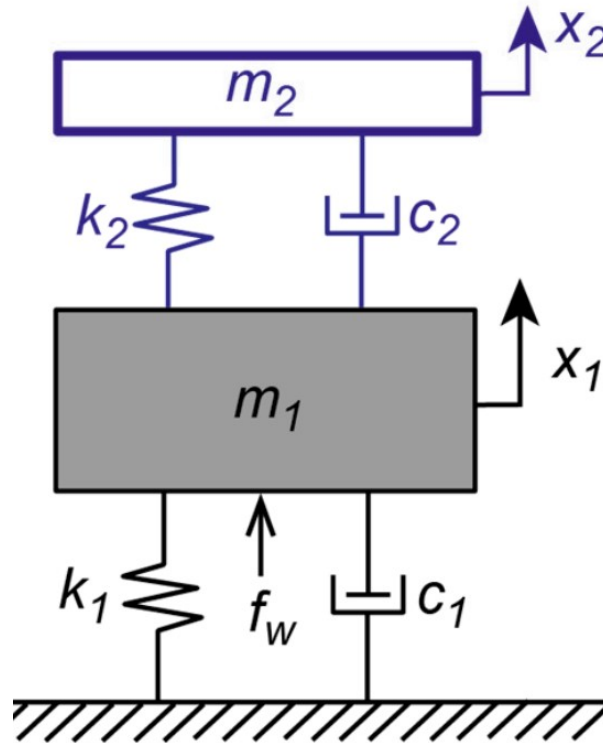


Figure 2.7: Tuned mass damper configuration [144]

In an effort to mitigate the large displacements and accelerations that give rise to the fatigue loads experienced by large flexible wind turbines, designers have

begun implementing damping devices such as the TMD in the blades and towers of wind turbines. Owing to their simple mechanical design, they can be readily implemented in both FE and MBD models, facilitating efficient assessment of their vibration control effectiveness.

In an early study, Teramura et al. [145] demonstrated that by installing a TMD in the tower of a turbine, the accelerations experienced during operation could be reduced by 50% while the tower damping for the first mode of vibration could be increased from 0.7% to 9%. Building on this concept, Murtagh et al. [146] derived an optimal tuning ratio for TMDs in lightly damped structures. The optimised TMD configuration was then tested numerically using rotationally sampled wind turbulence, where it was observed to largely reduce structural vibrations in wind turbines. Expanding the application of TMDs, Lackner and Rotea [147] modelled two TMDs (one in the fore-aft direction, one in the side-side direction) in the nacelle of NREL's 5-MW wind turbine for both monopile and barge platforms. The TMDs were found to have a positive effect on the systems, in particular the more dynamically sensitive barge configuration. Further extending research into their application in floating wind platforms, Si, Karimi, and Gao [148] modelled a TMD in NREL's OC3-Hywind spar-type floating wind turbine. Their study revealed that a TMD with low spring and damping coefficients provided significant vibration reduction at wind speeds above rated conditions but adversely affected performance at or below rated wind speeds. Conversely, a stiffer TMD design yielded moderate response mitigation across the full operational range. Building on this work, Si, Karimi, and Gao [149] identified the optimum design parameters of a TMD in a spar-type floating offshore wind turbine to maximise the load reduction capabilities of the device. Furthering their earlier research [147], Stewart and Lackner [150] evaluated the performance of TMDs under wind-wave misalignment conditions. In their study, a TMD was installed at the tower top of the monopile-supported 5-MW NREL wind turbine, where it was found to reduce side-to-side displacements by over 40%.

Building upon conventional TMD designs, Sun and Jahangiri [151] carried out a numerical analysis on a three-dimensional pendulum TMD (3D-PTMD), finding that the 3D-PTMD reduced the peak and RMS responses by 10% more than a traditional two-dimensional TMD when a mass ratio of 2% was selected. To assess the extent of the device's mitigation capabilities with regard to fatigue, Sun

and Jahangiri [152] extended this work by comparing the fatigue performance of a 3D-PTMD with that of dual linear TMDs installed in the side-to-side and fore-aft directions, with both configurations having a mass ratio of 2%. Fatigue damage was calculated using the rainflow counting algorithm and Miner’s rule, based on wind and wave conditions specific to the Delmarva Peninsula, US. The results demonstrated that 3D-PTMD significantly outperformed the dual TMDs across all conditions and as a result, greatly mitigates fatigue loads in comparison. Cao et al. [153] also examined the fatigue life of NREL’s 5-MW offshore wind turbine equipped with a tower-top TMD. This study employed a low-fidelity FE model composed of beam elements which was developed in MATLAB. Within this model, the wind and wave loads were calculated using BEM theory and Morison’s equation, respectively. The resulting stress time histories were adjusted using a SCF and a thickness correction factor relevant to the detail category. Subsequently, the fatigue damage was evaluated using the rainflow counting method in conjunction with the Palmgren-Miner rule, applying an S–N curve for submerged steel from the DNV standard. Based on the simulations, the TMD was optimised to minimise the fatigue damage experienced by the turbine tower. Through optimisation of the TMD parameters, the study achieved a 25% reduction in fatigue damage compared to conventional configurations. Martin del Campo, Pozos-Estrada, and Pozos-Estrada [154] similarly assessed the probability of failure for a range of land-based wind turbines equipped with TMDs under cyclone and seismic loading. Using motion records from real events in Mexico, a fragility analysis was conducted, revealing that the inclusion of TMDs significantly reduced the probability of failure and contributed to extended lifespans of structural components. The effectiveness of TMDs in mitigating seismic responses in wind turbines has additionally been demonstrated in several other studies [155, 156, 157, 158, 159, 160, 161], proving the device capable of seismic response control by reducing the loads and displacements induced in wind turbines.

Despite these encouraging results, studies have shown that the large mass ratios required, often up to 5% [162], along with narrow effective frequency ranges, can render conventional TMDs as suboptimal solutions for vibration control and in turn, fatigue reduction in offshore wind turbines [141]. To address these shortcomings, researchers have explored a range of enhanced TMD configurations, such as active TMDs and semi-active TMDs. These systems incorporate actuators to modify their damping characteristics in real-time, enabling them to adapt to variations in structural response characteristics and overcome issues such as mistuning and

limited frequency bandwidths [163].

To assess the benefits of active TMDs with regard to fatigue life, Lackner and Rotea [164] explored the effect of both passive and active TMDs on the structural response of the NREL 5-MW offshore wind turbine mounted on a floating barge platform. The passive TMD achieved a 10% reduction in fatigue loads, while the active TMD yielded a significantly greater reduction of 30%. Arrigan et al. [165] similarly achieved significant reductions in tower-top response by installing semi-active TMDs within each of the three blades and an additional device in the nacelle of the wind turbine. The system employed adaptive tuning, with damper parameters updated every second based on real-time frequency identification through windowed Fourier analysis of the structural response. This configuration demonstrated particular effectiveness in mitigating flapwise blade vibrations, while simultaneously reducing overall tower oscillations. Similar performance gains were demonstrated by Fitzgerald, Basu, and Nielsen [166] using an Active Tuned Mass Damper (ATMD) for in-plane vibration control of rotating blades. The results showed response reductions of up to 44% with the use of an ATMD, surpassing the 31% achieved by a similar passive TMD. In a subsequent study, Fitzgerald, Sarkar, and Staino [167] applied a multi-modal wind turbine model to produce fragility curves for a wind turbine to assess its reliability when an ATMD was installed, which was controlled with a Linear Quadratic Regulator (LQR) control strategy. This fragility analysis showed that the addition of the ATMD significantly improved the reliability of the wind turbine tower under dynamic loading.

In addition to active and semi-active devices, several alternative damper configurations have been developed based on similar control principles. Karimi, Zapateiro, and Luo [168] proposed a H_∞ static feedback controller using a semi-active tuned liquid column damper (TLCD). By incorporating an electronically controllable valve, the system could adjust its natural frequency, resulting in highly damped turbine vibrations. Sarkar and Chakraborty developed state-feedback [169] and output-feedback [170] semi-active control strategies for magneto-rheological tuned liquid column dampers (MR-TLCDs) to control vibrations of onshore wind turbine towers. Fitzgerald and Basu [171] implemented a new control device, namely a cable-connected active tuned mass damper (CCATMD), in an attempt to mitigate in-plane blade vibrations. CCATMDs were installed in each of the turbine's three blades, as well as at the tower top and were found to outperform traditional TMDs,

particularly under highly turbulent wind conditions.

Collectively, these findings highlight the potential of active systems to overcome the limitations of passive TMDs, however, their implementation is often constrained by the significant power demands required to operate hydraulic or electromechanical actuators. Such systems may consume power in the kilowatt to megawatt range to generate the control forces necessary to modify damping and stiffness properties [172]. A promising damping device, which has the potential properties to overcome the limitations of the TMD without the need for an external power supply, is the TMDI.

2.5.2 Tuned Mass Damper Inerter

In 2002, Smith [173] introduced the inerter device that enabled the conceptualisation and later the development of the novel TMDI. The TMDI is a new variant of the classical TMD, which operates in a similar manner, with the addition of an inerter device. The inerter is a mechanical device that converts linear motion into high-speed rotational motion similar to that of a flywheel, as depicted in Figure 2.8. When placed in parallel to the spring and viscous dampers in a TMD, the inerter acts to amplify the effective mass of the TMD (see Figure 2.9). The presence of the inerter virtually increases the mass of the damper, leading to greater vibration control capabilities without the burden of increasing the mass ratio. As a result, the inerter enables one to achieve improved vibration control using a fixed mass ratio or equal structural control using a lighter damper, which is particularly beneficial for large flexible structures such as wind turbine towers. The inclusion of the inerter not only improves response mitigation at fixed mass ratios but also broadens the frequency range over which the device remains effective.

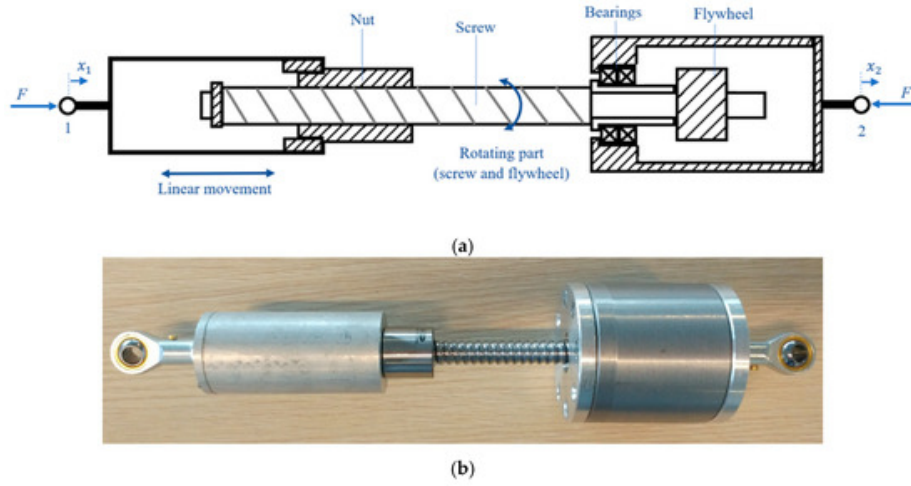


Figure 2.8: Inerter device [174]

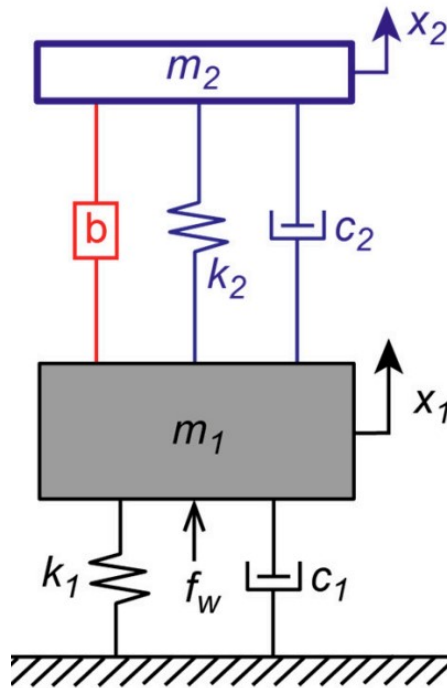


Figure 2.9: Tuned mass damper inerter configuration [144]

In recent years, inerter-based dampers have been extensively studied for their potential to reduce structural responses in vulnerable structures. Both experimental testing and numerical analyses have demonstrated the effectiveness of incorporating inerters, highlighting their benefits in enhancing overall vibration control [175, 176, 177, 178, 179, 180]. The positive results from studies such as these have encouraged the use of inerter-based damping devices for mitigating vibrational responses

of modern wind turbines. In one of the earliest contributions, Marian and Giaralis [181] proposed governing equations for the optimal design of a TMDI fitted in a stochastically support-excited structural system, highlighting its potential use in floating offshore wind turbines. Notably, their findings suggested that smaller mass ratios (less than 3%) can achieve greater response reductions than higher-mass configurations, making TMDIs particularly attractive in applications where added mass is constrained, such as wind turbine towers. Building upon this concept, Hu and Chen [182] successfully implemented a H_2 optimisation method to select the optimum parameters of an inerter-based dynamic vibration absorber (IDVA) in order to reduce the loads experienced by wind turbines under a range of rotational frequencies. Expanding on this work, Hu et al. [183] employed a TMDI damping device to investigate load mitigation in barge-type floating offshore wind turbines. The research used a simplified linear design model of NREL’s FAST-SC (structural control) [184] model to facilitate time-efficient parameter optimisation. The results revealed a trade-off between stroke space and fore-aft tower deflection but confirmed that for a given stroke limit, a TMDI could achieve superior structural control compared to conventional TMDs.

A significant contribution to the development of TMDIs for wind turbine applications was made by Sarkar and Fitzgerald [185] who developed closed-form expressions for optimum tuning and damping parameters for TMDIs installed in wind turbine towers. The study also reported that the vibration mitigation performance of TMDIs in floating offshore wind turbines tends to saturate beyond an inertance ratio of 0.4, with values in the range of 0.1–0.2 sufficient to deliver highly effective results. When installed in a spar-type offshore wind turbine and subjected to extreme loading conditions, a TMDI configured with these optimal parameters was found to reduce tower-top displacements by up to 53% in the side-to-side direction and 18% in the fore-aft direction, while large reductions of up to 90% in the TMDI’s stroke were observed in comparison with a conventional TMD. Villoslada, Santos, and Tomás-Rodríguez [186] similarly introduced a TMDI in the nacelle of NREL’s 5-MW wind turbine supported by a barge-type platform using a reduced degree of freedom (DOF) model. The results showed that the introduction of the TMDI was capable of reducing loads by 12% when neglecting stroke limits for the damping devices, however, this reduction is limited to 6% if stroke limits are considered.

A series of recent studies by Zhang and colleagues have further advanced the application of inerter-based damping systems in wind turbines. Initially focusing on blade vibration mitigation Zhang and Fitzgerald [187] implemented the Euler–Lagrange approach to construct a multi-modal model of a modern multi-megawatt wind turbine. Damping devices consisting of a TMD and TMDI were placed close to the tip of the three wind turbine blades and closed-form expressions were derived to calculate the optimal tuning and damping ratios of the damping devices. The inclusion of the inerter significantly reduced the edgewise vibrational response of the blades, while the stroke of the damping devices was reduced by 55% compared to the conventional TMD configuration. Extending this research, Zhang and Larsen [188] investigated an alternative damping device known as the Rotationally Inertial Damped Tuned Mass Damper (RIDTMD). This device, originally proposed by Garrido, Curadelli, and Ambrosini [189], is similar to a traditional TMD, however, the dashpot is replaced with a tuned viscous mass damper (TVMD). The study found that the RIDTMD provides significant reductions in edgewise blade vibrations in comparison with the conventional TMD at the expense of a slight increase in the damper stroke. In a subsequent study, Zhang and Høeg [190] implemented the RIDTMD in the nacelle of a spar-type floating offshore wind turbine. A six-degree-of-freedom model was used to perform parameter optimisation, and the performance of the optimised RIDTMD was compared with that of an optimally tuned conventional TMD. The results showed that the RIDTMD outperformed the TMD in reducing side-to-side tower displacements, again at the cost of a slightly increased stroke. Also exploring a novel inerter-based vibration control approach Zhang, Cao, and Dai [191] proposed an ungrounded tuned mass inerter system (TMIS) for vibration mitigation of wind turbine towers. The TMIS, consisting of a mass, spring and an inerter subsystem was found to provide increased vibration control for a given mass in comparison with traditional TMDS.

Following on from their earlier work with a conventional TMDI, Sarkar and Fitzgerald [192] proposed the use of a tuned mass damper fluid inerter in a spar-type floating offshore wind turbine in order to mitigate the vibrational response of the tower. The fluid inerter, due to its simplistic design and low maintenance costs, can reduce downtimes experienced by wind turbines with mechanical inerters. The TMDFI greatly outperforms traditional TMDs while rivalling the performance of traditional inerters. Continuing their research into inerter-based damping systems, and building on the work of Sarkar and Fitzgerald [185], Fitzgerald et al.

[193] investigated the effects of wind wave misalignment on the vibration capabilities of a TMDI installed in the nacelle of a spar-type offshore wind turbine. The damping device was found to greatly reduce the vulnerability of the wind turbine to wind and wave loading, particularly in cases in which the wind and waves were misaligned. This study further highlighted the superior performance of the TMDI compared to the conventional TMD. The enhanced mitigation capabilities demonstrated in this paper are consistently reflected throughout the studies discussed in this section, underscoring the significant advantages gained by incorporating an inerter into traditional TMD configurations. Collectively, these findings reinforce the potential of inerter-based systems to deliver more effective and reliable vibration control solutions for offshore wind turbine applications.

2.6 Conclusions

This literature review has highlighted the following key insights and gaps in current knowledge regarding offshore wind turbine fatigue:

- The accurate prediction of local stresses, particularly those arising from geometric discontinuities, is critical for reliable fatigue life estimation in wind turbine structures [194]. Such discontinuities, including welds, bolts, flanges and various geometric features act as stress concentrators, significantly influencing the initiation and propagation of fatigue cracks. However, the corresponding localised stress effects, which play a vital role in achieving accurate lifetime estimates, are often overlooked in global structural analyses, leading to non-conservative fatigue life estimates and, consequently, increased risk of premature failure.
- While frequency-domain methods are widely used for their computational efficiency, they are limited in their ability to capture the nonlinear behaviour inherent in wind turbine dynamics, such as those arising from control-structure interactions, which are essential for accurate response prediction [60] and, in turn, reliable fatigue assessment. Consequently, time-domain simulations, although more computationally demanding, provide superior capabilities for fatigue analysis by enabling a more realistic representation of turbine behaviour under operational conditions [70].

- FE models are highly effective at capturing detailed local stress fields required for fatigue assessment, however, they struggle to capture the complex interactions between the dynamic loading environment and the control strategies employed to pitch and yaw the wind turbine. In contrast, MBD models can be used to accurately capture the holistic response of wind turbines in an efficient manner, however, they lack the structural resolution required to provide detailed local stress time histories.
- Although several approaches have been proposed to improve fatigue modelling through hybrid or coupled methods, the practical application of these strategies is often limited. Accurate fatigue damage prediction requires a high number of simulations, and the computational cost of time-domain FE simulations renders such approaches infeasible without substantial simplifications [119]. As a result, these approaches either compromise result accuracy or become prohibitively expensive. To address this, researchers have employed SCFs in combination with MBD models to achieve holistic wind turbine responses while enhancing the accuracy of the resulting stress time histories. These SCFs can be estimated using empirical or analytical equations, but are most accurately determined through FE models. Furthermore, these SCFs can be utilised to account for corrosion wastage, an effect often overlooked in current design standards [137].
- The existing literature has demonstrated the potential for reducing structural responses and induced loading through the installation of TMDIs in the nacelles of wind turbine towers. The advantages of the mass amplification effect, which arises with the introduction of the inerter into damping devices such as the TMD, have been clearly illustrated, providing increased mitigation capabilities with decreased stroke space requirements. However, despite these promising results, none of the studies reviewed have investigated the potential application of TMDIs for reducing fatigue damage loads.

To address these limitations, this work proposes a new framework for the fatigue analysis of offshore wind turbines using a composite modelling approach. This approach integrates SCFs derived from a high-fidelity FE model with nominal stress time histories generated by a holistic MBD model. This MBD model is capable of capturing the complex interactions between the wind turbine response, the environmental loading and the control algorithm for the reference wind turbine.

The FE model addresses concerns regarding the accuracy of analytical or empirical formulas by accurately calculating the SCFs which arise due to the stiffening effect of an internally bolted ring flange connection at the base of the wind turbine tower, which acts to join the turbine tower to the transition piece.

The fatigue analysis in this work focuses on the butt weld which connects the wind turbine tower shell to the bolted flange connection. This butt weld is selected as similar welds at the base of turbine towers have been identified as the initiation sites of several costly turbine collapses. Additionally, its location in the splash zone makes it particularly vulnerable to corrosion. This vulnerability is captured in the FE model by incorporating the effects of corrosion-induced material loss, enabling the evaluation of how SCFs evolve over the turbine's service life and how this, in turn, impacts overall fatigue life.

This detailed fatigue analysis is carried out using conditions specified by NREL as a benchmark case and is extended to investigate two potential sites off the coast of Ireland, providing an example of a complete fatigue assessment framework for offshore wind turbine sites. Finally, the study addresses the current gap in understanding regarding the fatigue mitigation potential of TMDIs. To investigate this, a TMDI is integrated into the MBD model used within the composite modelling framework, enabling a comparative analysis of the turbine's fatigue life with and without the damping device.

Chapter 3

A Fatigue Analysis of the IEA 15-MW Reference Fixed-based Offshore Wind Turbine

3.1 Motivation

In March 2020, in association with NREL, the IEA released details of a state-of-the-art 15-MW monopile-based offshore wind turbine model developed as part of the IEA Wind TCP Task 37 [195]. This wind turbine serves as a benchmark for offshore wind energy research, providing a representative model of today’s modern wind turbines. The offshore location and large-scale dimensions of this wind turbine introduce an element of vulnerability to large fatigue loads, however, NREL stated that during the design process, no detailed fatigue load analysis was performed. In this chapter, we address this gap in knowledge by conducting a fatigue analysis on the tower of the IEA 15-MW reference wind turbine.

Fatigue analysis is a critical aspect of wind turbine structural assessments, ensuring the long-term reliability and safety of offshore wind energy systems. As highlighted in the literature reviewed in Chapter 2 and outlined in the IEC standard for wind energy generation systems [49], time-history-based fatigue analysis is a widely adopted approach for evaluating fatigue damage in wind turbine structures and is implemented in this study. This method involves generating stress time histories at key structural locations subjected to environmental loading. Such time

histories can be processed using the rainflow counting algorithm and the fatigue life subsequently estimated using S-N curves in conjunction with Palmgren-Miner's rule. Fatigue analysis of offshore wind turbines requires numerous simulations across a wide range of operational and environmental conditions in order to capture the stochastic nature of wind and wave loading. These simulations must capture the complex interaction between aerodynamic, hydrodynamic, and operational forces acting on the turbine structure. Consequently, the fidelity of the selected modelling technique plays a crucial role in accurately representing the structural response while balancing computational efficiency. In this study, an MBD modelling approach is employed due to its capability to simulate the coupled aero-hydro-servo-elastic behaviour of the wind turbine in an efficient manner.

The nonlinear aeroelastic MBD model developed on MATLAB is instantiated with details from the IEA Wind TCP Task 37 [195]. Using this MBD model, stress time histories are generated at the butt weld connecting the base of the tower to the transition piece, a critical location for fatigue assessment as established in Chapter 2. The extracted stress histories are post-processed to evaluate the fatigue damage using bi-linear S-N curves. The results from this analysis provide insights into the fatigue performance of the reference wind turbine under realistic offshore operating conditions. To obtain such realistic operational conditions, accurate environmental parameters are essential. The IEA 15-MW wind turbine was designed subject to wind and met-ocean conditions based on a site on the east coast of the U.S. These met-ocean conditions have been published to provide a standard basis of comparison for the offshore wind energy research community and thus are utilised in this study. By incorporating these benchmark conditions, the fatigue assessment performed in this chapter enables comparative analysis and provides a control basis for subsequent investigations in the following chapters.

In this chapter, the IEA 15-MW wind turbine is introduced, along with the formulation of the MBD model and the fatigue analysis process. Given the extensive literature on MBD-based structural modelling and time-history-based fatigue analysis, this chapter provides a high-level overview of the methodologies employed, focusing on the conceptual frameworks. For the MBD model, the key components, their interactions, and the types of dynamic responses simulated are outlined while omitting detailed kinematic derivations. Readers seeking a more detailed understanding of such derivations and the detailed modelling process are referred to the

work of Sarkar and Fitzgerald [80] and Jonkman [196]. Regarding the fatigue analysis, this chapter details the selected environmental parameters, the simulations performed, and the selection and implementation of S-N curves with Palmgren-Miner's rule. These analyses are conducted using the IEA 15-MW wind turbine, with its design specifications and operational characteristics discussed in the following section.

3.2 Details of the IEA 15-MW Wind Turbine

The design and specifications of the state-of-the-art 15-MW offshore reference wind turbine were released by the IEA in association with the NREL [195] to act as a benchmark model within the research community. The IEC Class 1B 15-MW wind turbine is a three-bladed upwind design with a hub height of 150 meters and a rotor diameter of 240 meters. The turbine is supported by a monopile foundation suitable for deployment in 30-meter-deep waters, with an embedment depth of 45 meters. With an estimated annual energy production of 77.4 GWh, it highlights the substantial renewable energy potential of such large-scale wind turbines. The wind turbine is a collective pitch-variable speed design, allowing it to operate across a range of rotor speeds, which are controlled by adjusting the single pitch angle of the three blades. It operates from a cut-in wind speed of 3 m/s to a cut-out wind speed of 25 m/s, reaching its rated power output at a wind speed of 10.59 m/s. Additional specifications of the turbine, as outlined by the IEA, are provided in Table 3.1.

Parameter	Value
Turbine class	IEC Class 1B
Specific rating	332 W/m ²
Rotor orientation	Upwind
Number of blades	3
Rotor diameter	240 m
Hub height	150 m
Blade length	117 m
Blade mass	65,250 kg
Root diameter	5.2 m
Max chord	5.77 m
Tip prebend	4 m
Precone	4 deg
Cut-in wind speed	3 m/s
Rated wind speed	10.59 m/s
Cut-out wind speed	25 m/s
Design tip-speed ratio	90
Minimum rotor speed	5 rpm
Maximum rotor speed	7.56 rpm
Maximum tip speed	95 m/s
Design tip-speed ratio	9
Drivetrain	Direct drive
Shaft tilt angle	6 deg
Rotor nacelle assembly mass	1,017 t
Transition piece height	15 m
Monopile embedment depth	45 m
Monopile base diameter	10 m
Tower mass	860 t
Monopile mass	1,318 t

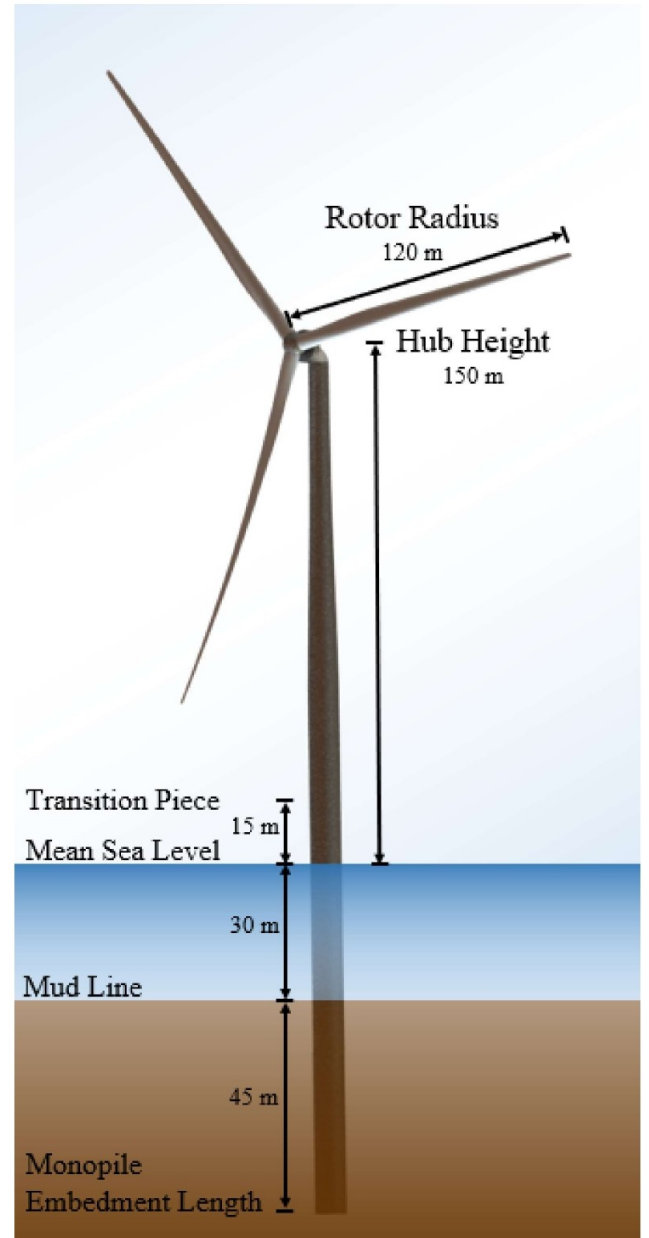


Table 3.1: IEA 15-MW wind turbine specifications- Figure 3.1: IEA 15-MW offshore reference wind turbine [195]

3.3 Multibody Dynamic Model

A nonlinear aeroelastic MBD model is employed to capture the operational dynamic response of the wind turbine under environmental loading. This MBD model was previously developed within the Trinity research group and is based on the open-source framework of OpenFAST, implemented through an in-house code. While a range of commercial and open-source software tools for multibody dynamic simulation of wind turbines are available, the in-house implementation provides greater flexibility, enabling model modifications and detailed exploration of key modelling parameters. In this thesis, the model is specifically employed to calculate the axial forces and bending moments occurring at the connection between the turbine tower and the transition piece. In this section, the construction and underlying concepts of the MBD model are discussed, detailing key aspects such as DOFs considered, boundary conditions and load generation.

3.3.1 Characterisation of Rigid and Flexible Bodies

The MBD approach enables key components to be modelled as rigid bodies (e.g. nacelle and hub) or as flexible members (e.g., tower, blades, platform, and low-speed shaft).

Rigid bodies are components in which the relative positions of their constituent particles remain unchanged when subjected to external forces. As a result, these components do not deform and retain their geometric shape under load. This idealisation simplifies real-world behaviour by restricting such components to rigid body motion, allowing only translation and rotation as a whole. This assumption is applicable to components that are sufficiently stiff or in cases where deformation is negligible within the context of the overall system dynamics.

Flexible bodies, in contrast to rigid bodies, are components in which the relative positions of their constituent particles change when subjected to external forces. Unlike rigid bodies, these components undergo deformation, resulting in relative displacements such as bending, twisting, stretching, and compression. The extent of these deformations depends on several factors, including the geometric shape of the component, its material properties, and the magnitude, direction, and distribution of the applied load. This ability to deform makes flexible bodies more

representative of real-world behaviour in cases where the flexibility significantly influences the overall system response.

The analysis of flexible bodies often involves complex mathematical formulations, such as finite element methods, to capture the intricate interactions between forces, material behaviour, and geometric changes, however, these calculations can be extremely computationally expensive. In this model, the modal summation method is used to calculate the dynamic responses of the flexible components of the wind turbine, specifically the turbine blades, drivetrain, and tower. This method models the complex dynamics of a flexible component by representing its response as a summation of its fundamental mode shapes, each scaled by a time-dependent modal coordinate. Each mode shape characterises a unique deformation pattern of the component associated with a corresponding natural frequency. By selecting the most significant ‘ n ’ mode shapes, the complex response to external loads can be effectively approximated using only ‘ n ’ DOFs, reducing computational complexity while maintaining accuracy. The number of mode shapes considered directly affects the precision of the solution, making it essential to carefully select an appropriate number for each component.

3.3.2 Multibody Dynamic Model Degree of Freedom Formulation

The MBD model developed in this work employs a total of 22 DOFs to capture the motion of both the rigid and flexible turbine components of the wind turbine. For the purposes of this model, the tower is defined to include the transition piece and the portion of the monopile above the seabed. The remainder of the monopile is modelled as a rigid body characterised using six DOFs, three translational (q_{Sg} , q_{Sw} , q_{Hv}) and three rotational (q_R , q_P , q_Y). In this thesis, a fixed-base condition is applied at the mudline, constraining these six DOFs and effectively neglecting soil–structure interaction. While this assumption generalises the system and reduces the computational complexity, it also introduces limitations by neglecting the flexibility and damping effects of the surrounding soil. By introducing the fixed-base constraint, the monopile behaves as a perfectly rigid foundation, resulting in an artificially stiffer system with higher natural frequencies compared to a more realistic flexible foundation. Additionally, by neglecting the soil structure interaction, the energy dissipation through both material and radiation damping of the surrounding

soil mass is disregarded in the model. These effects influence the dynamics of the turbine, reducing the accuracy of the predicted structural response.

Above the mudline, the deformation response of the turbine tower is modelled with four DOFs, based on the first two mode shapes in both the side-to-side (q_{TSS1} , q_{TSS2}) and fore-aft (q_{TFA1} , q_{TFA2}) directions. These modes capture the dominant bending dynamics of the tower, while axial and torsional deformations are neglected. Due to the rigid nature of the turbine hub and nacelle, only two DOFs are required to describe their motion. The rotation of the rotor nacelle assembly about the longitudinal axis of the turbine tower, enabling the turbine to align with the wind direction, is described as the nacelle yaw motion q_{yaw} . Additionally, the rotational position of the hub/rotor blades is assigned as the generator azimuth q_{GeAz} or rotor angle. The torsion of the low-speed shaft is considered using one DOF q_{DrTr} , with its flexibility modelled using a spring-damper system. Finally, the elastic deflections of the three blades are represented by nine DOFs. Each blade's elastic response is described based on the first and second flapwise mode shapes (q_{BiF1} , q_{BiF2}) and the first edgewise mode shape (q_{BiE1}), where i represents the blade number.

3.3.3 Kane's Dynamics

The 22 DOF MBD model is developed using Kane's Dynamics [197]. Kane's method was selected for its vector-based formulation, which provides a systematic and efficient way to derive the equations of motion for MBD systems. This approach is particularly advantageous for modelling the intricate interactions among the numerous components of a wind turbine, making it well-suited for implementation in computational tools like MATLAB in comparison with more traditional approaches such as Euler-Lagrange and D'Alembert's principle.

Kane's dynamics enables the generalised active forces and generalised inertial forces for each of the turbine's components to be first described in a local reference frame of the given component. These localised forces can subsequently be expressed in the global or inertial reference frame via multiplication with a transformation matrix. Components which deform or rotate about one axis at a time can be transformed from the coordinates in their local reference frame into the global reference frame using Euler rotation, while components which rotate or deform about multiple axes simultaneously are translated using small-angle approximation with

singular value decomposition. The transformation matrices are composed based on the geometric properties of the turbine, in addition to combinations of rotations and displacements about the 22 DOFs employed to describe the motion of the turbine components.

Stemming from Newton's laws of motion, the generalised equilibrium equations for a simple holonomic multi-body system, as derived from Kane's dynamics, can be expressed as:

$$F_r + F_r^* = 0 \quad \text{for } r = 1, 2, \dots, N \quad (3.1)$$

where N denotes the total number of DOFs in the system. The term F_r is known as the *generalised active force* corresponding to the r^{th} DOF of the system. It is determined for a set of n rigid bodies, each defined by a reference frame N_i and centre of mass point X_i , using the following expression:

$$F_r = \sum_{i=1}^n [{}^E \mathbf{v}_r^{X_i} \cdot \mathbf{F}^{X_i} + {}^E \omega_r^{N_i} \cdot \mathbf{M}^{N_i}] \quad (3.2)$$

$\mathbf{F}^{X_i}$ is the force vector acting on the centre of mass of point X_i and $\mathbf{M}^{N_i}$ is the moment vector acting on the N_i rigid body. The terms ${}^E \mathbf{v}_r^{X_i}$ and ${}^E \omega_r^{N_i}$ represent the partial linear and partial angular velocity of the point X_i and rigid body N_i respectively, associated with the r^{th} DOF in the inertial (E) reference frame. The *generalised inertia force* corresponding to the r^{th} DOF is given as:

$$F_r^* = - \sum_{i=1}^n \left[{}^E \mathbf{v}_r^{X_i} \cdot (m^{N_i E} \mathbf{a}^{X_i}) + {}^E \omega_r^{N_i} \cdot {}^E \dot{\mathbf{H}}^{N_i} \right] \quad (3.3)$$

where it is assumed that the inertia forces for each rigid body N_i are applied at its centre of mass point, denoted as X_i . The term ${}^E \dot{\mathbf{H}}^{N_i}$ refers to the time derivative of the angular momentum of the rigid body N_i around its centre of mass X_i within the inertial reference frame E . Further details regarding these general equations are outlined by Kane and Levinson in their seminal work on dynamics [197].

By taking the time derivatives of the system's generalised coordinates, the partial linear and angular velocities of key points and rigid bodies can be determined. These velocities are then used to formulate the complete non-linear time-domain equations of motion for the fixed-base wind turbine, expressed in their general form

as:

$$\mathbf{M}(\mathbf{q}, \mathbf{u}, t) \ddot{\mathbf{q}} = -\mathbf{f}(\dot{\mathbf{q}}, \mathbf{q}, \mathbf{u}, t) \quad (3.4)$$

Here, $\mathbf{M}$ represents the inertial mass matrix, which is a non-linear function of the system's DOFs $\mathbf{q}$, control inputs $\mathbf{u}$, and time t . The force vector $\mathbf{f}$ similarly depends non-linearly on the DOFs, control input and time in addition to the time derivative of the DOFs $\dot{\mathbf{q}}$. The numerical simulations performed in this work were conducted using MATLAB[®] [198], where this non-linear system of equations was solved using the Runge-Kutta 4th order method, commonly referred to as 'ODE 4'. The DOFs obtained from these simulations were subsequently used to calculate the axial forces and bending moments at the base of the wind turbine tower.

3.3.4 Generalised Forces

The generalised inertial forces, presented in Equation 3.5 (see Appendix A for definitions), arise in the wind turbine model from the distributed masses of its various components. The masses of both rigid and flexible components, including the monopile, tower, nacelle, hub, blades, and generator, contribute to the generalised inertial forces:

$$F_r^* = F_r^*|_X + F_r^*|_T + F_r^*|_N + F_r^*|_H + F_r^*|_{B1} + F_r^*|_{B2} + F_r^*|_{B3} + F_r^*|_G \quad (3.5)$$

The generalised active forces in the wind turbine system, presented in Equation 3.6 (see Appendix A for definitions), encompass forces applied directly to its components, internal forces within flexible members, and constraint relationships between rigid bodies:

$$\begin{aligned} F_r = & F_r|_{\text{AeroT}} + F_r|_{\text{AeroB1}} + F_r|_{\text{AeroB2}} + F_r|_{\text{AeroB3}} + F_r|_{\text{GravT}} + F_r|_{\text{GravN}} \\ & + F_r|_{\text{GravH}} + F_r|_{\text{GravB1}} + F_r|_{\text{GravB2}} + F_r|_{\text{GravB3}} + F_r|_{\text{YawStiff}} \\ & + F_r|_{\text{YawDamp}} + F_r|_{\text{Gen}} + F_r|_{\text{Brake}} + F_r|_{\text{ElasticT}} + F_r|_{\text{ElasticB1}} \\ & + F_r|_{\text{ElasticB2}} + F_r|_{\text{ElasticB3}} + F_r|_{\text{DampT}} + F_r|_{\text{DampB1}} + F_r|_{\text{DampB2}} \\ & + F_r|_{\text{DampB3}} + F_r|_{\text{ElasticDrive}} + F_r|_{\text{DampDrive}} \end{aligned} \quad (3.6)$$

These forces arise from various sources, including gravitational effects, generator torque, and external loads acting on both rigid and flexible components. The external forces applied to the wind turbine model include contributions from both

aerodynamic and hydrodynamic loading. The resulting internal forces within flexible members, generated by deflections, contribute restoring forces to the generalised active forces. The corresponding damping forces in the blades and tower are calculated using a modal damping model, where the damping is assumed to be proportional to the stiffness of each mode. For the drive train, stiffness-proportional damping is applied, while viscous damping is used for the yaw motion, accounting for energy dissipation during system operation. Constraint relationships between bodies, such as the yaw spring and damper connecting the tower and nacelle, also play a role as they undergo work during motion. However, rigid connections, which do not perform work, do not contribute to these forces.

As many of the generalised inertial and active forces are highly dependent on both time and the dynamic behaviour of the system's components. To accurately capture these complex interactions, the MBD model is developed using a modular framework. In this approach, individual modules are designed to handle specific mathematical calculations which can then be interconnected enabling full coupling of the global system. The hydrodynamics module is used to model the sea conditions experienced by the wind turbine based on the Pierson–Moskowitz spectrum [199], which characterises sea states by the significant wave height and peak spectral period. This module employs the Airy wave theory to account for wave and current interactions, allowing for two-dimensional flow to be considered. The linear Airy wave theory assumes: waves to be sinusoidal, that the wave height is much shorter than the length or period of the wave and that the wave height is small in comparison with the depth of the water. The hydrodynamic forces acting on the monopile and tower are estimated using Morison's equation [200], combined with strip theory. This approach is valid under the condition that the wavelength exceeds five times the diameter of the structure, a criterion satisfied in this work. The hydrodynamic loads are calculated with specified model constants of $C_M = 1.969$ for the inertia coefficient and $C_D = 0.6$ for the drag coefficient [195]. This model neglects radiation damping, as the energy dissipation due to wave generation by the structure's motion is considered negligible compared to the effects of viscous drag.

In the aerodynamic module, the loads on the turbine blades are calculated using Blade Element Momentum (BEM) theory implementing wind profiles generated via TurbSim. TurbSim [201] is a software package distributed by NREL that utilises

spatial coherence and wind shear to generate turbulent 3-D wind fields. Turbsim implements the IEC's [49] Kaimal spectral turbulence model [202] to model the atmospheric turbulence with each wind file being generated with a unique random seed. These wind files are supplied to the BEM theory module which combines blade element theory, which calculates lift and drag forces on an airfoil, with Momentum Theory, which is based on the conservation of momentum. The resulting equations are solved using a root-finding method proposed by Ning [203]. This solution solves for the inflow angle rather than the axial and tangential induction factors avoiding the requirement of a 2D iterative solution approach. To enhance the accuracy of the aerodynamic analysis, Prandtl's hub and tip loss correction factors are implemented to account for vortices generated by the blade tips, while the Pitt and Peters correction addresses the influence of skewed inflow. Once the aerodynamic calculations are complete, the blade pitch is then calculated using NREL's Reference OpenSource Controller (ROSCO) [204].

The MBD model, incorporating these calculations within its modular framework, has been rigorously validated through a code-to-code comparison with the state-of-the-art wind turbine simulator OpenFAST [205], provided by NREL. This validation demonstrated strong agreement between the models, with detailed results outlined in Baisthakur and Fitzgerald [206] and Baisthakur and Fitzgerald [207]. For a comprehensive understanding of the derivations of the formulas implemented in this MBD model, readers are referred to Sarkar and Fitzgerald [80], while additional details on the modelling can be found in Sarkar et al. [208] and Sarkar and Fitzgerald [185].

3.4 Fatigue Analysis

The fatigue analysis is conducted adhering to the guidelines outlined in IEC 61400-1 [49], performing simulations via the MBD model discussed in Section 3.3. The MBD model is used to calculate the axial forces and bending moments at the base of the wind turbine tower, applying a load partial safety factor of $\gamma_f = 1$ across all simulations. In this study, simulations are performed for IEC Design Load Case (DLC) 1.2 exclusively, while other load cases, such as DLCs 2.4, 3.1, 4.1, and 6.4 are neglected. DLC 1.2 represents typical power production conditions for the wind turbine for which a normal turbulence model can be assumed. This focus on DLC 1.2 is justified within the scope of this thesis as studies such as Huo and Tong

[39] indicate that the majority of fatigue damage in wind turbines occurs during normal operation. However, it should be acknowledged that several studies [209, 210] have demonstrated that non-operational conditions, such as start-up, shut-down, idling, fault, and storm events, can also contribute significantly to the overall fatigue life of offshore wind turbines. Consequently, while DLC 1.2 captures the dominant operational fatigue behaviour, the exclusion of non-operational load cases may lead to an overestimation of fatigue life and therefore represents a limitation of the present work.

In line with IEC standards, the selected met-ocean conditions must adequately represent the full long-term distribution experienced by the wind turbine. For this study, the met-ocean conditions were selected based on data from NREL for a site on the east coast of the United States [211] with all wind and wave loadings assumed to be fully aligned. For DLC 1.2, environmental conditions are selected based on discrete wind speeds, which are subsequently paired with suitable met-ocean conditions. The IEC requires that the wind speeds must be selected from the range of cut-in to cut-out wind speeds with a resolution of at least 2 m/s and that at least six 10-minute simulations of usable simulation data for each environmental condition is implemented. Consequently, each simulation in this study was conducted for a duration of 675 seconds, with the initial 75 seconds discarded in order to eliminate the effects of the initial transient phase. For each environmental condition, six wind files with unique random seeds were paired with corresponding wave files, which were also generated using random seeds.

3.4.1 Benchmark Environmental Conditions

The environmental parameters for this fatigue analysis were adopted from Stewart et al. [211]. These parameters, provided by NREL, are representative of a site on the east coast of the United States. These environmental conditions were employed in the design of the IEA 15-MW wind turbine and were made available to the offshore wind energy research community as a standardised set of environmental conditions. These conditions accurately characterise the physical behaviour of offshore environments and serve as a benchmark for comparison within the research community.

The environmental data was sourced from the National Oceanic and At-

mospheric Administration (NOAA), collected through a network of floating buoys located along the U.S. coastlines and in surrounding international waters. NREL processed this raw dataset to derive wind speed parameters at a 150-meter hub height, suitable for the 15-MW wind turbine. The data was provided at a resolution of 2 m/s, as detailed in Table 3.2. For each wind speed, corresponding met-ocean conditions were defined, including significant wave height and peak spectral wave period, to comprehensively characterise the environmental conditions at the site.

Table 3.2: NREL U.S. East Coast environmental parameters

Wind Speed (m/s)	Significant Wave Height (m)	Peak Spectral Period (s)
4	1.10	8.52
6	1.18	8.31
8	1.32	8.01
10	1.54	7.65
12	1.84	7.44
14	2.19	7.46
16	2.60	7.64
18	3.06	8.05
20	3.62	8.52
22	4.03	8.99
24	4.52	9.45

The probability distribution of the environmental conditions was modelled based on the wind speeds using a two-parameter Weibull distribution, with a shape factor of 2.12 and a scale factor of 9.767. Turbulence intensity data corresponding to the wind speeds was not provided. To address this, a turbulence model proposed by Wang et al. [212] and recommended by the IEC [213] was employed to establish a relationship between wind speed and turbulence intensity, as illustrated in Figure 3.2. A site of class "A" was selected to provide conservative results for this model, reflecting higher turbulence conditions.

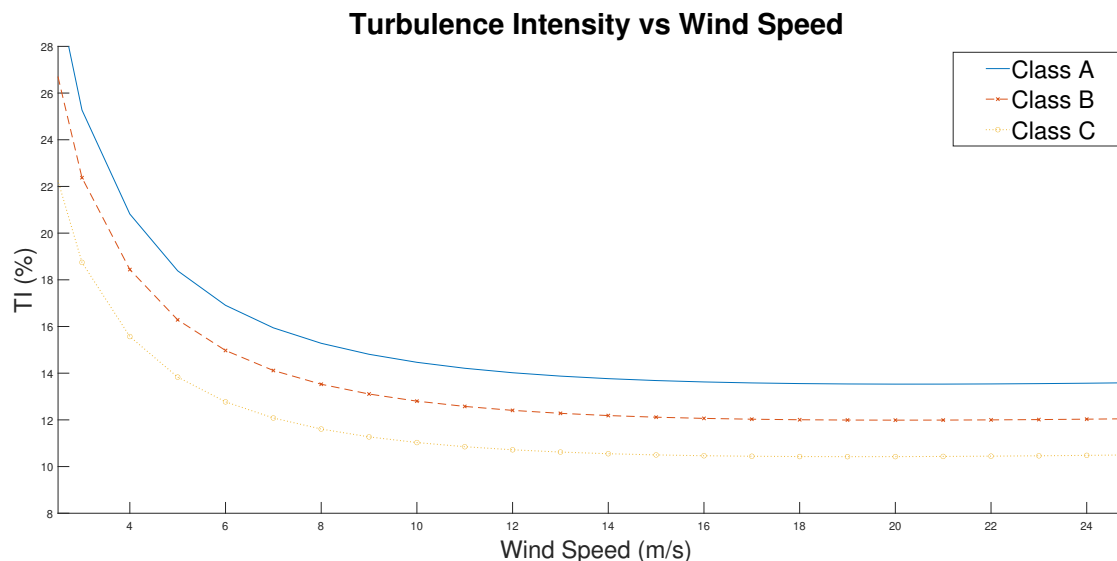


Figure 3.2: Turbulence intensity vs. wind speed proposed by Wang et al. [212]

To facilitate reproducibility and enable consistent comparisons under these environmental conditions, NREL provided six randomised turbulence seeds alongside the detailed environmental parameters, as outlined in Table 3.3. These random seeds were implemented as inputs into the Turbsim software [201] to generate the 3-D wind fields.

Table 3.3: NREL metocean condition random turbulence seeds

Simulation	Random Turbulence Seed
1	-1811332072
2	13173288
3	-1472715972
4	2098744693
5	185130004
6	-802506215

3.4.2 Evaluation of Time-Varying Stress in the Tower Weld Joint

The fatigue analysis focuses on a proposed butt weld acting to connect the turbine tower to the transition piece. This weld is denoted as “proposed” as the IEA documentation for the turbine lacks specific details regarding the precise locations and configurations of connection joints within the support structure. This location

was selected as it is a critical connection for maintaining the structural integrity of the tower and is highly susceptible to fatigue damage, as highlighted in Chapter 1.

Employing the environmental conditions detailed in Section 3.4.1, the MBD model is used to generate corresponding time histories of the axial forces and bending moments experienced at the tower base. The shear forces at the tower base are neglected as the resulting shear stresses are significantly lower than those arising from the axial and bending stresses [41]. Additionally, across the full range of environmental conditions, the axial forces exhibited minimal variation over time and were therefore considered constant in the calculations. Using Equation 3.7, stress time histories were derived from the axial force and bending moments based on the cross-sectional area and section modulus at the tower base.

$$\sigma_E(t) = \frac{-N}{A} + \frac{M_{FA}(t) \cos(\theta)}{Z} + \frac{M_{SS}(t) \sin(\theta)}{Z} \quad (3.7)$$

In this equation, $\sigma_E(t)$ is the stress at the tower base at a time t during simulations conducted under environmental conditions E , N is the axial force, A is the area of the tower cross-section, M_{FA} and M_{SS} are the bending moments in the fore-aft and side-to-side directions and θ represents the angle of the resultant moment vector relative to the fore-aft direction, given by:

$$\theta = \tan^{-1} \left(\frac{M_{SS}(t)}{M_{FA}(t)} \right) \quad (3.8)$$

The section modulus Z is derived from the second moment of area I for a hollow circular cross-section:

$$Z = \frac{D_{outer}}{2I}, \quad I = \frac{\pi (D_{outer}^4 - D_{inner}^4)}{64} \quad (3.9)$$

Here, D_{outer} and D_{inner} are the outer and inner diameters of the tower cross-section, respectively. This formulation calculates the maximum stress observed at the tower base at any given time. As the magnitudes of the fore-aft and side-to-side moments vary over time, the location of the maximum stress also shifts. To adopt a conservative approach, the maximum stress value from the entire time history is applied to a single location.

3.4.3 Palmgren-Miner's Rule with Rainflow Counting Algorithm

The post-processing of the simulation results was conducted using a modified version of NREL's software package, MLife [214]. MLife performs short-term fatigue analysis by processing the time histories of each set of wind speeds coupled with their corresponding wave conditions. To calculate short-term fatigue, MLife employs Palmgren-Miner's rule combined with the Goodman correction, in conjunction with the rainflow counting algorithm as specified in Annex G of IEC 61400-1 [49]. For each environmental condition, the six random seeds are evaluated to determine the total fatigue damage accumulated over 3600 (6x600) seconds. The key modifications made to MLife in this work included adding the ability to perform fatigue analysis using bi-linear S-N curves and enabling the user-defined distribution of environmental parameters to account for both the wind and wave conditions.

The user-defined distribution is used to estimate the frequency of the various environmental conditions over the operational life of the turbine. To extend the short-term fatigue results into long-term fatigue results, the short-term fatigue damage values are integrated using the estimated frequencies to find the long-term damage accumulated. This analysis was applied on an annual basis, calculating the accumulated fatigue damage rate at the turbine tower connection butt weld for an individual year. Following the concept of partial damage encapsulated in Palmgren-Miner's rule, the component is considered to have failed when the cumulative damage equals unity. To determine the lifespan of the turbine tower, the reciprocal of the damage rate is calculated.

3.4.4 S-N Curves for Butt Weld Connections

To calculate the fatigue damage, an S-N curve is required to be defined. Given that the turbine tower sections are assumed to be prefabricated in an off-site manufacturing facility, the welding procedure is assumed to be performed in the flat position. Consequently, the DNV recommended practice standard class D S-N curves [101] can be implemented for the butt weld connecting the tower and transition piece. Specifically, the class D curve for butt welds in steel structures exposed to seawater with cathodic protection. This S-N curve is designed with a 97.7% probability of

survival, and its properties are detailed in Table 3.4¹.

Table 3.4: DNV SN curve parameters for steel structures exposed to seawater with cathodic protection [101]

Condition	m1	$\log \bar{a}_1$	m2	$\log \bar{a}_2$	Fatigue Limit (MPa)	k
Seawater (cathodic protection)	3	11.764	5	15.606	83.41	0.2

In this table, m1 is the negative inverse slope of the first segment of the bilinear S-N curve, m2 is the negative inverse slope of the second segment of the bilinear S-N curve, $\log \bar{a}_1$ is the intercept of the first segment of the S-N curve with the $\log N$ axis, $\log \bar{a}_2$ is the intercept of the second segment of the S-N curve with the $\log N$ axis and k is the thickness exponent.

As this S-N curve is constructed based on a reference plate with a thickness of 25 mm, a thickness effect correction must be applied via a modification of the stress range as presented in Equation 4.59 before use.

$$\text{Log}N = \log \bar{a} - m \log \left(\Delta \sigma \left(\frac{t}{t_{ref}} \right)^k \right) \quad (3.10)$$

in which $\Delta \sigma$ is the stress range, N is the predicted number of cycles to failure for the stress range, t_{ref} is the reference thickness equal 25 mm and t is the thickness of the component.

3.5 Fatigue Life Evaluation

The fatigue analysis conducted for the butt weld connection between the wind turbine tower and the transition piece yielded a fatigue damage rate of 0.0318 per annum, based on the environmental conditions proposed by NREL and the S-N curves provided by DNV. This damage rate corresponds to a fatigue life of 31.45 years, demonstrating that the design of the butt weld connection is robust, providing sufficient durability for long-term operation under the expected loading conditions. The expected lifespan of 31.45 years significantly exceeds the typical 20-year design life of modern offshore wind turbines. The cumulative fatigue damage after 20 years of operation is estimated to be 0.636, indicating that the structure retains a margin

¹ There is a discrepancy between the tables and charts in DNV GL, the chart values are deemed correct and thus used in this work.

of safety in excess of 1.5. This suggests that the wind turbine tower is not only capable of withstanding the anticipated operational and environmental loadings over its design life but also presents opportunities for lifetime extension.

The fatigue damage associated with each individual environmental condition can be further assessed in Figure 3.3. In this figure, the fatigue damage arising from each of the environmental conditions across the 6 seeds is presented. The pattern presented in this plot is reflective of the typical operational regions for variable speed wind turbines as presented in Figure 3.4 adopted from Mousavi, Bevan, and Kucukdemiral [215]. The fatigue damage rates exhibit distinct trends across different operational wind speed regions. In Region 2, where the turbine operates between its cut-in wind speed (3 m/s) and its rated wind speed (10.69 m/s), fatigue damage increases significantly as wind speeds approach the rated wind speed. This is due to the higher aerodynamic loads experienced by the turbine as it aims to generate maximum power output for a given wind speed. At the rated wind speed (10.69 m/s), the pitch control system activates, feathering the blades to reduce aerodynamic loads. This results in an initial drop in fatigue damage rates, however, as wind speeds continue to increase beyond this point, fatigue damage rates rise again. This secondary increase is attributed to the growing dynamic loads caused by the higher wind speeds, which counteract the load-reducing effects of blade pitching.

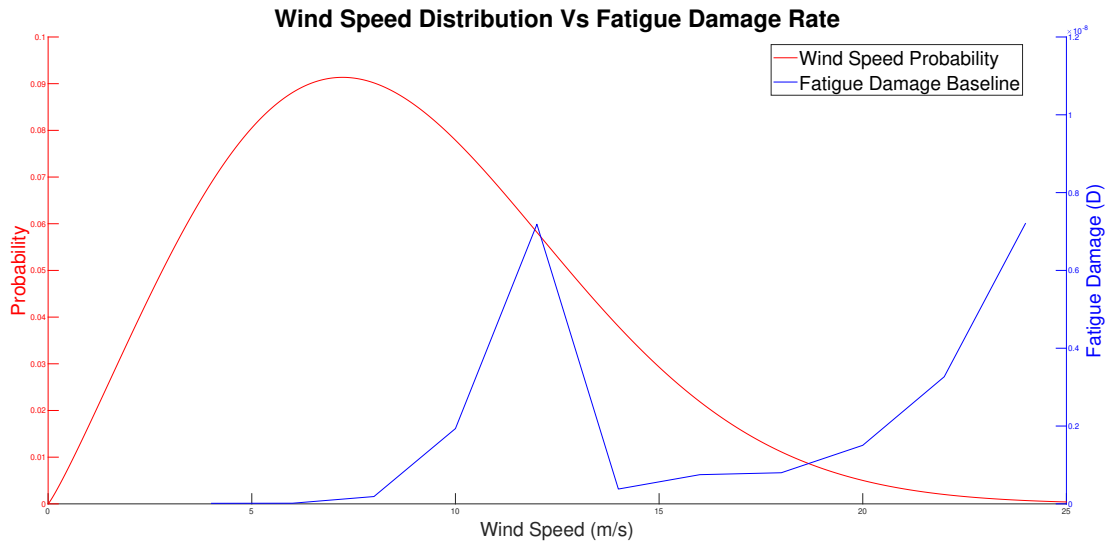


Figure 3.3: Fatigue damage rate vs. wind speed

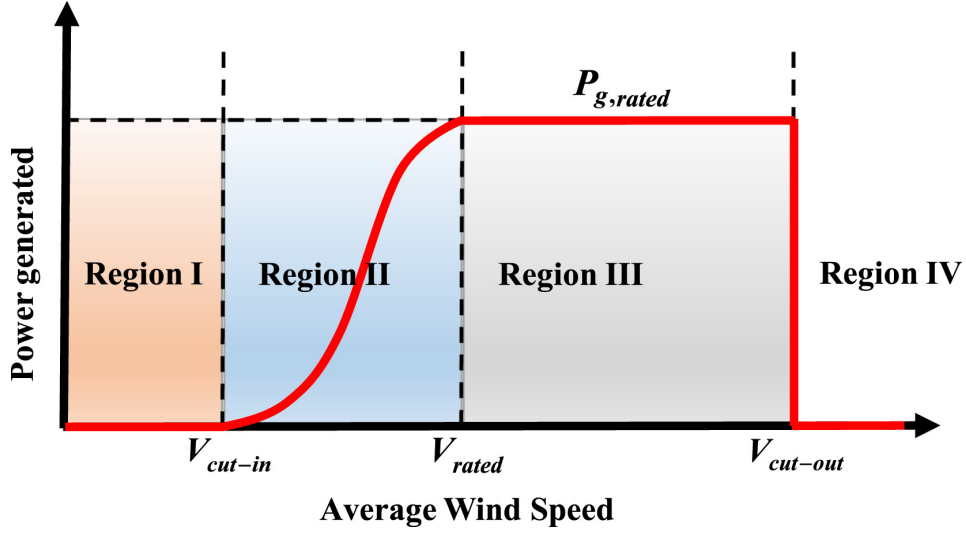


Figure 3.4: Variable speed wind turbine operational regions [215]

By overlaying the fatigue damage rates with the wind speed probability density, the critical environmental conditions contributing to significant fatigue damage over the wind turbine's lifespan can be identified. Specifically, wind speeds between 6 m/s and 14 m/s are found to be the most critical. In this range, the turbine experiences substantial aerodynamic loading, leading to elevated fatigue damage rates, while these wind speeds also occur with high frequency, as reflected in the wind speed probability density. This combination makes these wind speeds particularly influential on the long-term fatigue life of the wind turbine.

3.6 Conclusion

This chapter uses the current state-of-the-art procedure to calculate the expected fatigue life of a butt weld connecting the tower base and transition piece of the state-of-the-art IEA 15-MW offshore wind turbine. Stress time histories at this critical connection were generated using a fully coupled 22 DOF aero-hydro-servo-elastic MBD model of the 15-MW wind turbine. This MBD model was employed to capture the holistic behaviour of the turbine under operational conditions. The environmental conditions considered for this analysis were based on a site located on the east coast of the United States, provided by NREL. The simulations were performed in accordance with IEC standards, producing six 10-minute simulations for each of the 12 conditions considered. The fatigue life was estimated using a

modified version of NREL’s MLife incorporating a bi-linear S-N curve for steel exposed to seawater with corrosive protection, as specified by DNV guidelines. The long-term fatigue damage was calculated with the Palmgren/Miners rule based on a Weibull probability distribution with a shape factor of 2.12 and a scale factor of 9.767.

The fatigue life of the wind turbine butt weld connection was calculated to be 31.45 years, based on a yearly damage rate of 0.0318. This lifespan exceeds the typical 20-year life expectancy for modern wind turbines, presenting potential opportunities for lifetime extension. In particular, this analysis identified wind speeds between 6 m/s and 14 m/s to be the most critical environmental conditions for the calculation of fatigue damage for the IEA 15-MW wind turbine, as they combine high aerodynamic loading with high occurrence probability.

The results and methodology presented in this chapter serve as a control study within the thesis. By performing the fatigue analysis in accordance with the IEC 61400 design standard [213] and using S-N curves adopted from the DNV standards [216], this approach reflects the state-of-the-art industry practices, while also ensuring the analysis is reproducible and accessible to other researchers. The use of the IEA 15-MW reference offshore wind turbine as the basis for this analysis offers a standardised representation of modern wind turbines, widely accepted across the research community. Utilising this well-documented model provides consistency and enhances the relevance of the results to the broader research community. To support reproducibility, NREL’s benchmark environmental conditions, aligned with the turbine’s original design basis, were used. These conditions, widely adopted in offshore wind research, provide a realistic and consistent reference point for the analyses in later chapters.

Although the time-domain simulations in this chapter were conducted using an in-house MBD model, this model has been validated against NREL’s OpenFAST wind turbine simulator. OpenFAST is an open-source, industry-standard tool that has been verified through international benchmarking efforts, ensuring the reproducibility and credibility of this analysis [76]. While identical simulations could be carried out using OpenFAST, the in-house model was chosen for its enhanced flexibility, enabling the integration of advanced methods explored in subsequent chapters.

While this chapter establishes a robust control case for the remainder of the thesis, it is important to acknowledge that the calculations presented here are based on idealised assumptions. Specifically, factors such as localised stress concentrations and the degradation of corrosion protection over time are not considered in this work. The following chapter builds upon this foundation by investigating the role of stress concentrations in offshore wind turbine structures. By incorporating detailed representations of geometric features, the analysis aims to quantify how localised stresses affect fatigue performance. This allows for a more accurate and realistic assessment of the structural integrity of the structure and serves as a crucial step toward developing advanced fatigue assessment methods.

Chapter 4

Composite Modelling Framework

4.1 Motivation

The MBD model discussed in Chapter 3 is capable of accurately capturing the wind turbine’s operational response to both aerodynamic and hydrodynamic loading in an efficient manner. However, this model considers the turbine’s structural components on a global scale based on the component stiffnesses and mass distributions, which do not take into consideration the influences of local geometry changes, flanges, welds, access doors, platforms, etc. Such geometric discontinuities can have a large influence on the stresses experienced in their vicinity, giving rise to stress concentrations. Stress concentrations stemming from geometric discontinuities such as flange connections have been identified as a contributing factor in the failure of numerous onshore wind turbines [114]. Therefore, these stress concentrations must be carefully considered and incorporated into the fatigue analysis of structural components to ensure accurate assessment and mitigation of potential failure risks.

To accurately detect stress concentrations which arise from such discontinuities, high-fidelity models such as FE models are required. To address this, a detailed FE model of the IEA 15-MW wind turbine is constructed, detailing accurate geometric and material properties of the wind turbine. This model is employed to investigate the fatigue damage stemming from stress concentrations which arise due to a proposed ring flange connection designed to connect the turbine tower to the transition piece.

To achieve this, a composite modelling approach is adopted, coupling results from the multi-body structural dynamics model with the FE model, exploiting the strengths of both modelling approaches. The composite model is created through one-way coupling, integrating a SCF derived by the FE model with stresses computed by the MBD model to produce enhanced local stresses at the connection between the tower and the transition piece. Once constructed, the composite model is implemented to assess the fatigue life of the wind turbine tower, explicitly accounting for the effects of the proposed connection flange.

The composite model framework, developed to comprehensively assess the fatigue life of the wind turbine tower, is constructed with two individually powerful modelling approaches. To provide an understanding of the distinct roles and capabilities of the modelling approaches employed within the composite model, the following section provides a detailed discussion of the framework, beginning with an overview of the MBD and FE modelling strategies and their respective contributions to the composite model.

4.2 Composite Model Framework

The implementation of MBDs and FE models in structural assessments varies significantly due to the inherent strengths and weaknesses of each modelling strategy. The MBD model is primarily focused on dynamic analysis, providing an efficient approach for simulating complex systems. One of its key strengths is its ability to perform system-level response analysis with relatively low computational cost. By enabling control over the DOFs used in the analysis, the MBD model supports reduced-order modelling, which further reduces computational expense. This efficiency makes it particularly suitable for analyses requiring high numbers of simulations, such as sensitivity analysis and parametric studies. The modular design of MBD models also facilitates seamless integration with software tools, allowing theories and mathematical formulations to be easily interchanged or updated. Additionally, the modular design makes the MBD model particularly effective for holistic analysis, as it can incorporate coupled loads from aero and hydrodynamics and allows for the implementation of controller algorithms, enabling accurate simulation of blade pitch actuation, providing a comprehensive operational model.

The MBD model's foundation in solving dynamic equations for the DOFs at

each time step allows large volumes of derived results to be stored efficiently, minimising storage requirements. However, this dependence on the DOFs can make the visualisation of results difficult, often requiring additional post-processing software tools to provide interoperability. Further limitations of the MBD model include its restriction to dynamic analysis and its intermediate level of detail, which restricts the range of result outputs. Additionally, the construction of such models can be complex, requiring numerous interconnected modules and frequent parameter updates. The MBD model also lacks versatility when modifications to component dimensions are required and cannot account for material nonlinearity, which may restrict its applicability in certain cases.

On the other hand, the FE model is highly versatile, offering superior flexibility for structural assessments due to its ability to perform a wide range of simulations, including modal, static, dynamic, and thermal analyses. One of its key strengths lies in its ability to deliver detailed analysis of components, enabling the calculation of local displacements, stresses, and strains. Using FE models, specific locations of interest can be investigated with greater accuracy through local seeding, enabling models to accurately capture stress concentrations, fracture mechanics, and contact simulations. FE modelling is also capable of analysing complex problems with irregular component shapes and material properties, such as the wind turbine blades. This empowers detailed analysis to be performed, including the ability to capture high-frequency modes of independent components. The versatility of FE models extends to the modelling of boundary conditions, constraints and load application types. Moreover, the FE model enables the components to be parametrically modelled, enabling alterations of the dimensions of the models' components, making it particularly useful for design optimisation.

The visualisation capabilities of FEM are highly sophisticated, providing intuitive representations of deformations, load paths, stress fields, thermal gradients, and other critical parameters of interest. However, these powerful visualisation features come at the expense of considerable storage requirements. In addition, the high-fidelity nature of FEM comes at a high computational cost, particularly for dynamic analysis, which can be prohibitively computationally intensive. Implementing coupled loading scenarios, such as aero and hydrodynamic loads, adds another layer of complexity, often requiring inputs from external software and relying on significant simplifications. Furthermore, the accuracy of FEM results is heavily dependent

on the quality of the model's construction, including the choice of elements, loading types, boundary conditions, and mesh densities.

The composite model developed in this work is designed to integrate the strengths of both the MBD and FE models, addressing the individual limitations of each while maximising their combined effectiveness. By combining the MBD model's efficient and holistic system-level dynamic response capabilities with the FE model's precision in simulating local stresses, the composite model aims to deliver accurate results for critical wind turbine components at a reduced computational cost.

To accomplish this, the composite model is developed (see Figure 4.1), coupling results from the multi-body structural dynamic model with a detailed FE model. This blended modelling approach enables the determination of local fatigue loads in specific components, such as a proposed ring flange connection in the wind turbine substructure. Within this framework, the FEM is utilised to calculate SCFs at geometric discontinuities in fatigue-prone regions of the wind turbine support structure. Simultaneously, the nonlinear aeroelastic MBD model captures the holistic response of the wind turbine, generating stress time histories for a range of environmental conditions based on the turbine's operational state. By post-processing and coupling these results, the composite model produces localised stress time histories in fatigue-critical areas, which are then used to estimate the fatigue life of the component.

To complete the development of the composite model, a proposed ring flange connection was designed for the IEA 15-MW wind turbine, and a detailed FE model of the wind turbine was constructed to incorporate this connection. The following sections provide an in-depth discussion of the ring flange design, the FE model's construction, the validation processes, and the fatigue analysis conducted on the butt weld connecting the flange to the tower base.

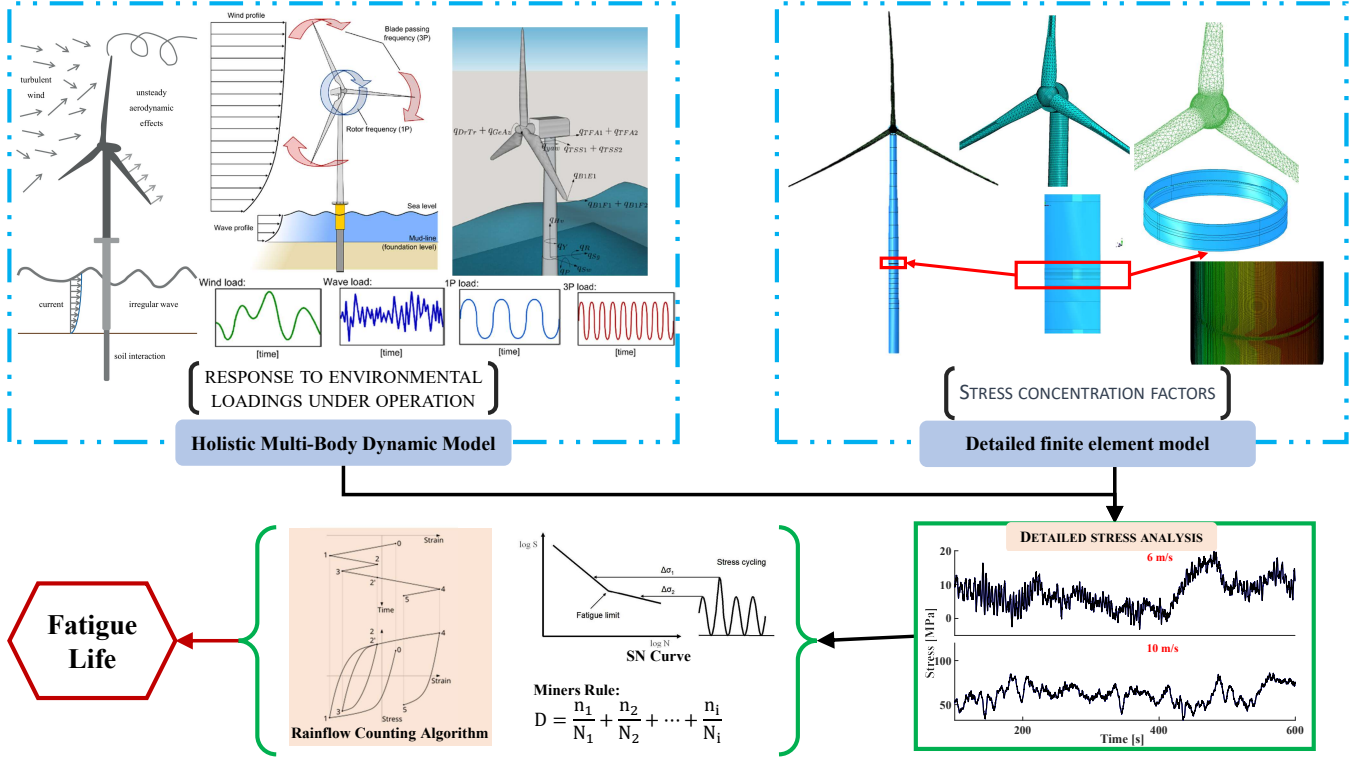


Figure 4.1: Composite modelling framework flowchart

4.3 Internally Bolted Ring Flange Design

The specifications of the 15-MW reference wind turbine are detailed in the IEA Wind TCP Task 37 technical report [195], with additional supporting data files available on GitHub. While these resources provide a comprehensive overview of the turbine's design, several simplifications are evident, particularly in the modelling of the tower, transition piece, and monopile. These components are designed as simple isotropic steel tubes with varying thicknesses. Notably, the reports lack specific details regarding connection types, locations and dimensions within the support structure's components. To address this gap, this work focuses on designing and testing a connection between the tower base and the transition piece to evaluate its influence on the fatigue damage experienced by the wind turbine tower. The connection is assumed to be a bolted ring flange connection, which is the most commonly employed design for joining wind turbine tower segments and substructures [113].

The ring flange connection is designed using elastic-plastic theory, based on the ultimate limit state of the connection, following the Petersen/Seidel approach as outlined in IEC 61400-6 [217]. The detailed calculations in this segment adhere to the procedures described in Seidel [218]. The Peterson/Seidel approach analyses the structural behaviour of the tower flange connection by considering a segment corresponding to a single bolt as depicted in Figure 4.2, adapted from Seidel and Schaumann [219]. The dimensions of such a segment used in these calculations are illustrated in Figure 4.3, which has been adopted from International Electrotechnical Commission [217].

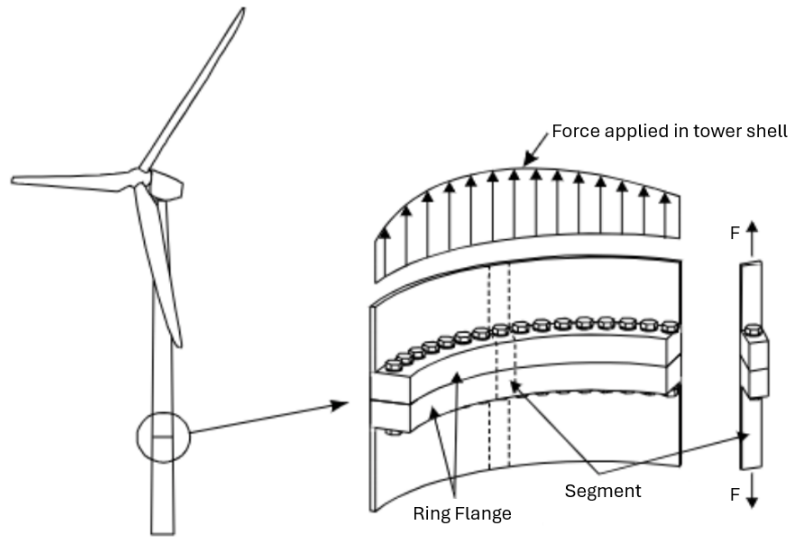


Figure 4.2: Segment-based structural analysis of tower flange connections, adapted from [219]

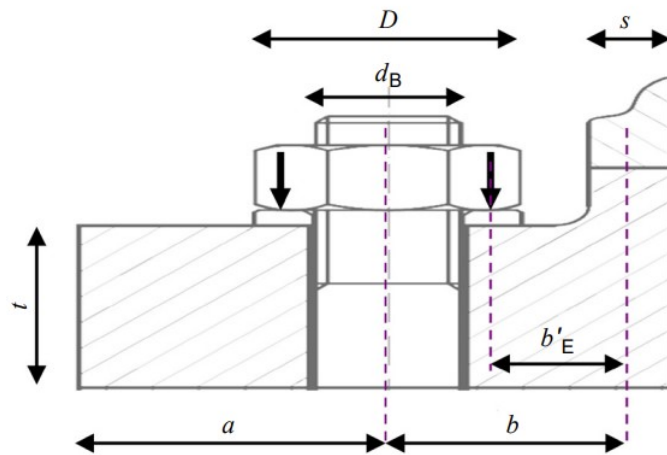


Figure 4.3: IEC 61400-6 flange dimensions [217]

The turbine tower wall and flange component are constructed using S355 steel, a high-strength low-alloy steel with a minimum yield strength of 355 MPa. This material is commonly used in wind turbine towers and is chosen for its strength and durability [70, 220, 221].

The initial step in the flange design process is the calculation of the b and c dimensions, where c is the width of a single bolt segment of the flange. Minimum values for these parameters can be obtained from Table 2 in Chapter 1 of Seidel [218]. These values are dependent on the wall thickness of the tower and the diameter of the bolts selected for use in the flange connection. In this analysis, the wall thickness at the top of the monopile transition piece is specified as 0.0411m, while the wall thickness at the base of the tower is 0.0395m, as provided by the IEA data files. Two different design configurations were tested, each considering variations in the wall thickness of both the tower and the transition piece. Among these designs, the configuration with the larger flange dimensions, corresponding to a wall thickness of 0.0411 meters, was selected. Due to the significant loads experienced by wind turbine structures, flanges designed for these applications typically require large bolt diameters, such as M72 bolts [46, 222]. Consequently, M72 bolts were selected for use in these calculations. However, Table 2 in Chapter 1 of Seidel [218] does not provide a “ $b - s/2$ ” value for bolts larger than M48. To address this, the table was extrapolated to obtain a suitable value for M72 bolts, resulting in a value of 0.1074m. To account for the uncertainty in this extrapolation, additional verification steps were implemented. Specifically, Formula 1 from Section 1.2 of Seidel [218] was applied: $b_{max} + s/2 \leq 2d$. Using this formula, the maximum allowable value for b_{max} was calculated as: $b_{max} \leq 2(0.072) - 0.0411/2 \leq 0.1235m$. The extrapolated value of 0.1074m was further validated against the Eurocode 3-1-8 [223] recommendation that the distance between bolts and an edge should exceed $1.2d = 1.2x(0.072) = 0.0864m$. Thus, 0.108m was selected as a suitable value for b .

In a similar manner, a value for c_{min} was extrapolated from Table 2, Chapter 1 in Seidel [218], yielding $c_{min} = 0.1604m$. The maximum width of the flange segment c_{max} , can be calculated using equation 199 in Chapter 11 of Seidel [218] as:

$$c_{max} = \frac{F_{t,R}}{1.8 \cdot s \cdot \sigma_x} \quad (4.1)$$

Here, $\sigma_x = 239.5$ MPa represents the critical buckling stress for the steel used in the tower wall, a value adopted from Seidel [218] which was determined in accordance

with DIN 18800 Part 4. σ_{max} is found by applying the material and load safety factors of 1.35 and 1.1 respectively:

$$\sigma_{max} = \frac{1}{\gamma_m} \cdot \frac{1}{\gamma_f} \cdot \sigma_x = \frac{1}{1.1} \times \frac{1}{1.35} \times 239.5 \text{ MPa} = 161.28 \text{ MPa} \quad (4.2)$$

The term $F_{t,R}$ denotes the maximum tensile resistance of a bolt, calculated as:

$$F_{t,R} = \frac{0.9 \cdot A_s \cdot f_{ub}}{1.25} = \frac{0.9 \times 0.0035 \times 1000 \times 10^6}{1.25} = 2520 \text{ kN} \quad (4.3)$$

In this equation, $f_{ub} = 1000 \text{ MPa}$ is the ultimate tensile strength of 10.9-grade high-strength steel bolts, as specified in Eurocode 3-1-8 [223]. A_s is the tensile stress area of the bolt, sourced from Engineers Edge [224] as 0.0035 m^2 . Stemming from these calculations the c_{max} is calculated as:

$$c_{max} = \frac{F_{t,R}}{1.8 \cdot s \cdot \sigma_x} = \frac{2,520,000}{1.8 \times 0.041058 \times 161.28 \times 10^6} = 0.2089 \text{ m} \quad (4.4)$$

Combining these results, the allowable range for the flange segment width c is $0.1604 \text{ m} \leq c \leq 0.2089 \text{ m}$. With b determined, the Pitch Circle Diameter (PCD) of the bolt arrangement can be calculated as:

$$PCD = D_{outer} - 2 \cdot \left(\frac{s}{2} + b \right) = 10 - 2 \times \left(\frac{0.041058}{2} + 0.108 \right) = 9.7429 \text{ m} \quad (4.5)$$

Knowing the PCD and an appropriate value for c , the number of bolts in the flange connection can be determined using Equation 4.6, from which the exact c value can be determined in Equation 4.7.

$$Bolts = \frac{\pi \cdot PCD}{c} = \frac{\pi \times 9.7429}{0.1604} = 190.803 \approx 190 \quad (4.6)$$

$$c = \frac{\pi \cdot PCD}{NumBolts} = \frac{\pi \times 9.7429}{190} = 0.1611 \text{ m} \quad (4.7)$$

To determine the required dimension a , which spans from the flange inner edge to the centreline of the bolt, the maximum of two values is considered:

$$a = \max(a_{klaff}, a_{GZT}) \quad (4.8)$$

Here, a_{Klaff} represents the dimension necessary to prevent the separation of the flanges beyond the bolt axis when subjected to a tensile force Z . This tensile force

acting on the flange segment is calculated as:

$$Z = \sigma_{\max} \cdot s \cdot c = 161,280,000 \times 0.0411 \times 0.1611 = 1067.9kN \quad (4.9)$$

Using this tensile force Z , a_{Klaff} can be calculated with the following formula:

$$a_{klaff} = 2.5 \cdot \left(\frac{120 \cdot Z \cdot b \cdot c + 4 \cdot \sqrt{900 \cdot Z^2 \cdot b^2 \cdot c^2 + 1410 \cdot Z^2 \cdot A \cdot b \cdot c - 1500 \cdot Z \cdot A \cdot b \cdot F_v \cdot c}}{400 \cdot F_v \cdot c - 376 \cdot Z \cdot c} \right) \quad (4.10)$$

Where A is the cross-sectional area of the bolt:

$$A = \pi \cdot \left(\frac{D}{2} \right)^2 = \pi \times \left(\frac{0.072}{2} \right)^2 = 0.0041m^2 \quad (4.11)$$

The bolt preload F_v can be calculated using Equation 4.12 as defined in Section 3.6 of Eurocode 3-1-8 [223]. In this formula, γ_{M7} is taken as 1.1 for high-strength bolts.

$$F_v = 0.7 \cdot A_s \cdot \frac{f_{ub}}{\gamma_{M7}} = 0.7 \times 0.0041 \times \frac{1000 \times 10^6}{1.1} = 2227.3kN \quad (4.12)$$

Substituting these values into the equation for a_{Klaff} yields:

$$a_{Klaff} = 0.1233m. \quad (4.13)$$

Next a_{GZT} is calculated as:

$$a_{GZT} = 1.25 \cdot b = 1.25 \times 0.108 = 0.135m \quad (4.14)$$

Using these two values, an appropriate value for a is determined:

$$a = \max(a_{klaff}, a_{GZT}) = \max(0.1233, 0.135) = 0.135m \quad (4.15)$$

Once values for a , b and s are determined, the thickness of the flange t can be determined based on the maximum of three design conditions:

$$t_{\text{ser}} \geq 1.1 \cdot a_{klaff} \quad (4.16)$$

$$t_{\text{ser}} \geq 1.1 \times 0.1233 \geq 0.135m \quad (4.17)$$

$$t_b = \sqrt{\frac{\left(\frac{F_{t,R} \cdot a_{GZT}}{a_{GZT} + b} \right) \cdot 8b - F_{t,R} \cdot (D + d_B)}{2 \cdot c' \cdot f_{y,k}}} = 0.1179m \quad (4.18)$$

Where D is the diameter of the washer used (0.125m), d_B is the borehole diameter

(0.075m) and $c' = c - d_B = 0.1611 - 0.075 = 0.0861m$. The thickness of the design flange is then found by:

$$t = \max \left(\begin{array}{c} t_{\text{ser}} \\ \frac{a_{\text{Klafl}} - b}{a - b} \cdot 3b, \text{ if } a_{\text{Klafl}} > b \\ t_b \end{array} \right) = \max \left(\begin{array}{c} 0.1356 \\ 0.1833 \\ 0.1179 \end{array} \right) = 0.1833 \approx 0.19m \quad (4.19)$$

Once the dimensions of the flange are determined, the design must be evaluated under the maximum loads expected to act on the flange. The ultimate bending moments and axial forces experienced by the flange were obtained through simulations conducted using the MBD model. The simulations incorporated loads from the IEC 61400-1 [49] Damage Load Cases (DLC) 1.3, 1.4, 1.5, 2.3, 3.2, 3.3, 4.1, 4.2, 6.2, 6.3 and 7.1. From these simulations, the maximum combination of axial force and bending moment was identified and converted to an equivalent tensile force acting on a single bolt segment using Equation 4.20 obtained from Tran, Kang, and Lee [44]:

$$N = \frac{1}{n_b} \left(\frac{8 \cdot M_{ed} \cdot D_o}{D_o^2 + D_i^2} - N_{ed} \right) = \frac{1}{190} \left(\frac{8 \times 4.4901 \times 10^8 \times 10}{10^2 + 9.9178^2} - 1.816 \times 10^7 \right) = 1157.6kN \quad (4.20)$$

Where N is the equivalent tensile force, M_{ed} is the maximum bending moment, N_{ed} is the corresponding axial force, D_o and D_i are the outside and inside diameters of the tower, respectively, and n_b is the number of bolts.

Within the Peterson/Seidel design approach, four failure criteria are defined based on potential failure of the bolts, tower wall, and flange, all of which must be satisfied. Each criterion is evaluated to ensure that the corresponding resistance force exceeds the applied tensile force.

$$Z \leq F_u = \max(F_{U,A}, F_{U,B}, F_{U,C}, F_{U,D}) \quad (4.21)$$

The resistance forces are calculated using the forces and moments derived in Equations 4.22- 4.27. The yield strengths of the S355 steel in the tower wall and the flange used in these equations were obtained from Table 6 of BS EN 10025-2:2019

[225], based on their respective thicknesses. For the tower wall, the yield strength is $f_{y,d,BI} = 345$ MPa, while for the flange, it is $f_{y,d,Fl} = 285$ MPa.

$$M'_{pl,2} = \frac{c' \cdot t^2}{4} \cdot \frac{f_{y,d,Fl}}{\gamma_M} = \frac{0.0891 \times 0.19^2}{4} \times \frac{285 \times 10^6}{1.1} = 208.3 kNm \quad (4.22)$$

$$M_{pl,Fl} = \frac{c \cdot t^2}{4} \cdot \frac{f_{y,b,Fl}}{\gamma_M} = \frac{0.1611 \times 0.19^2}{4} \times \frac{345 \times 10^6}{1.1} = 376.7 kNm \quad (4.23)$$

$$M_{pl,BI} = \frac{c \cdot s^2}{4} \cdot \frac{f_{y,d,BI}}{\gamma_M} = \frac{0.1611 \times 0.0411^2}{4} \times \frac{285 \times 10^6}{1.1} = 20.7 kNm \quad (4.24)$$

$$\Delta M_{pl,2} = \frac{F_{t,Rd}}{2} \cdot \frac{D + d_B}{4} = \frac{2,520,000}{2} \times \frac{0.125 + 0.75}{4} = 63 kNm \quad (4.25)$$

$$N_{pl,3} = c \cdot s \cdot \frac{f_{y,d,BI}}{\gamma_M} = 0.1611 \times 0.04058 \times \frac{345 \times 10^6}{1.1} = 2,016.4 kN \quad (4.26)$$

$$V_{pl,3} = c \cdot \frac{t}{\sqrt{3}} \cdot \frac{f_{y,d,Fl}}{\gamma_M} = \frac{0.1611 \times 0.19}{\sqrt{3}} \times \frac{285 \times 10^6}{1.1} = 4,578.69 kN \quad (4.27)$$

$$b'_E = b - \frac{\left(\frac{D}{2} + \frac{d_B}{2}\right)}{2} = 0.1074 - \frac{\left(\frac{0.125}{2} + \frac{0.75}{2}\right)}{2} = 0.0574 m \quad (4.28)$$

The first of the four failure criteria assesses the resistance of the bolts and is calculated as:

$$F_{U,A} = F_{t,Rd} \quad (4.29)$$

To calculate the remaining criteria, it is first necessary to determine $M_{Pl,3}$. This value is defined as the smaller of two moments: $M_{Pl,V,Fl}$ and $M_{Pl,N,BI}$ which are both dependent on F_u , the minimum resistance of criteria A, B, C and D. Consequently, the calculation requires an iterative approach. To begin the iterative process, an initial assumption of $F_u = F_{t,Rd}$ is made. The calculations are performed until the change in the calculated minimum resistance falls within a pre-defined tolerance, ensuring convergence. The procedure is carried out by calculating the following:

$$M_{Pl,N,BI} = \left(1 - \left(\frac{F_u}{N_{pl,BI}}\right)^2\right) \cdot M_{pl,BI} \quad (4.30)$$

$$M_{Pl,V,Fl} = \left(\sqrt{1 - \left(\frac{F_u}{V_{pl,Fl}}\right)^2}\right) \cdot M_{pl,Fl} \quad (4.31)$$

$$M_{Pl,3} = \min(M_{Pl,N,BI}, M_{Pl,V,Fl}) \quad (4.32)$$

$$F_{u,B} = \frac{F_{t,R} \cdot a + M_{Pl,3}}{a + b} \quad (4.33)$$

$$F_{u,D} = \frac{M'_{pl,2} + \Delta M_{pl,2} + M_{Pl,3}}{b} \quad (4.34)$$

$$F_{u,E} = \frac{M_{pl,2} + M_{Pl,3}}{b'_E} \quad (4.35)$$

$$F_u = \max(F_{U,A}, F_{U,B}, F_{U,C}, F_{U,D}) \quad (4.36)$$

Performing the iterative procedure confirms that the flange is capable of withstanding the maximum tensile force of 1157.6 kN. Therefore, the flange is considered adequate with the specified dimensions as presented in Figure 4.4:

190 M72 bolts

$t = 0.190\text{m}$

$a = 0.135\text{m}$

$b = 0.108\text{m}$

$s = 0.041058\text{m}$

A fillet radius of 10 mm is applied between the flange and the weld neck to prevent edge distortion in accordance with IEC [217].

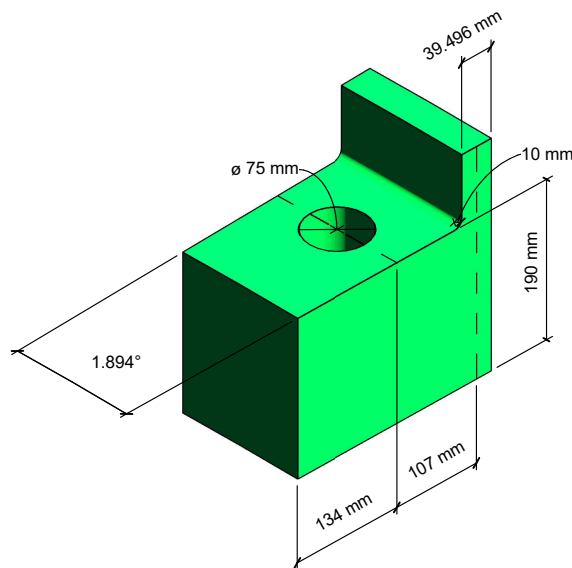


Figure 4.4: Designed tower ring flange connection dimensions

4.4 Internally Bolted Ring Flange Stiffening Effect

The presence of the flanges within the turbine tower and transition piece significantly influences the local stress distributions in their vicinity. Due to their substantial thickness and width relative to the tower wall, the flanges act as a stiffening element, altering the deformation behaviour of the tower.

When a compressive load is applied along the longitudinal axis of the tower, the Poisson effect leads to a radial expansion of the tower wall. This expansion, which would typically result in an outward bulging of the structure, is constrained by the flange. As the flange resists this radial deformation, the restriction induces additional localised bending stresses. These bending stresses increase to a peak at a distance close to the flange before gradually diminishing with further distance from the flange [46]. A similar phenomenon occurs under tensile loading, where the Poisson effect induces radial contraction. The flange again restrains this deformation, generating localised bending stresses. This stiffening effect can be particularly pronounced in the vicinity of the flange, introducing stress concentrations that must be considered in structural design and assessments. To capture these stress concentrations, a finite element model of the wind turbine is developed.

4.5 IEA 15-MW Offshore Wind Turbine Finite Element Model

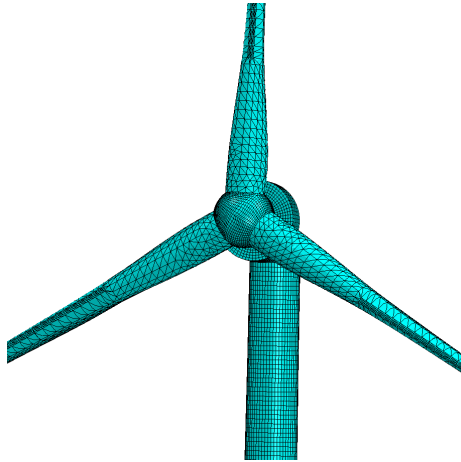
An accurate three-dimensional FE model of the IEA 15-MW monopile-based offshore wind turbine was developed using Abaqus/CAE software [226] to investigate the structural behaviour of the turbine, with a particular focus on accurately calculating the SCFs that arise in the tower wall above the proposed ring flange connection. The FE model was constructed using a parametric modelling approach, which was implemented through a custom Abaqus-Matlab interaction code. This approach allowed for the automation of model generation and facilitated the efficient adjustment of key geometric and material properties. The parametric components included in the model are the ring flange connection, the tower, the transition piece and the monopile. As discussed in Section 4.2, the accuracy of FE models is heavily dependent on the manner of the model's construction. In the following sections, the

modelling approach is described, detailing the elements employed, meshing strategies, and the validation process used to ensure the model's reliability.

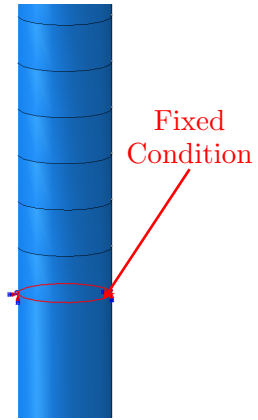
4.5.1 Structural Properties and Constraints

The detailed FE model incorporates geometric components supplied by the IEA, which have been adapted for use in Abaqus, along with additional parts derived from IEA data files [195]. The high-fidelity model includes intricate components such as the wind turbine blades, yaw mechanism, bearings and flanges, a number of which can be seen in Figure 4.5. The geometric and material properties as defined in Gaertner et al. [227] were utilised in creating the FE model, providing a structurally precise depiction of the turbine. The monopile, transition piece, turbine tower, yaw mechanism, flange and bedplate were constructed using steel with a Young's Modulus of 200 GPa, while a slightly stiffer steel (205 GPa) was assigned to the drive train components including the low-speed shaft, bearings and rotor. Additionally, cast iron was used for the hub.

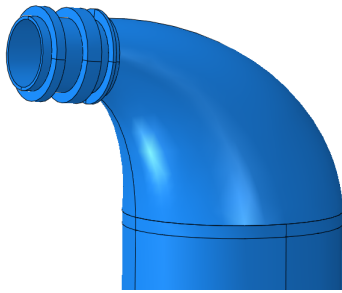
Given the inherent complexity of wind turbine blades, which are composed of advanced composite materials with intricate configurations, high-fidelity structural analysis models for the IEA 15-MW Reference Wind Turbine (RWT) blades are not currently available. Furthermore, the available data for the structural properties of the blades, including the positioning and quantities of key structural elements such as the spar, is limited. To address this, a simplified yet representative modelling approach was adopted. A geometric profile of the turbine blade, consisting of the airfoil shapes along its length, was created in Abaqus based on the IEA data files, which included a detailed shape file in SolidWorks. This profile was modelled as a solid component, which was divided into 50 sections along its length. For each section, the second moment of area about the flapwise axis was calculated based on the geometric profile of the corresponding airfoil segment provided by the IEA (see Figure 4.6). The second moment of areas of each of the segments were then implemented with the stiffness properties, provided in the ElastoDyn files available on the NREL GitHub repository, to provide an equivalent cumulative Young's modulus for each section. By applying this derived property in addition to the mass properties provided by the IEA, the representative model produced a wind turbine blade with an identical centre of mass and a natural frequency aligned with the first flapwise natural frequency indicated by the IEA.



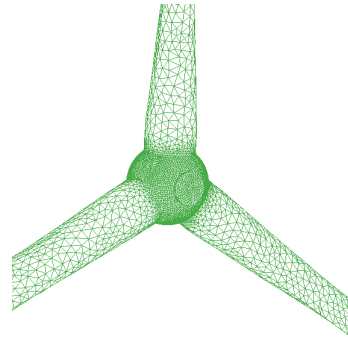
(a) Detailed mesh of hub blades and rotor



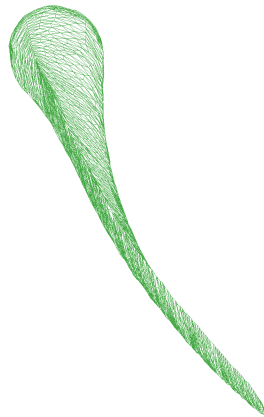
(b) Fixed condition at mud-line



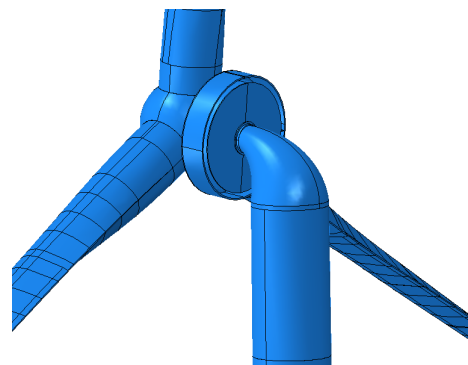
(c) Detailed bearings



(d) Hub and blades meshed



(e) Blade meshed



(f) Detailed bedplate rotor and stator

Figure 4.5: IEA 15-MW finite element model details

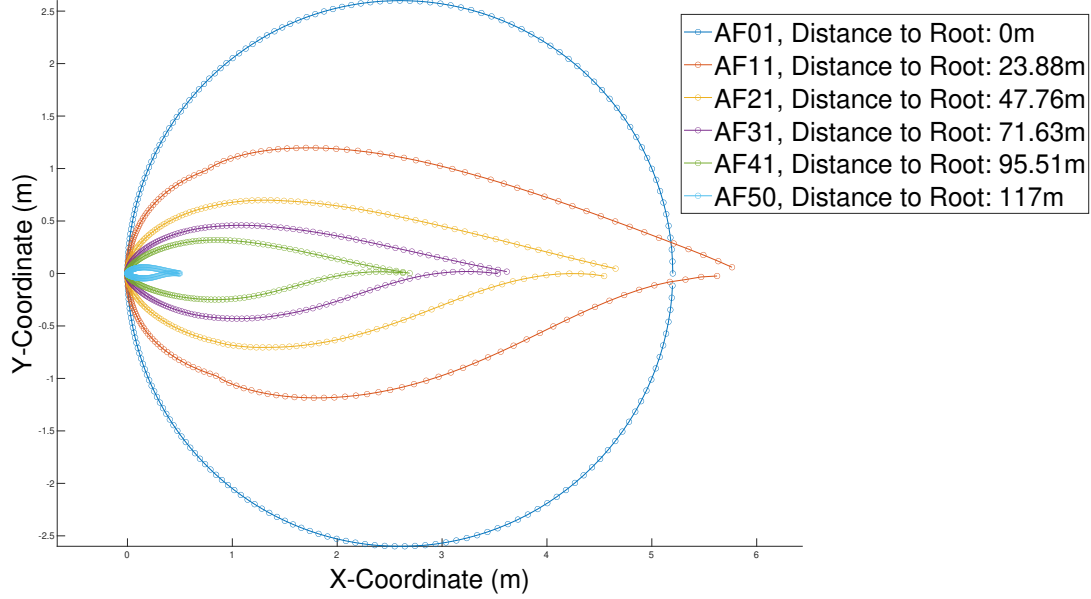


Figure 4.6: Airfoil evolution along the wind turbine blade span

For the support structure, the transition piece and monopile components are represented as a single component in the FE model, following a similar approach to the specified IEA geometry and data files. At the top of the single component, which represents the transition piece, the internally bolted ring flange designed in Section 4.3 is used to connect this component to the base of the turbine tower. The 190mm thick flange connections of the tower and transition piece are secured using 190 M72 10.9 grade high-strength steel bolts. The modelling of the bolts is considered outside the scope of this work, and consequently, the two flanges are connected using a tie constraint, assuming full surface contact between the mating flanges. As the flanges are positioned at the top of the transition piece and at the base of the tower, as depicted in Figure 4.7, the respective components are shortened by the thickness of the flange, ensuring that a tower top height of 144.582 meters was maintained.

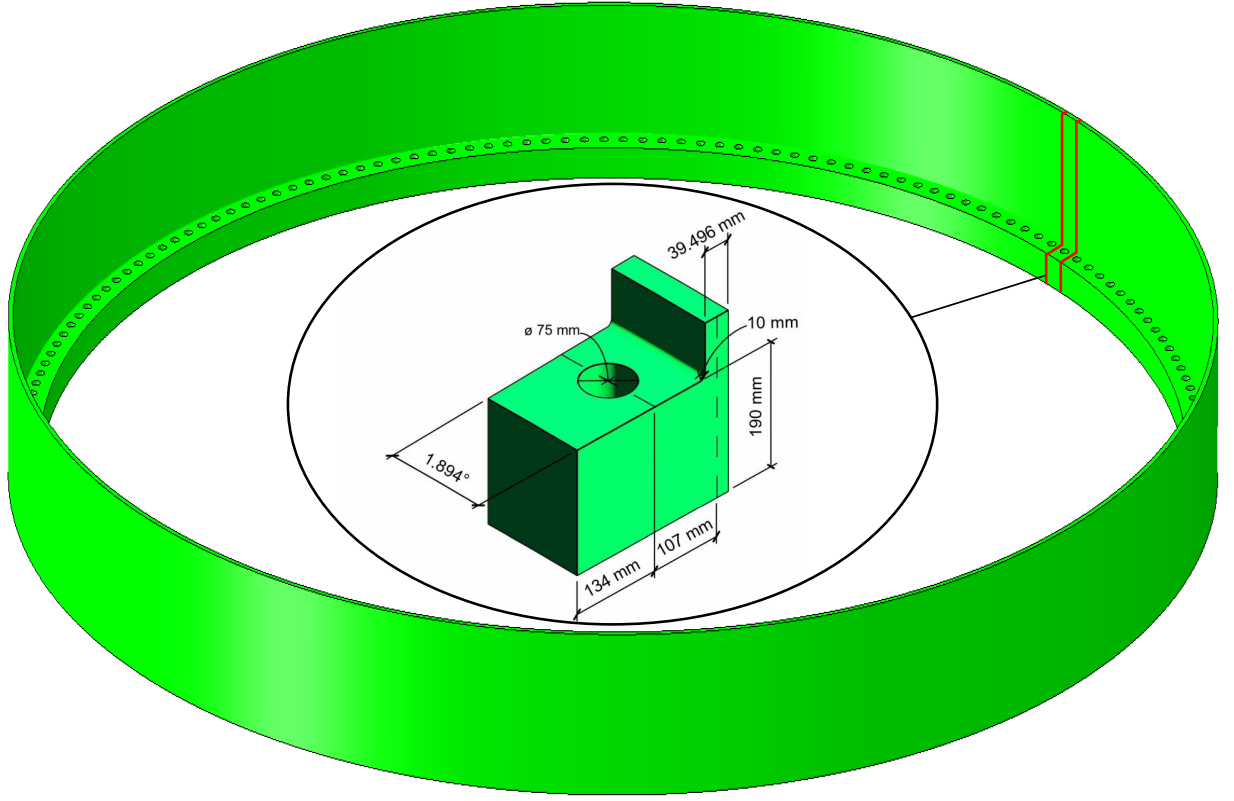


Figure 4.7: Internally bolted ring flange connection

Once the turbine assembly was completed, as depicted in Figure 4.5, a fixed boundary condition was applied at the mudline of the monopile, located 30 meters below Mean Sea Level (MSL), thus neglecting soil-structure interactions. The wind turbine was modelled in a parked configuration, with the yaw system aligned up-wind. The rotor was held stationary, and the blades were restricted from pitching, maintaining a feathered position throughout the simulations. With the geometric and material properties, as well as the associated constraints, now established, the following section delves into the detailed selection of component types used in the FE analysis.

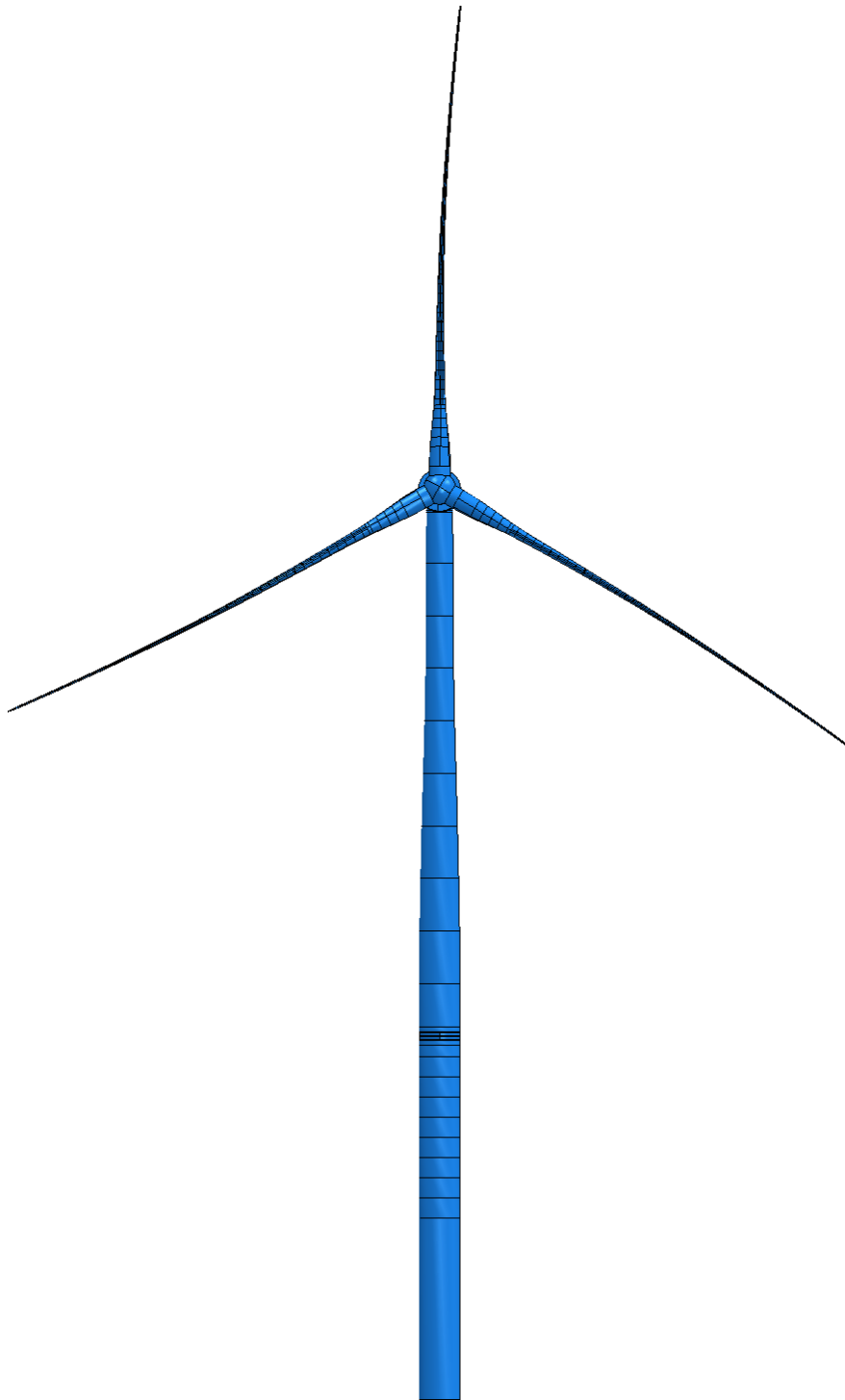


Figure 4.8: Finite element model of IEA 15-MW wind turbine

4.5.2 Component Selection

Two main component types can be employed when carrying out a FE analysis, namely Rigid Bodies and Finite Elements. The physical characterisation of Rigid and Flexible bodies described in Section 3.3.1 applies comparably to Rigid Bodies and Finite elements respectively. This correspondence allows for a consistent approach to be adopted between the MBD and FE modelling approaches.

Rigid Bodies: are non-deformable, meaning that the element shape does not alter and hence can be described in full by the motion of one node using just six DOFs and as a result are computationally efficient. A rigid body can be modelled in Abaqus by either creating a part as an analytical/discrete rigid body or by applying a rigid body constraint. Analytical rigid bodies are parts which can be characterised by simple geometric forms, these parts can be modelled mathematically as lines and arcs, which are not required to be meshed. In contrast, discrete rigid bodies are parts that are more complex in shape and can not be described easily mathematically, thus requiring meshing by the FE software. Alternatively, rigid body constraints can be applied to deformable components to reduce simulation costs when deformations and stress distributions of a component are negligible or not required for the analysis. Any standard element type can be used for rigid body constraints, enabling the component to act as a discrete rigid body.

Finite Elements: include all the remaining deformable element families, these elements require many DOFs to describe their positioning and properties such as temperature, etc. The Abaqus element library contains a wide scope of finite element types that include but are not limited to the following categories:

- Continuum Elements
- Structural Elements
- Inertial, Rigid, and Capacitance Elements
- Connector Elements
- Special-Purpose Elements

To minimise computational cost, the nacelle elements including the yaw mechanism, the bedplate, the nacelle flanges, the low-speed shaft, the low-speed

shaft bearings, the rotor, the stator and the hub were modelled as rigid components. The remaining components including the blades, the tower, the monopile/transition piece and the internally bolted ring flange connections were modelled as deformable finite elements. This modelling approach is consistent with the assumptions made in the MBD model with the exception of the low-speed shaft. The low-speed shaft is considered as a rigid element in the FE model, given that the turbine is modelled in a non-operational state, and its inclusion as a rigid element does not influence the desired results.

4.5.3 Element Selection

As previously mentioned, the Abaqus element library provides a wide range of element types, each suited to different modelling applications and analysis requirements. In this study, the selection of appropriate elements was guided by key factors related to the component under analysis, including its geometric characteristics, the required level of accuracy, and considerations of computational efficiency.

The rigid components housed within the nacelle in addition to the hub were meshed using hexahedral elements of type C3D8R (see Figure 4.9). This element is an eight-noded brick element with reduced integration, preventing the effects of shear locking. Although C3D8R elements can become overly stiff in bending and may produce inaccuracies in stress concentrations, they were selected for their low computational cost. Given that accurate stress and deformation analyses for these rigid components are not relevant in this study, the use of C3D8R elements is deemed appropriate.

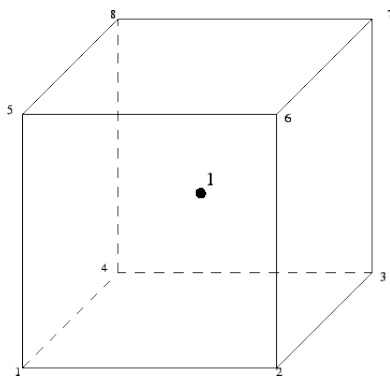


Figure 4.9: C3D8R element nodes and integration point [228]

The flexible blade components were meshed using C3D10 quadratic tetrahedron elements (see Figure 4.10). This 10-node element is a good general-purpose element which was selected for use due to its high geometric versatility, which was required to accurately capture the intricate geometry of the turbine blades.

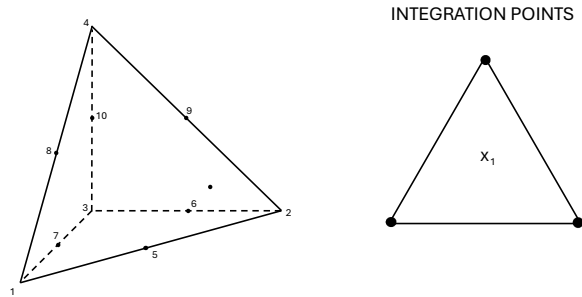


Figure 4.10: C3D10 element nodes and integration points

In the case of the turbine tower and monopile/transition piece, the thickness is several orders of magnitude smaller than both the longitudinal length and diameter. Consequently, these components were constructed using shell elements. The cylindrical nature of both the components further justified this choice of element type as shell elements are particularly effective at accurately modelling curved surfaces undergoing deformation. For these components, conventional shell elements were used based on a reference surface generated from IEA specifications with the thicknesses of the components specified using sectional properties. Simpson's integration rule was employed enabling the stress concentrations on the shell surfaces to be calculated accurately. In a similar manner to Hu, Baniotopoulos, and Yang [229], the shell element selected was the S4R element (see Figure 4.11), a 4-noded, quadrilateral, doubly-curved, conventional stress/displacement shell element which accounts for both membrane and bending effects. This element has reduced integration employing just one integration point, avoiding the effects of shear locking while hourglass controls are utilised to ensure accurate displacements of the elements. This element is a general-purpose element enabling Abaqus to apply thick shell theory or the discrete Kirchhoff thin shell approach based on the thickness of each section.

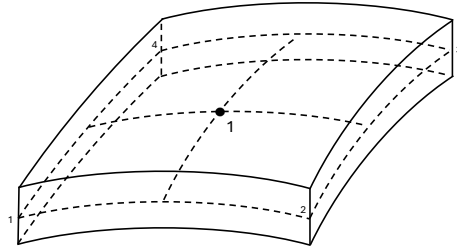


Figure 4.11: S4R shell element nodes and integration point

The internal ring flange components were meshed using C3D20R elements (see Figure 4.12). The C3D20R is a quadratic brick element with reduced integration, designed as a general-purpose nonlinear element. This element was selected as it behaves well in bending and does not typically suffer from the effects of hourglassing. When provided with a relatively fine mesh, this element is capable of modelling accurate stress concentrations as required in this work.

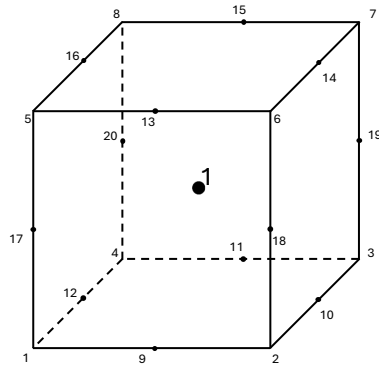


Figure 4.12: C3D20R element nodes and integration points

To validate the choice of element, a mesh convergence study is performed. Generally, if the selected element is appropriate, the results will gradually converge toward a consistent value as the seed size is reduced and the mesh is refined. Smooth and steady convergence typically indicates a reliable element, whereas erratic or non-converging results suggest issues with element selection or mesh quality.

4.5.4 Local Seeding

Local seeding was implemented in the FE model to minimise the computational cost of the FE model while maximising the accuracy of the results. This approach allowed for variations in mesh density across components, enhancing accuracy in areas of interest while reducing refinement in regions of lower concern. By applying local seeding within the model, the computational resources could be focused on specified areas, enabling efficient simulations without compromising the reliability of the results.

The local seeding strategy imposed divided the turbine into four main regions, as depicted in Figure 4.13.

- Region 1: This region consists of the flange connection and wall elements up to a height of 1 m above the flange top surface. This region includes the flange at both locations, at the base of the turbine tower and at the top of the monopile/transition piece. This particular region is selected as the key findings from the model are obtained from this section, and thus, the accuracy of stresses in this area are prioritised.
- Region 2: This region contains the top and bottom sections of the tower and transition piece, excluding Region 1. This second region acts as a transition zone in both the tower and monopile. A further subdivision within this region is made at a distance of 2 m from the flange, with 1 m belonging to Region 1. The remaining 1 m of this subdivision is meshed with the same seed length as the flange, while the rest transitions gradually to the coarser mesh of Region 3 using a single bias.
- Region 3: This region includes all of the remaining segments of the tower and monopile components.
- Region 4: This region encompasses all of the remaining components, including the rotor nacelle assembly and the wind turbine blades.

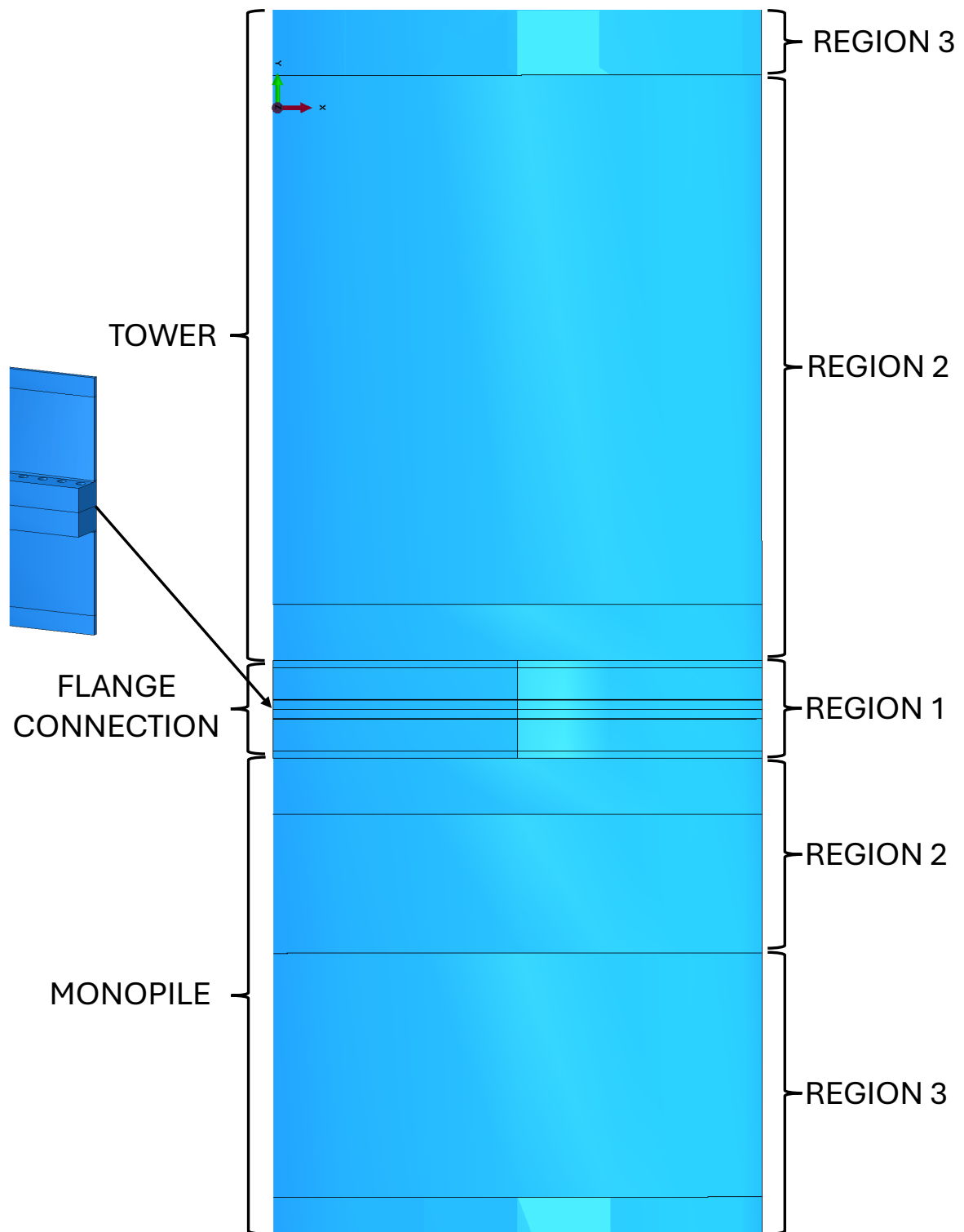


Figure 4.13: Finite element model mesh regions

4.5.5 Mesh Convergence Analysis

Based on the local seeding strategy above, a mesh convergence study was conducted to both validate the element selections and to determine an appropriate seed length for each region of the turbine components. The mesh convergence aimed to maximise the accuracy of the stresses induced in Region 1 while minimising the computational expense for the total model. The mesh convergence analysis was performed by implementing a static analysis with a bending moment applied at the top of the turbine tower while observing the maximum axial stress (S22) experienced in the turbine tower flange connection (Region 1).

Region 4, including the blade, hub, and nacelle components, being of secondary interest for this study, in which the stresses are not of great concern, are meshed with a maximum seed size of 1 m or with the maximum seed size at which the software allowed for each component.

To determine appropriate seed lengths for Regions 1, 2, and 3, a two-step mesh convergence study was conducted, as shown in Figure 4.14. In the first step, mesh convergence was performed for Region 1, while a fine mesh was applied in Region 3. An optimal seed length of 0.0197 meters was identified for Region 1, resulting in a mesh with over 1.4 million elements. This seed size corresponds to half the tower wall thickness, and thus complies with the DNV guidelines, which recommend a maximum seed length of 50% of the component thickness to ensure a minimum of two elements across the depth. By implementing such a resolution, stress gradients arising across the wall thickness can be captured effectively.

After selecting a suitable seed length for Region 1, a further mesh convergence was implemented on Region 3, with Region 2 being adjusted accordingly to comply with Region 1 and 3 seed lengths. As shown in Figure 4.14, the seed lengths in these regions had minimal influence on the resulting flange stress. Consequently, a seed length of 0.6 meters was selected for Region 3, minimising computational demands. By applying this local seeding strategy, the corresponding FE model was constructed, comprising of 1,440,800 elements, balancing computational cost and solution accuracy.

The resulting meshing strategy is illustrated in Figure 4.15, highlighting the variation in element seed lengths across Regions 1, 2, and 3. Region 1, where stress

accuracy is prioritised, is shown with a finer mesh, while Regions 2 and 3 exhibit a gradual transition to coarser mesh densities. Figure 4.16 presents a detailed view of the mesh applied to the flange segment, emphasising the refined discretisation used in this critical area. The transition between Regions 2 and 3 is depicted in Figure 4.17, where a single bias is used to achieve a smooth change in mesh density, ensuring both computational efficiency and stress continuity across regions. For this mesh configuration, a simulation computing both axial and bending SCFs required 9.84 hours of runtime on a Dell Precision 3240 Compact workstation equipped with an Intel Xeon W-1250 CPU and 32 GB of RAM.

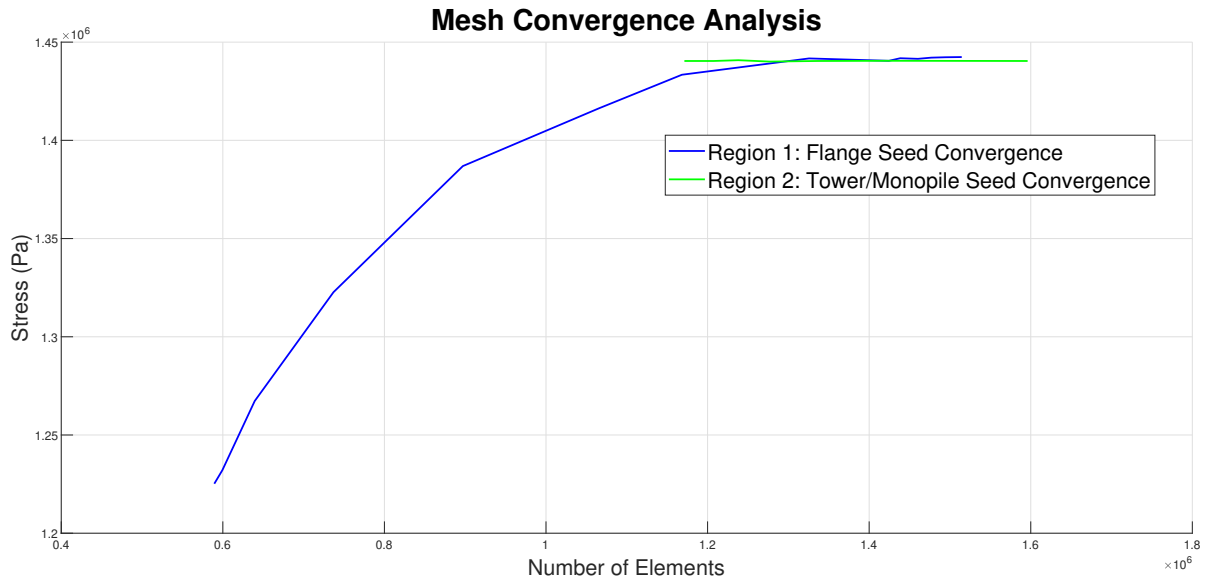


Figure 4.14: Mesh convergence analysis

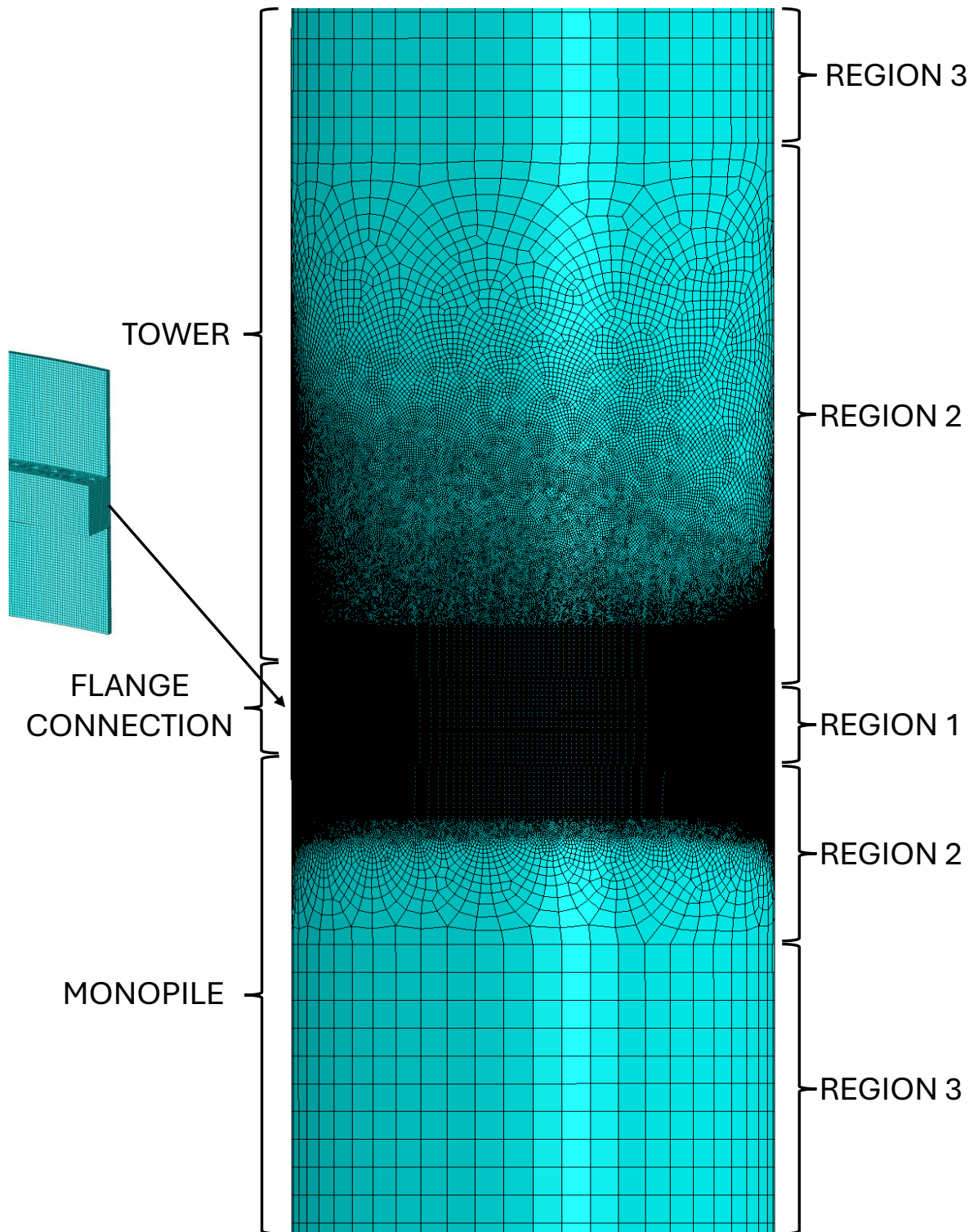


Figure 4.15: Finite element model mesh regions: meshed

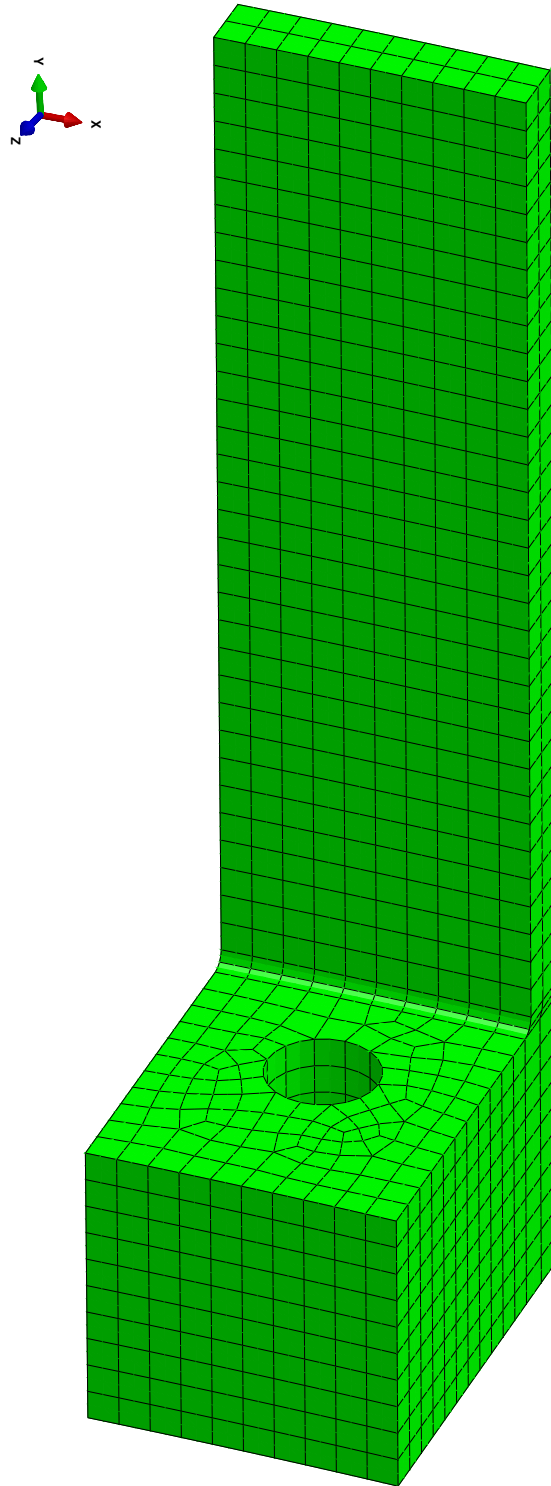


Figure 4.16: Segment of internally bolted ring flange connection meshed

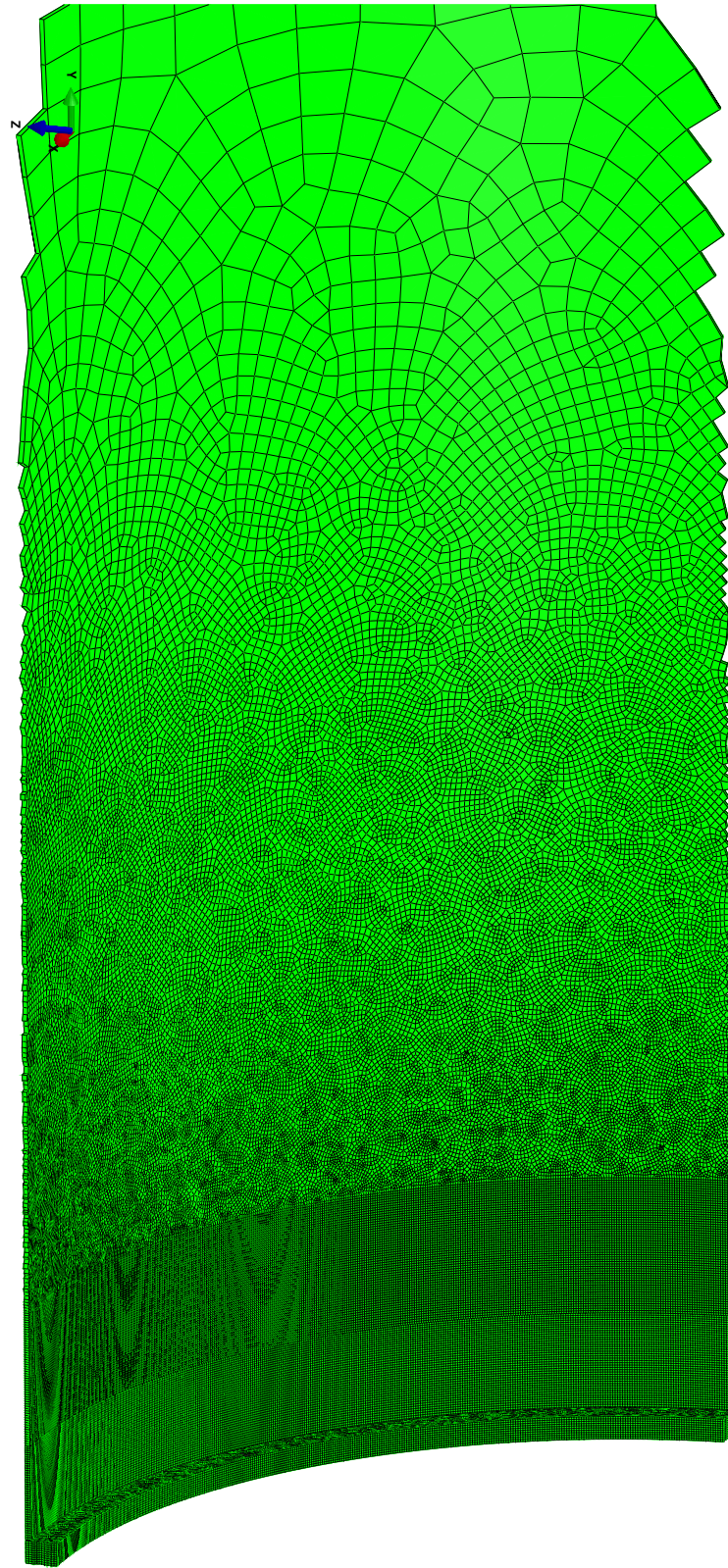


Figure 4.17: Transition mesh at tower base

4.5.6 Validation of Finite Element Model by Modal Analysis

To verify the accuracy and reliability of the FE model, a modal analysis is conducted to determine the natural frequencies and the corresponding mode shapes of both the turbine tower and blades. These results are then compared against the benchmark values provided by the IEA.

In the case of the turbine tower, the modal analysis is performed on two models of the wind turbine, one with the flange connection and the reference model without the connection. This analysis is performed considering the full structure with the monopile fixed at the mudline. For the reference model, the fundamental side-to-side natural frequency was identified as 0.17008 Hz, closely matching the IEA-specified value of 0.17 Hz. This represents a deviation of less than 0.05%, indicating excellent agreement. To examine the influence of the flange on the natural frequency, Table 4.1 presents the side-to-side natural frequency of the turbine tower for the FE model, both with and without the internally bolted ring flange. The table demonstrates that the inclusion of the ring flange results in a negligible frequency shift of 0.0588%. This minimal change indicates that the flange does not significantly contribute to the global stiffness of the tower structure. Consequently, the addition of the flange will not have a meaningful influence on the dynamic response of the turbine tower in the MBD model simulations.

Table 4.1: Effect of internally bolted ring flange on tower side-to-side natural frequency (%)

	No Flange	Flange
Natural Frequency (Hz)	0.17008	0.17018
Natural Frequency shift (%)	-	0.0588

Further confidence in the FE model is established by examining the mode shape corresponding to the first natural frequency in the side-to-side direction with the benchmark mode shape specified by the IEA in the OpenFAST software files. This specified benchmark mode shape was generated using BModes, a finite-element code developed by NREL to provide coupled mode shapes for OpenFAST [230]. The mode shape produced by the FE model aligns closely with the benchmark mode shape, as illustrated in Figure 4.18. This strong agreement is confirmed by a Modal Assurance Criterion (MAC) value of 0.9999, indicating excellent correlation between

the two mode shapes.

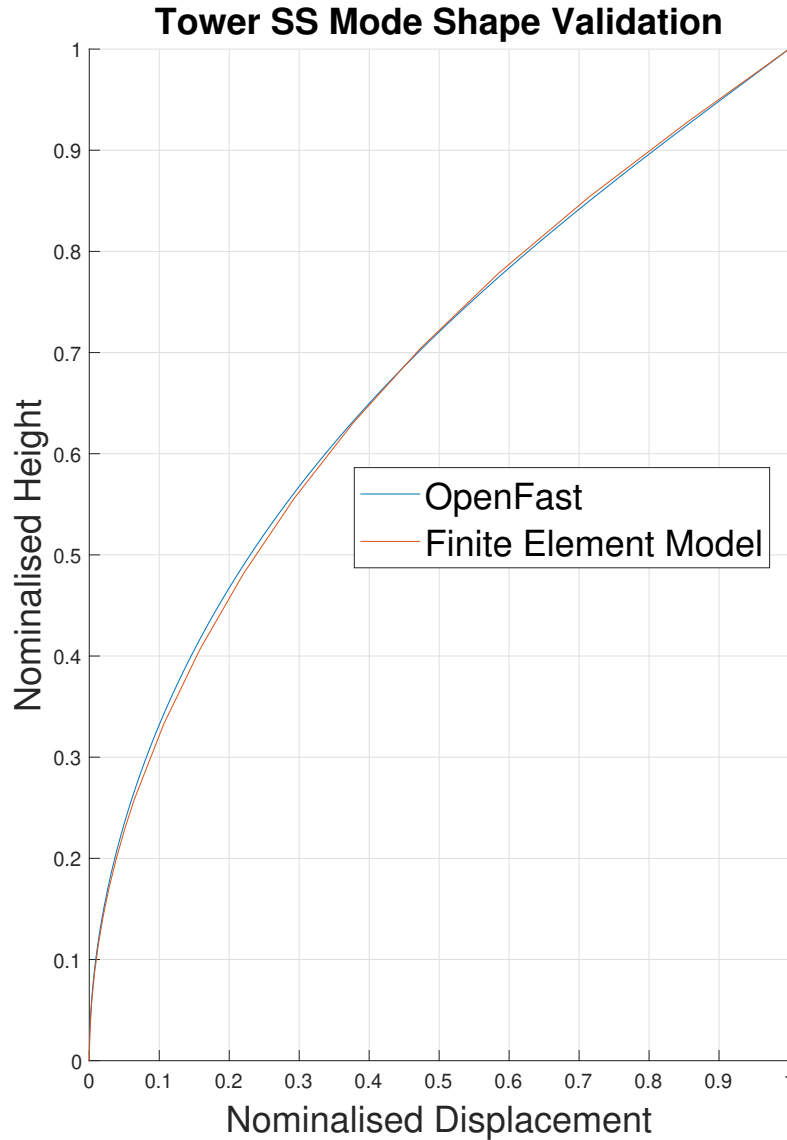


Figure 4.18: Tower fundamental mode shape (side-to-side)

Additional favourable results are obtained when comparing the natural frequency of the turbine blade in the flapwise direction. For this analysis, the blade was considered independently, with all other turbine components fixed in place. The fundamental natural frequency in the flapwise direction was calculated as 0.535 Hz, matching considerably well with the IEA-specified value of 0.555 Hz. This represents a deviation of approximately 3.6%, which is acceptable given the limited data

available for the blade component. Furthermore, the respective mode shape demonstrates a close correspondence between the FE model mode shape and IEA-specified mode shape as illustrated in Figure 4.19. This agreement was quantitatively confirmed by a MAC value of 0.9990, illustrating a strong correlation. The strong agreement between the FE model predictions and the IEA-specified values demonstrates the validity of the modelling approach employed in Section 4.5.1 for the turbine blades. The first three mode shapes of the wind turbine tower and blades obtained from the Abaqus simulation are presented in Figure 4.20. These illustrations and the strong alignment of quantitative results with the IEA benchmarks demonstrate that the model accurately captures the expected dynamic characteristics of the system, validating the representation of the structural stiffness and mass distributions and establishing confidence in its use for subsequent analyses.

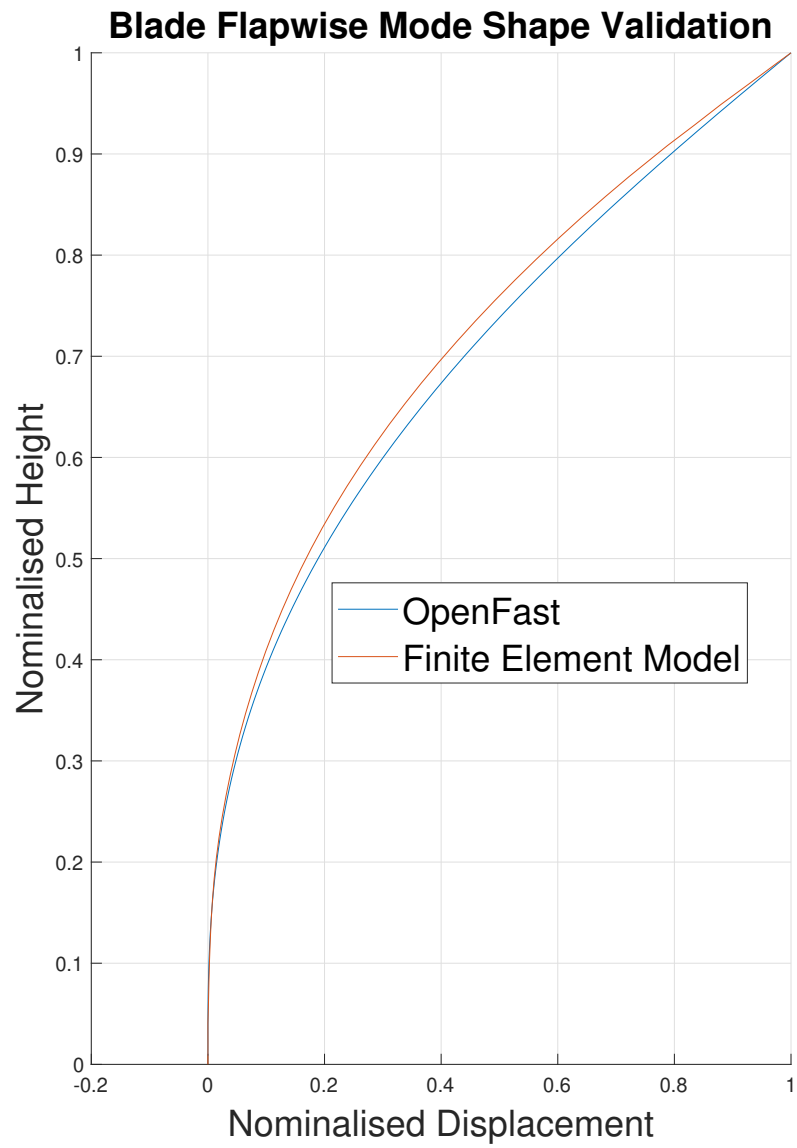


Figure 4.19: Blade fundamental mode shape (edgewise)

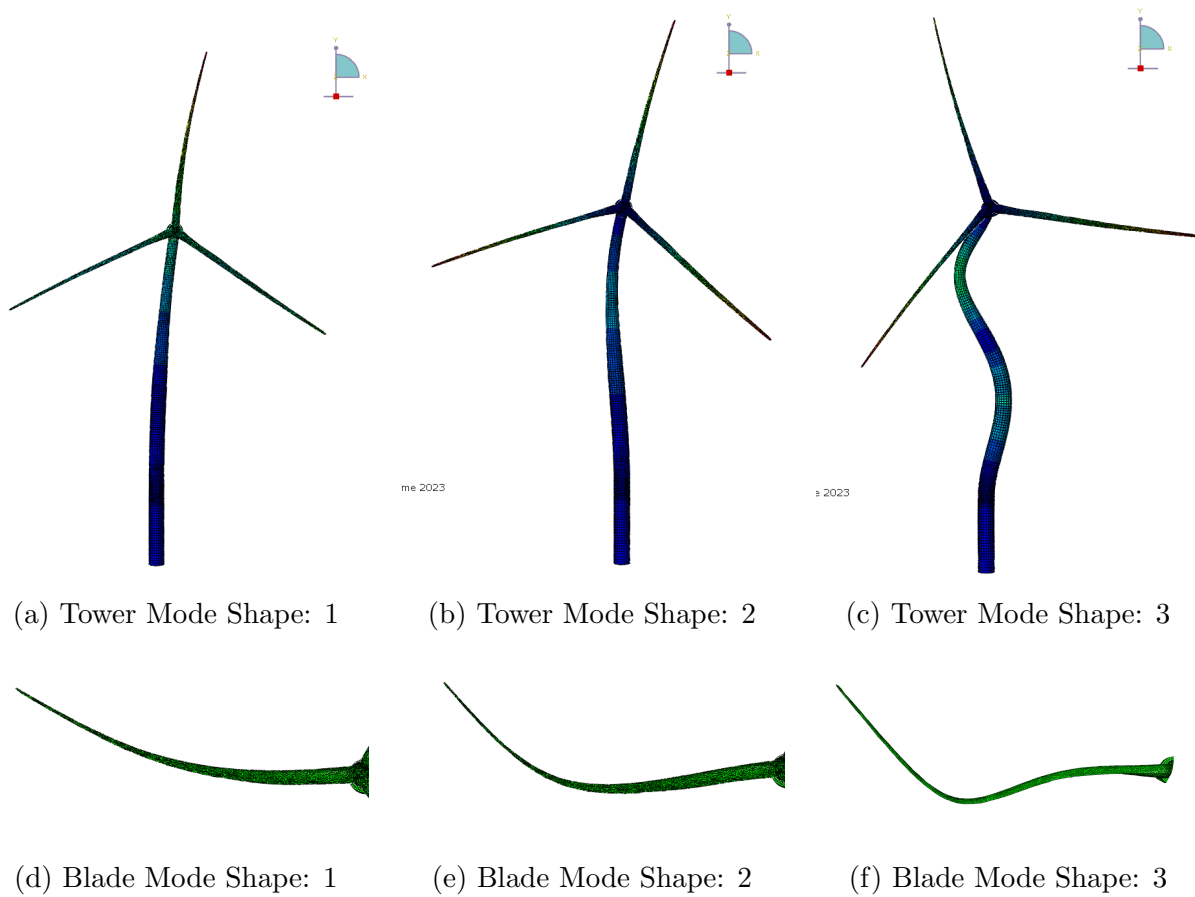


Figure 4.20: Mode shape illustrations

4.5.7 Stress Concentration Factor Calculation via Classical Shell Theory

To validate the SCFs calculated using the FE model, an analytical approach is also employed to estimate the stresses caused by the geometric discontinuity introduced by the flange. This analytical method follows the work of Seidel [46], in which the bending moments induced in the tower wall by a flange are determined using Shell Theory. In this approach, the flange connection is modelled as a ring beam under the assumption that flange rotation is constrained.

Shell theory is a specific branch of structural mechanics used to analyse the behaviour of curved, thin-walled components. A shell is defined as a structural element with one dimension (thickness) significantly smaller than its other two dimensions. In this work, shell theory is applicable as the turbine tower has a relatively small wall thickness compared to its diameter and length. Several key assumptions are made in classic shell theory: the material is assumed to be homogeneous, isotropic, and linearly elastic, undergoing small strains and small displacements. The shell's geometry is represented by its mid-surface and thickness, and transverse shear deformations are considered to be negligible. Normal stresses perpendicular to the shell's surface are negligible compared to in-plane stresses. Finally, the Kirchhoff-Love assumption assumes that straight lines normal to the mid-surface remain straight and normal after deformation.

Shell theory accounts for both membrane forces (acting in-plane) and bending effects (acting out-of-plane). Membrane forces dominate in regions where the shell's curvature is smooth, while bending effects become significant near discontinuities, boundaries, or localised loads. The bending effects in this approach are captured based on shell bending theory, which is capable of capturing the bending moments and transverse shear forces that arise due to the discontinuity in stiffness between the thin shell of the tower wall and the relatively thick flange.

The following equations outline the analytical formulation used to compute the bending stresses near the flange connection based on the flange design in Section 4.3. The parameters implemented in this calculation are:

- E - Young's modulus: 200 GPa

- a_1, a_2 - Inner flange radius to bolt centre line: 0.135 m
- b_1, b_2 - Bolt centre line to centre line of shell: 0.108 m, 0.1072 m
- t_1, t_2 - Flange thickness: 0.190 m
- s_1, s_2 - Thickness of the shells: 0.0394960 m, 0.041058 m
- r_1, r_2 - Radii at the mid thickness of the shells:
 $r_1 = D_{outer}/2 - s_1/2 = 10/2 - 0.0394960/2 = 4.980252$ m
 $r_2 = D_{outer}/2 - s_2/2 = 10/2 - 0.041058/2 = 4.979471$ m
- b_{RB} - Width of the ring beam cross-section: $a + b + s_1/2 = 0.135 + 0.108 + 0.0394960/2 = 0.262748$ m
- h_{RB} - Height of the ring beam cross-section: $= t_1 + t_2 = 0.19 + 0.19 = 0.38$ m
- r_{RB} - Radius of the ring beam

$$r_{RB} = r_1 - \frac{\left(\frac{s_1}{2} + a_1 + b_1\right)}{2} = 4.980 - \frac{\left(\frac{0.0395}{2} + 0.135 + 0.108\right)}{2} = 4.8686 \text{ m} \quad (4.37)$$

The bending moment at a distance x from the mating plane of the flanges can be calculated using the following formula.

$$m(x) = v \cdot \frac{1}{n} \cdot e^{-nx} \cdot \sin(nx) + m \cdot e^{-nx} \cdot (\cos(nx) + \sin(nx)) \quad (4.38)$$

where for a cylindrical shell

$$n_1 = \lambda_1 = \frac{\sqrt[4]{3 \cdot (1 - \mu^2)}}{\sqrt{r_1 \cdot s_1}} = \frac{\sqrt[4]{3 \times (1 - 0.3^2)}}{\sqrt{4.980252 \times 0.0394960}} = 2.8983 \text{ m}^{-2} \quad (4.39)$$

$$n_2 = \lambda_2 = \frac{\sqrt[4]{3 \cdot (1 - \mu^2)}}{\sqrt{r_2 \cdot s_2}} = \frac{\sqrt[4]{3 \times (1 - 0.3^2)}}{\sqrt{4.979471 \times 0.041058}} = 2.8414 \text{ m}^{-2} \quad (4.40)$$

The bending moment m and shear force v in the shell can be calculated by solving the general equation:

$$\delta \cdot X + \delta_0 = 0 \Rightarrow X = \delta^{-1} \cdot \delta_0 \quad (4.41)$$

Where X is the vector of the moment and shear forces:

$$X = \begin{bmatrix} \nu_1 \\ m_1 \\ \nu_2 \\ m_2 \end{bmatrix} \quad (4.42)$$

The vector of deformations is:

$$\delta_0 = \frac{1}{E} \begin{bmatrix} -\mu \cdot \frac{r_1}{s_1} \cdot p_1 \\ -\frac{12 \cdot M_0 \cdot r_{RB}^2}{b_{RB} \cdot h_{RB}^3} \\ -\mu \cdot \frac{r_2}{s_2} \cdot p_2 \\ \frac{12 \cdot M_0 \cdot r_{RB}^2}{b_{RB} \cdot h_{RB}^3} \end{bmatrix} \quad (4.43)$$

In which p_1 is a meridional force in the tower wall resulting from a nominal stress level of -8.09 MPa, which is selected to provide a comparison with FE results.

$$p_1 = \sigma \cdot t = -8.09 \times 10^6 \times 0.0394960 = -3195174.5 N \quad (4.44)$$

$$p_2 = p_1 \cdot \frac{r_1}{r_2} = -3195675.7 N \quad (4.45)$$

M_0 is the eccentricity moment stemming from the variance in wall thicknesses of the tower segments:

$$M_0 = p_1 \cdot e_1 \cdot \frac{r_1}{r_{RB}} - p_2 \cdot e_2 \cdot \frac{r_2}{r_{RB}} = 262.13 Nm \quad (4.46)$$

$$\delta_0 = \frac{1}{E} \begin{bmatrix} -\mu \cdot \frac{r_1}{s_1} \cdot p_1 \\ -\frac{12 \cdot M_0 \cdot r_{RB}^2}{b_{RB} \cdot h_{RB}^3} \\ -\mu \cdot \frac{r_2}{s_2} \cdot p_2 \\ \frac{12 \cdot M_0 \cdot r_{RB}^2}{b_{RB} \cdot h_{RB}^3} \end{bmatrix} = \begin{bmatrix} 0.00006043 \\ -0.00002586 \\ 0.00006043 \\ -0.00002586 \end{bmatrix} \quad (4.47)$$

The flexural rigidity of the shell is calculated as:

$$K_1 = \frac{E \cdot s_1^3}{12 \cdot (1 - \mu^2)} = \frac{200 \times 10^9 \times 0.0394960^3}{12 \times (1 - 0.3^2)} = 1128409.41 Nm \quad (4.48)$$

$$K_2 = \frac{E \cdot s_2^3}{12 \cdot (1 - \mu^2)} = \frac{200 \times 10^9 \times 0.041058^3}{12 \times (1 - 0.3^2)} = 1271548.18 Nm \quad (4.49)$$

The final component of Equation 4.41 is the matrix of flexibility coefficients δ which

is calculated as:

$$\delta = \frac{1}{E} \begin{bmatrix} \frac{E}{K_1} \cdot \frac{1}{2\lambda_1^3} + \frac{r_{RB} \cdot r_1}{b_{RB} \cdot h_{RB}} & \frac{E}{K_1} \cdot \frac{1}{2\lambda_1^2} & \frac{r_{RB} \cdot r_2}{b_{RB} \cdot h_{RB}} & 0 \\ \frac{E}{K_1} \cdot \frac{1}{2\lambda_1^2} & \frac{E}{K_1} \cdot \frac{1}{\lambda_1} + \frac{12 \cdot r_{RB} \cdot r_1}{b_{RB} \cdot h_{RB}^3} & 0 & -\frac{12 \cdot r_{RB} \cdot r_2}{b_{RB} \cdot h_{RB}^3} \\ \frac{r_{RB} \cdot r_1}{b_{RB} \cdot h_{RB}} & 0 & \frac{E}{K_2} \cdot \frac{1}{2\lambda_2^3} + \frac{r_{RB} \cdot r_2}{b_{RB} \cdot h_{RB}} & \frac{E}{K_2} \cdot \frac{1}{2\lambda_2^2} \\ 0 & -\frac{12 \cdot r_{RB} \cdot r_1}{b_{RB} \cdot h_{RB}^3} & \frac{E}{K_2} \cdot \frac{1}{2\lambda_2^2} & \frac{E}{K_2} \cdot \frac{1}{\lambda_2} + \frac{12 \cdot r_{RB} \cdot r_2}{b_{RB} \cdot h_{RB}^3} \end{bmatrix} \quad (4.50)$$

$$\delta = \frac{1}{2 \times 10^{11}} \begin{bmatrix} \left(\frac{2 \times 10^{11}}{1.13 \times 10^6} \times \frac{1}{2(2.90)^3} \right) + \left(\frac{4.87 \times 4.98}{0.2627 \times 0.38} \right) & \frac{2 \times 10^{11}}{1.13 \times 10^6} \times \frac{1}{2(2.90)^2} & \frac{4.87 \times 4.98}{0.2627 \times 0.38} & 0 \\ \frac{2 \times 10^{11}}{1.13 \times 10^6} \times \frac{1}{2(2.90)^2} & \left(\frac{2 \times 10^{11}}{1.13 \times 10^6} \times \frac{1}{2.90} \right) + \frac{12 \times 4.87 \times 4.98}{0.2627 \times 0.38^3} & 0 & -\frac{12 \times 4.87 \times 4.98}{0.2627 \times 0.38^3} \\ \frac{4.87 \times 4.98}{0.2627 \times 0.38} & 0 & \left(\frac{2 \times 10^{11}}{1.27 \times 10^6} \times \frac{1}{2(2.84)^3} \right) + \frac{4.87 \times 4.98}{0.2627 \times 0.38} & \frac{2 \times 10^{11}}{1.27 \times 10^6} \times \frac{1}{2(2.84)^2} \\ 0 & -\frac{12 \times 4.87 \times 4.98}{0.2627 \times 0.38^3} & \frac{2 \times 10^{11}}{1.27 \times 10^6} \times \frac{1}{2(2.84)^2} & \left(\frac{2 \times 10^{11}}{1.27 \times 10^6} \times \frac{1}{2.84} \right) + \frac{12 \times 4.87 \times 4.98}{0.2627 \times 0.38^3} \end{bmatrix} \quad (4.51)$$

$$X = \begin{bmatrix} \nu_1 \\ m_1 \\ \nu_2 \\ m_2 \end{bmatrix} = \begin{bmatrix} -5465.36 N \\ 981.92 Nm \\ -5042.28 N \\ 844.15 Nm \end{bmatrix} \quad (4.52)$$

By substituting these values into Equation 4.38, the bending moment along the inside tower wall can be calculated. The stresses arising from the bending moments can finally be calculated using:

$$M = \frac{Et^3}{12(1 - \nu^2)} \cdot (\chi_1 + \nu\chi_1), \sigma_b = \frac{E}{1 - \nu^2} \cdot \frac{t}{2} \cdot (\chi_1 + \nu\chi_1) \quad (4.53)$$

$$\sigma_b = \frac{6 \cdot M}{t^2} \quad (4.54)$$

Comparing the stresses induced by the axial load with the combined axial and bending stresses, a SCF distribution can be calculated along the length of the tower shell. Based on the dimensions of the designed flange, a maximum SCF of 1.089 is calculated at a distance of 0.556 meters from the mating plane of the flange connection. These values serve as benchmarks for validating the FE model, ensuring its accuracy in capturing the localised stress effects.

4.5.8 Stress Concentration Factor Calculation

Once the FE model is finalised, the SCF at the butt weld connection between the turbine tower and ring flange can be calculated. This SCF will be utilised with the nominal stresses produced using the MBD model to calculate local stresses surrounding the flange connection. There are two main components which contribute to the stress concentration at this location, the geometric effect of the ring flange and the notch effect stemming from the butt weld itself. The stress concentration arising from the notch effect is considered in the S-N curves provided by the DNV recommended practice guidelines [101], which are utilised in this fatigue analysis. As a result, the FE model will not consider any notch effects and will solely focus on the SCF arising from the geometric discontinuity of the ring flange.

A geometric SCF represents the localised high stresses arising due to a geometric discontinuity [103]. SCFs are a dimensionless metric that describes the ratio of hotspot stress to nominal stress, where the hotspot stress is defined as the geometric stress created by the considered detail. Geometric SCFs can be calculated using several approaches, in this work, an approach described in Peterson's book [103] is adopted. In this approach, the hotspot or geometric stress is calculated using a model with a geometric discontinuity present and is compared with the nominal stress calculated in an identical model without the discontinuity. In the case of this particular research, the structural discontinuity is the ring flange connection, thus, two models of the turbine are compared, one with the flange connection and the reference model without the connection. Using the FE model, the nominal and hotspot stresses are calculated based on an idealised structure with no fabrication or installation misalignment and all material behaviour is considered linear as specified by the DNV GL recommended practice guidelines [101].

There are no existing guidelines which define the exact location of butt welds in the current literature or design standards. The IEC [217] merely specifies that the distance from the weld transition to the end of the fillet radius (l) must correspond to at least half of the wall thickness (b):

$$l + r \geq b/2 \quad (4.55)$$

Where the fillet radius (r) must be at least 10mm from the top of the flange surface to prevent edge distortion. Similar geometric limits are also specified in DIN 18088-

3, as noted by Seidel [46], however, no defined location is provided. The IEC states that the neck height should only be added favourably, however, these standards place no limits or recommendations on the maximum distance from the weld to the flange plate. As a result, in this work the butt weld is selected to be located in the most onerous location along the tower wall in a bid to explore the worst-case scenario.

In this model, the tower shell elements are connected to the flange solid elements using a shell-to-solid coupling constraint. Such connections can give rise to artificial stress concentrations and it is recommended by Abaqus to locate these couplings at least a distance of the shell thickness away from a region of interest. As a result, the connection is placed at a distance of 1m from the top surface of the flange and stress readings are taken to a height of 0.8 m above the flange's upper surface. The 0.8 m distance was selected based on the maximum stresses arising in similar flange connections in Seidel [46].

The DNV recommended practice standard [101] states that for structures which are subject to several types of loading, in this case, axial forces and bending moments, the SCFs can be calculated for each loading scenario individually and later superimposed on the individual stresses. Consequently, two calculations are required for this work. The axial SCFs are found by applying an axial load of 10,000 kN at the turbine tower top and comparing the stresses calculated in the reference model against the stress produced in the model with the proposed flange. The bending moment SCFs are calculated by placing a concentrated moment of 100,000 kNm at the top of the turbine tower. This analysis is similarly carried out on the reference model and the model with the ring flanges installed, comparing the induced stresses.

In this study, two locations of SCFs are investigated separately, namely, the flange and the tower wall. To enable this, the flange component was divided into two sets: the flange, which contained all the elements within the flange thickness and the fillet radius, and the tower wall, containing all the elements above the fillet radius and below the 0.8m limit discussed above. In the reference model with no flange, these groups were identified based on the height of the flange and the fillet radius. SCFs were then separately calculated for both the flange sets and tower wall sets for both the axial and bending cases. In the case of the flange set, a maximum SCF was of interest, which was obtained by comparing the maximum

stress from the reference model with the maximum stress in the model with the flange present. For the tower wall set, the distribution of SCFs along the height of the tower wall was also of interest. As a result, each element in the reference set was paired with the corresponding element at the same location in the model with the flange present. This allowed for the comparison of stresses to be conducted at each individual element, creating a SCF distribution.

4.5.9 Stress Concentration Factor Validation

The validation of the FE model carried out in Section 4.5.6 is extended in this section to include the calculation of SCFs for both the flange segment and the tower wall.

Limited data on SCFs for internally bolted ring flanges are available in the literature, making it challenging to establish a direct comparison with the flange set. The closest comparable case found by the author is in the DNV standard, which provides a SCF for a stiffened shell. This SCF is provided as a basis for preliminary validation and is calculated using the following equation [101]:

$$SCF = 1 + \frac{0.54}{\alpha} \quad (4.56)$$

$$\alpha = 1 + \frac{1.56 \cdot t \sqrt{r \cdot t}}{A_r} \quad (4.57)$$

where A_r is the area of the ring stiffener without the effective shell, r is the radius of the shell and t is the thickness of the shell plating. In the case of the wind turbine tower flange connection designed in Section 4.3, the DNV SCF is calculated as:

$$\alpha = 1 + \frac{1.56 \cdot t \cdot \sqrt{r \cdot t}}{A_r} = 1 + \frac{1.56 \times 0.039496 \times \sqrt{(5 - 0.039496) \times (0.039496)}}{\pi(5 - 0.039496)^2 - \pi(5 - 0.2627248)^2} = 1.004 \quad (4.58)$$

$$SCF = 1 + \frac{0.54}{1.004} = 1.5378 \quad (4.59)$$

The DNV SCF is compared to the SCF of 1.18 obtained from the FE model for the flange connection under axial loading. The maximum SCFs calculated in the FE model occur on the outer surface of the flange under both axial and bending moments. This location is illustrated in Figure 4.21, in which a segment of the flange connection is subjected to bending loads. The SCF calculated using the DNV standard is significantly higher than the corresponding FE model result, leading

to a substantial overestimation of the stresses experienced by the flange. This overestimation, in turn, would result in a severe underestimation of the connection's fatigue life. Consequently, the SCF provided by the DNV standard is not suitable for detailed fatigue analysis, as it produces overly conservative stress predictions that will not accurately reflect the actual structural performance of the flange connection.

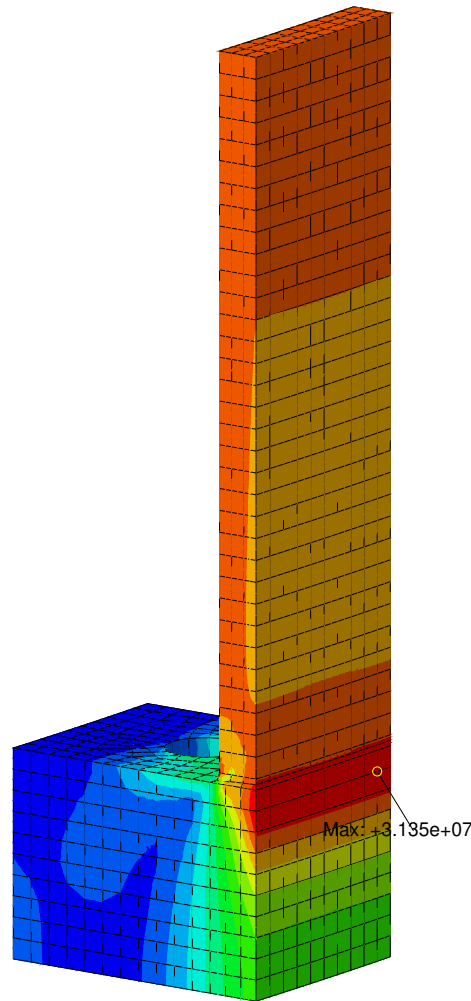


Figure 4.21: Stress distribution in ring flange segment

The SCFs calculated along the tower wall are validated against the analytical equation proposed by Seidel [46], as discussed in Section 4.5.7. This equation calculates the stress distribution along the inner shell wall of the wind turbine tower in the proximity of the flange subjected to a nominal axial stress. The resulting stress distribution is compared with the stresses calculated in the FE model subjected to identical loading conditions.

The results from the analytical solution and FE model are presented in Figure 4.22. The analytical solution exhibits behaviour similar to that observed in the FE model, with both curves following comparable trends along the length of the tower wall. However, the analytical equation tends to overestimate the bending stresses induced by the presence of the flange in comparison with the FE model. Specifically, the analytical solution yields a maximum SCF of 1.089, while the FE model calculates a lower SCF of 1.066. Furthermore, the maximum SCF in the analytical solution occurs closer to the mating surface of the flanges, peaking at 0.556 m, compared to the FE model's maximum SCF, which occurs at 0.574 m. Despite these minor discrepancies, the overall close agreement between the two results reinforces confidence in the accuracy of the FE model's stress predictions.

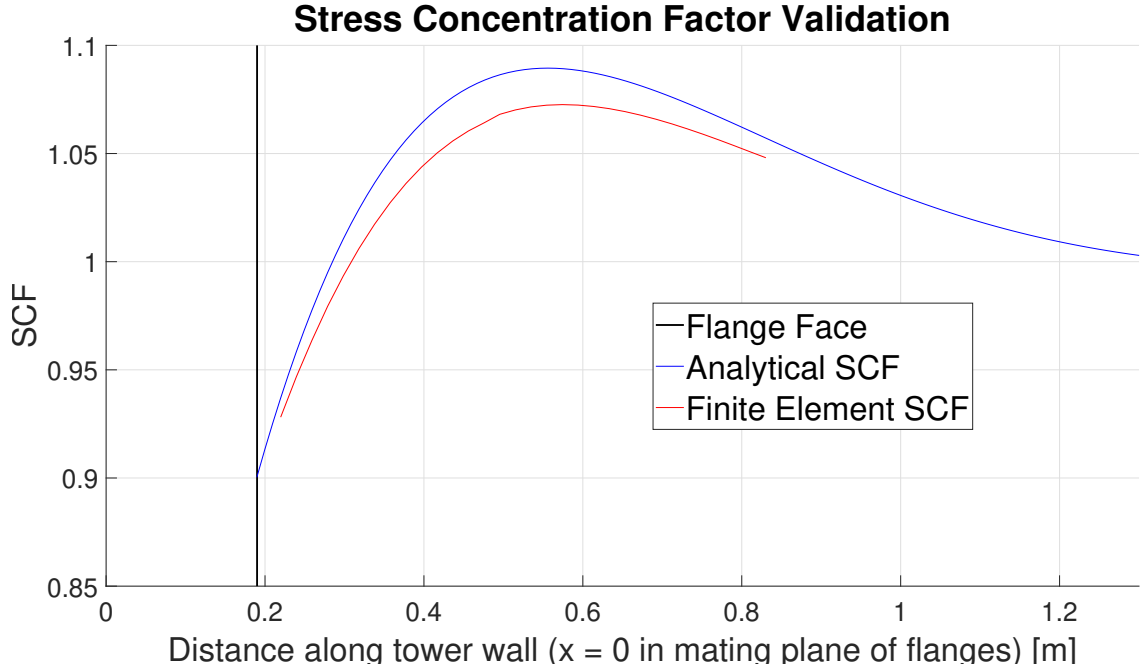


Figure 4.22: Stress concentration factor distribution: finite element model vs. analytical approach

A similar pattern of stress distribution is additionally observed in the FE model when the flange connection is subjected to bending moments, as illustrated in Figure 4.23. In this image, the flange connection is located at the bottom of this image. The contour plot illustrates the stress distribution as the distance from the flange connection increases, displaying a pattern of stress concentrations consistent with those calculated by the analytical equation for axial loading.

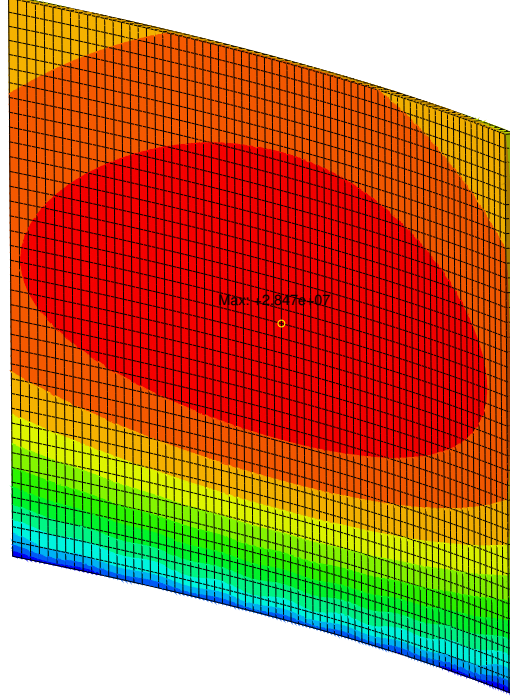


Figure 4.23: Stress distribution in tower shell wall with ring flange connection present

4.6 Fatigue Analysis via Composite Model Framework

The fatigue analysis was conducted following the exact same methodology and environmental conditions as outlined in Chapter 3. The only exception arises in the calculation of stresses at the weld connection, which in this chapter are determined using:

$$\sigma_E(t) = SCF_{Axial} \cdot \frac{-N}{A} + SCF_{Bending} \cdot \frac{M_{FA}(t) \cos(\theta)}{Z} + SCF_{Bending} \cdot \frac{M_{SS}(t) \sin(\theta)}{Z} \quad (4.60)$$

In this equation $\sigma_E(t)$ is the stress at the tower base at a time t , during simulations conducted under environmental conditions E . SCF_{Axial} and $SCF_{Bending}$ are the axial and bending SCFs, which are calculated by the FE model to be 1.0663 and 1.0632, respectively. N is the axial force, A is the area of the tower cross-section, M_{FA} and M_{SS} are the bending moments in the fore-aft and side-to-side directions

and θ represents the angle of the resultant moment vector relative to the fore-aft direction, given by:

$$\theta = \tan^{-1} \left(\frac{M_{SS}(t)}{M_{FA}(t)} \right) \quad (4.61)$$

The section modulus Z is derived from the second moment of area I for a hollow circular cross-section:

$$Z = \frac{D_{outer}}{2I}, \quad I = \frac{\pi (D_{outer}^4 - D_{inner}^4)}{64} \quad (4.62)$$

Here, D_{outer} and D_{inner} are the outer and inner diameters of the tower cross-section, respectively.

4.6.1 Fatigue Life Evaluation with Stress Concentration Factors

The annual damage rate at the butt weld connection for the proposed flange design connecting the wind turbine tower and transition piece for a site located off the west coast of the United States is calculated as 0.0407. Table 6.12 extends this result, calculating the lifespan of the turbine using the reciprocal of the damage rate as detailed in Section 3.4.3. The table then compares the lifespan of the butt weld considering the influence of the flange calculated using the composite model, with the lifespan calculated with the traditional approach in Section 3.5.

Table 4.2: Fatigue life of butt weld at NREL benchmark site

	Traditional Approach	Composite Model Framework
Lifespan (Years)	30.45	24.57

The fatigue life calculated using the composite model is 24.57 years, marking a reduction of nearly 6 years compared to the traditional approach. This significant decrease in the lifespan of the turbine tower is a direct consequence of stress concentration at the ring flange and highlights the importance of accounting for stress concentrations in the fatigue analysis framework. The SCFs calculated for both the axial and bending stresses act to increase the local stresses by approximately $\approx 7\%$. This translates into a $\approx 14\%$ increase in the amplitude of the stress cycle ranges which are incorporated into the bi-linear S–N curve. As the slope of the S–N curve in the high-cycle region is steep ($m = 5$), increases in stress ranges can lead to large

reductions in fatigue life. As a result, the bending-induced stress amplification at the ring flange accelerates local fatigue damage accumulation. This increase in the stresses results in a 19.3% reduction of the expected lifespan of the wind turbine, demonstrating the sensitivity of the fatigue life to the localised stress effects.

4.7 Conclusions

This chapter presents a composite modelling approach to generate accurate stress time histories at critical locations on the 15-MW wind turbine structure. The composite model integrates the capabilities of both MBD simulations and FE analysis, leveraging the strengths of both methods. This integrated approach enables precise fatigue life assessments of critical wind turbine components, ensuring accurate predictions of operational lifespan. Within this composite model, the MBD model is used to simulate environmental loading conditions from wind and waves while incorporating the turbine's control systems using NREL's Reference OpenSource Controller (ROSCO) [204]. This provides a holistic representation of the wind turbine's global structural response during operational conditions. However, this model alone cannot capture localised stress concentrations arising from geometric discontinuities, such as the flange connection between the tower and the transition piece. To address this limitation, a detailed FE model of the wind turbine is developed, incorporating a specifically designed internally bolted ring flange acting to connect the wind turbine tower base to the transition piece. The FE model of the turbine is used to calculate SCFs, which accurately account for the effects of local geometries. To ensure the model's reliability, it is validated against NREL's specifications and compared with an analytical SCF approach.

By combining the stress time histories from the MBD model with SCFs from the FE model, comprehensive stress time histories are produced, encompassing both the global structural response and local stress distributions. Based on the comprehensive stress-time histories, a detailed fatigue analysis is performed, implementing environmental conditions based on NREL's West Coast site. This fatigue assessment focuses particularly on the butt weld which connects the ring flange connection to the wind turbine base. The results of this fatigue analysis were then compared with the traditional approach, which neglects details such as the ring flange connection. The consideration of SCFs in the local geometry proved to be crucial for the estimation of the fatigue life of components. Using the traditional approach,

the fatigue life was estimated to be 30.45 years. However, when considering the geometric effects of the ring flange, the fatigue life was reduced by 19.3%, yielding a more realistic estimate of 24.57 years. This reduction placed the wind turbine's lifespan much closer to the expected service life of a modern wind turbine, significantly decreasing the factor of safety in the design.

This integrated modelling approach provides a precise methodology for the fatigue life assessment of critical wind turbine components, increasing confidence in long-term fatigue performance. By implementing this refined analysis, engineers can better estimate fatigue loads, leading to more informed design choices regarding flange dimensions, tower diameters and shell thicknesses. Ultimately, this contributes to the development of safer, more sustainable, and economically viable wind turbine structures, supporting the long-term objective of reducing the LCOE production.

While this framework enhances the precision of fatigue life assessments, it is subject to several limitations. The deployment of wind turbines in offshore environments exposes their components to highly corrosive marine conditions. In this study, it is assumed that corrosion protection measures remain fully effective throughout the turbine's operational life. However, in reality, these protective systems typically have a limited lifespan under such harsh conditions, eventually leading to the progressive degradation of structural components over time. Moreover, the environmental data provided by NREL, while valuable as a benchmark set of parameters characterising real site conditions, is limited in its representation of sea states. Specifically, the dataset assigns a single sea-state condition to each wind speed, thereby overlooking the natural variability between wind and wave parameters. This simplification reduces the accuracy with which realistic loading scenarios are captured in the fatigue analysis.

These limitations are addressed in the subsequent chapter, which aims to provide a more realistic representation of both the loading conditions and corrosion processes experienced by offshore wind turbines throughout their service life.

Chapter 5

Impact of Corrosion and Metocean Conditions on Fatigue Performance via a Composite Modelling Framework

5.1 Motivation

Chapter 1 provides an overview of Ireland’s renewable energy landscape, with a particular focus on the share of electricity generated from wind energy. Within Europe, Ireland has emerged as one of the leading countries in renewable energy generation, primarily due to its extensive onshore wind infrastructure. However, a key distinction between Ireland and other renewable energy leaders, such as Denmark, lies in the underdevelopment of its offshore wind sector. Despite Ireland’s significant wind resources, driven by its proximity to the Atlantic Ocean, Ireland currently has no offshore wind capacity, representing a significant untapped opportunity, given Ireland’s vast offshore wind potential. To explore this opportunity, this chapter investigates the installation of an IEA 15-MW wind turbine in two locations on the east and west coasts of Ireland.

A critical challenge in realising this potential lies in ensuring the long-term structural resilience of turbines in the harsh marine environment. Accurate fatigue assessments are therefore fundamental for ensuring the structural integrity, opti-

mising structural steel requirements, planning maintenance strategies [231], and assessing life extension possibilities of offshore wind turbines. Accordingly, this chapter aims to provide a comprehensive evaluation of the 15-MW wind turbine's expected life span at the two proposed offshore sites, which have the potential to enhance Ireland's offshore capacity and capitalise on the country's abundant wind resources.

In Chapters 3 and 4, fatigue analyses were conducted using both the standard approach and a proposed composite model framework for the 15-MW wind turbine. These analyses were initially performed for a site off the east coast of the United States, providing a baseline understanding of the turbine's fatigue performance under typical offshore conditions. In this chapter, the analysis is extended to the two proposed Irish sites, enabling a site-specific evaluation of the turbine's fatigue life, taking into account the unique environmental conditions at each location. By comparing the results across these sites, the study provides valuable insights into how geographical variations influence the structural integrity and operational lifespan of the turbine.

In addition to fatigue loading, the harsh marine environment in which wind turbines operate subjects turbines to corrosion-induced degradation. This has proven to significantly impact their long-term durability, with studies indicating that approximately 8% of offshore structural failures are attributed to corrosion-related damage [232]. Findings such as this give rise to one of the key objectives of this work, which is to investigate the impact of material thinning in fatigue-prone areas, such as connection flanges, on the lifespan of wind turbine support structures. To address this, the fatigue analysis in this chapter is expanded to incorporate not only the effects of wind and wave loading but also the effects of the corrosive marine environment. This is achieved by integrating corrosion wastage effects into the parametric FE model within the composite model framework. By accounting for material thinning in fatigue-prone areas, the analysis provides a more realistic assessment of the turbine's fatigue life and structural performance under real-world operating conditions.

5.2 Irish Metocean Conditions

The application of accurate met-ocean data is a critical component in the design and assessment of offshore wind turbines [70]. The use of accurate data can lead to significant cost savings in offshore wind projects, as demonstrated by Hegseth, Bachynski, and Leira [117], who reported an 11% decrease in the construction costs of a turbine tower and platform when precise met-ocean data was implemented. Furthermore, the incorporation of in-situ environmental measurements significantly reduces the uncertainties associated with the fatigue lifetime estimation of offshore structures [194]. Consequently, for the proposed wind turbine sites off the Irish coast, obtaining accurate environmental data was essential to facilitate precise fatigue lifetime assessments.

To obtain such data, this study utilised the Irish Marine Data Buoy Observation Network (IMDBON), a network of weather buoys strategically deployed in the surrounding waters of Ireland as shown in Figure 5.1. Operated by the Marine Institute in collaboration with Met Éireann and the UK Met Office, the IMDBON provides real-time meteorological data crucial for marine and atmospheric studies. The data collected from these buoys has provided this study with accurate in-situ measurements for a number of meteorological parameters, including wind speeds, significant wave heights, peak spectral wave periods and both wind and wave directions. These measurements were subsequently processed in accordance with the IEC 61400 [213] design guidelines and the recommendations of the EU-funded HIPER-WIND project [233].

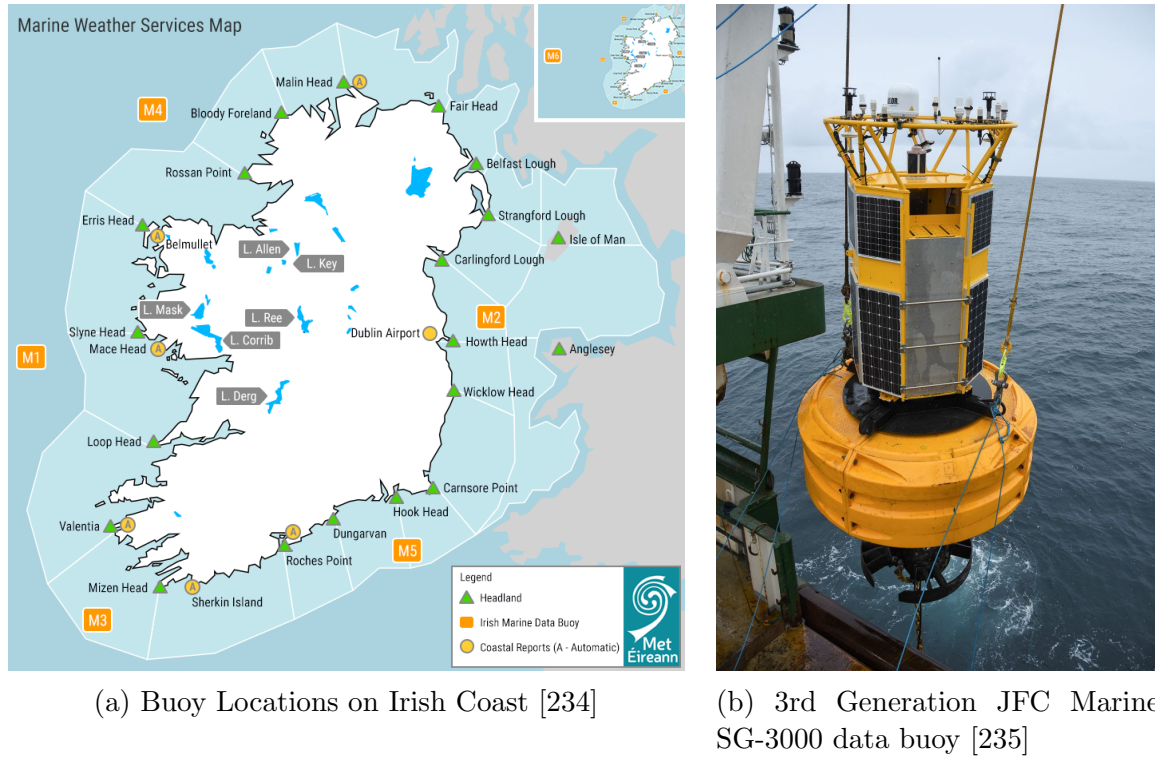


Figure 5.1: Irish marine data buoy observation network

In this study, data was sourced from two buoy locations on the east and west coast of Ireland, namely the M2 (east coast) and M4 (west coast) buoys, respectively, which have been selected as potential wind farm sites for assessment. The weather buoys deployed in these locations are JFC Marine SG-3000 data buoys, which are the third generation of buoys installed at these sites [236]. These buoys have a height of 6 meters and are equipped with an anemometer positioned approximately 4.5 meters above sea level [237]. Data collection from these sites has been ongoing since 2001, however, the dataset exhibits high intermittency due to factors such as scheduled maintenance, equipment malfunctions, and extreme weather conditions. For instance, Figure 5.2 illustrates a weather buoy beached on the coast of Ballinskelligs, Co. Kerry, after storm Eowyn, highlighting the challenges posed by extreme weather events. However, the reliability of data collection improved significantly in 2019 after the deployment of the third generation of buoys. Consequently, the dataset utilised in this work spans from 2019 to 2024. Owing to the large quantity of usable data available between 2019 and 2024, the interpolation of missing values was deemed unnecessary [211], avoiding any possible bias that interpolation

would introduce. In any instances where one or more of the key parameters being investigated were not available, the data entry was disregarded.



Figure 5.2: Buoy bocht- Irish marine data buoy beached on the coast of Ballinskelligs, Co. Kerry

The design regulations outlined in IEC 61400-3-1 [213] stipulate that the correlation of wind and wave conditions should be considered in the design of wind turbines for site-specific locations via the use of long-term joint probability distributions of the relevant parameters. The HIPERWIND project [233] recommends that the significant wave heights are modelled conditionally based on wind speed, while peak spectral wave periods should be modelled conditionally on the significant wave height. Following these recommendations, the long-term joint probability distributions were constructed from the IMDBON data using the method of bins. The maximum bin intervals of the joint probability distributions were defined as specified in the IEC standard [213], with maximum allowable intervals of 2 m/s for wind speeds, 1 meter for significant wave heights and 1 second for peak spectral wave periods. The resulting joint probability distributions are displayed in Figure 5.3. Once established, these distributions form the basis for fitting statistical models, as demonstrated in Vanem et al. [233]. The details of this process are provided in the following sections.

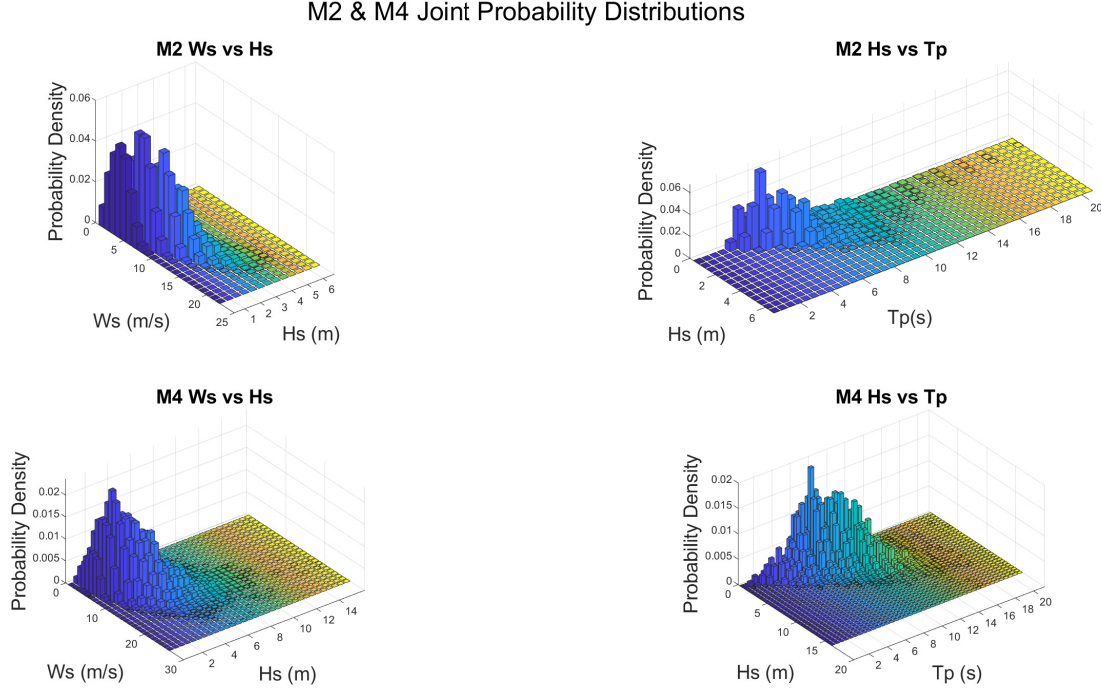


Figure 5.3: M2 (east coast) and M4 (west coast) metocean conditions joint probability distributions

5.2.1 Characterisation of Wind Speeds

Wind speeds at the buoys are recorded as 10-minute averages and reported on an hourly basis. To serve as suitable inputs for the composite model of the wind turbine, the wind speed velocities, which were recorded at a height of 4.5 meters above sea level, must be converted to the hub height of the turbine at 150 meters. This was achieved using the wind shear power law, applying a power law exponent (α) of 0.14, under normal wind conditions as per IEC 61400-3 [213].

$$V_{\text{hub}} = V_{\text{ref}} \left(\frac{Z_{\text{ref}}}{Z_{\text{hub}}} \right)^{\alpha} \quad (5.1)$$

To analyse this data, the method of bins was first applied to the wind speeds based on 2 m/s bin widths. Following this, a hybrid model combining a 2-parameter Weibull probability distribution with a Generalised Pareto Distribution (GPD) was fitted to the wind speed distributions. The Weibull distribution was implemented

to capture the body of the distribution, while the GPD was employed to accurately capture the tails of the distribution. Accordingly, the cumulative distribution function for wind speeds can be expressed as:

$$F_U(u) = \begin{cases} F_W(u) & \text{if } 0 \leq u \leq v \\ F_W(v) + [1 - F_W(v)]G(u) & \text{if } u > v \end{cases} \quad (5.2)$$

Where F_W is the 2-parameter Weibull cumulative distribution and $G(u)$ is the GPD. The 2-parameter Weibull cumulative distribution can be expressed as:

$$F_W(u; \lambda_W, k_W) = 1 - \exp \left(- \left(\frac{u}{\lambda_W} \right)^{k_W} \right) \quad (5.3)$$

where u is the wind speed, λ_W is the scale parameter and k_W is the shape parameter.

The cumulative distribution function of the GPD can be expressed as:

$$G(u; \xi, v, \sigma_G) = \begin{cases} 1 - \left(1 + \frac{\xi(u-v)}{\sigma_G} \right)^{-\frac{1}{\xi}} & \text{if } \xi \neq 0 \\ 1 - \exp \left(-\frac{u-v}{\sigma_G} \right) & \text{if } \xi = 0 \end{cases} \quad (5.4)$$

where ξ is the shape parameter, v is the threshold and σ_G is the scale parameter. The wind speed distribution parameters: λ_W , k_W , ξ , σ_G and v are fitted to the data presented by the weather buoys using the maximum likelihood estimation method. The results of this hybrid fitting approach can be observed in Figure 5.4. The fitted probability distributions exhibit strong agreement with the measured data across the full range of wind speeds, successfully capturing both the central trend and the tail behaviour of the observed distributions. This strong correspondence confirms the suitability of the hybrid Weibull–GPD model for representing site-specific wind speed characteristics at these offshore locations.

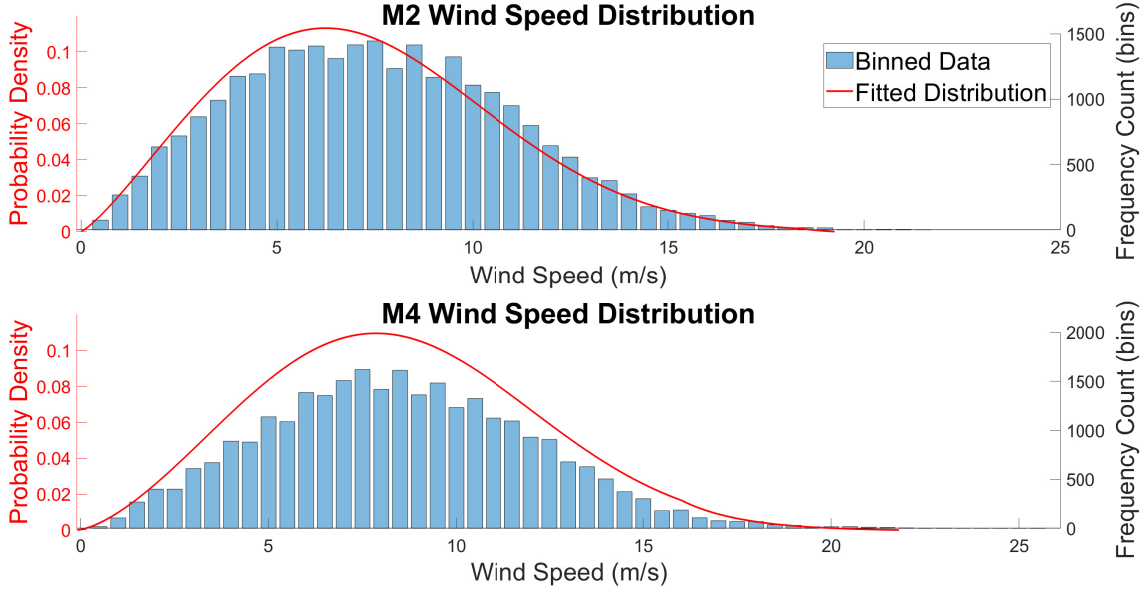


Figure 5.4: M2 (east coast) and M4 (west coast) wind speed probability distributions

5.2.2 Turbulence Intensity Model

Despite the rich data supplied by the IMDBON, no data regarding the corresponding turbulence intensities for wind speeds is available. Consequently, a turbulence model was adopted from Wang et al. [212] as recommended by International Electrotechnical Commission [213] relating wind speed to a given turbulence intensity. This model is presented in Figure 3.2 of Chapter 3. To ensure a conservative assessment, a site classification of Class A was selected, representing the highest turbulence category.

5.2.3 Characterisation of Significant Wave Height

In alignment with the recommendations of the HIPERWIND project [233], the significant wave heights were modelled conditionally based on the wind speeds. To achieve this, the binning method applied to the wind speed data was similarly applied to the significant wave height data. Specifically, all significant wave heights corresponding to one wind speed bin (e.g. 2-4 m/s) were grouped together. The probability distributions associated with each of these grouped significant wave heights were then determined using the method of bins. The resulting conditional probability distributions were subsequently modelled using 3-parameter translated

Weibull distributions. An individual Weibull distribution was generated for each wind speed bin, resulting in 3 x n Weibull parameters based on n wind speed bins. The 3-parameter Weibull probability density function implemented can be expressed as:

$$f_{H_S|U}(u; k_{W,h}, \lambda_{W,h}, \gamma_{W,h}) = \frac{k_{W,h}}{\lambda_{W,h}} \left(\frac{u - \gamma_{W,h}}{\lambda_{W,h}} \right)^{k_{W,h}-1} \exp \left[- \left(\frac{u - \gamma_{W,h}}{\lambda_{W,h}} \right)^{k_{W,h}} \right] \quad (5.5)$$

where $\gamma_{W,h}$ represents the location parameter, $\lambda_{W,h}$ represents the scale parameter, and $k_{W,h}$ represents the shape parameter. The parameters of each wind speed bin were fitted using the maximum likelihood estimation.

5.2.4 Characterisation of Peak Spectral Wave Period

In a similar manner to the significant wave heights, a conditional probability distribution was constructed for peak spectral wave periods based on the significant wave height. The method of bins was applied to the raw data, enabling the creation of a joint probability distribution between significant wave heights and peak spectral wave periods. This allowed for a conditional distribution of peak spectral wave periods, given a significant wave height to be calculated for each significant wave height bin. These conditional distributions were then approximated using lognormal distributions, which can be expressed as:

$$f(x; \mu, \sigma) = \frac{1}{\tau_\sigma(h) \sigma \sqrt{2\pi}} \exp \left(- \frac{(\ln(\sigma) - \mu_\sigma(h))^2}{2(\tau_\sigma(h))^2} \right) \quad (5.6)$$

Where the parameters $\mu_\sigma(h)$ and $\tau_\sigma(h)$ are modelled as a function of significant wave height, assuming a power-law relationship based on the mean and standard deviation of the distributions of each significant wave height bin:

$$\mu_\sigma(h) \equiv \text{mean}[\ln \sigma_u | (H = h)] = a_1 + b_1 h^{c_1} \quad (5.7)$$

$$\tau_\sigma(h) \equiv \text{std.dev}[\ln \sigma_u | (H = h)] = a_2 + b_2 h^{c_2} \quad (5.8)$$

Where the parameters a_1 , b_1 , c_1 , a_2 , b_2 , c_2 are fitted to the data employing the non-linear least squares approach.

5.2.5 Comparison of Metocean Conditions Across Selected Offshore Locations

The application of this statistical modelling characterises the environmental conditions at the two sites in compliance with the IEC [213] and Hyperwind methodologies [233], providing the necessary environmental parameters for integration into the composite model framework discussed in Section 4.2. The resulting probability distributions accurately capture the frequency of occurrence for the various environmental conditions at the two sites, making them essential for long-term fatigue damage estimation.

Before carrying out the detailed fatigue analysis, a preliminary comparison of the M2 and M4 sites is conducted by examining the wind speeds, significant wave heights and peak spectral wave periods at the two sites. For context, these sites are additionally compared with the NREL Benchmark conditions, which were implemented in Chapters 3 and 4, as well as metocean data reported by Li, Gao, and Moan [238] for a site located in the central North Sea, a region that hosts a significant share of Europe’s offshore wind capacity [239]. This initial assessment provides a basis for evaluating the relative severity of the environmental conditions at the Irish sites and for benchmarking them against established international offshore wind locations.

Figure 5.5 displays the probability distribution of wind speeds for the four selected sites. The wind speeds at M4 are notably higher in comparison with M2, which is expected given M4’s location on the west coast of Ireland. This site is fully exposed to the Atlantic Ocean, with the nearest landmass to the west being the United States. The prevailing southwesterly wind blows unobstructed across the ocean, allowing for the development of high wind speeds. In contrast, the M2 site is located in the Irish Sea on the east coast, where wind speeds are reduced as the winds travel across the island of Ireland. The North Sea splits the M2 and M4 sites, having a higher probability of higher wind speeds in comparison with M2 but a lower probability in comparison with M4. This is a direct consequence of Ireland acting as a topographic barrier to the westerly winds, which are dominant in the North Sea region. However, the North Sea, in contrast to the Irish Sea, is a vast, open body with drastically reduced shelter, particularly when exposed to weather systems arising from the north, giving rise to increased wind speeds.

NREL's benchmark site has a peak probability similar to that of both the M4 site and the North Sea, however, the NREL site has a notable increase in probability of high wind speed in excess of 15m/s stemming from the high number of Tropical Cyclones which occur along its coastline.

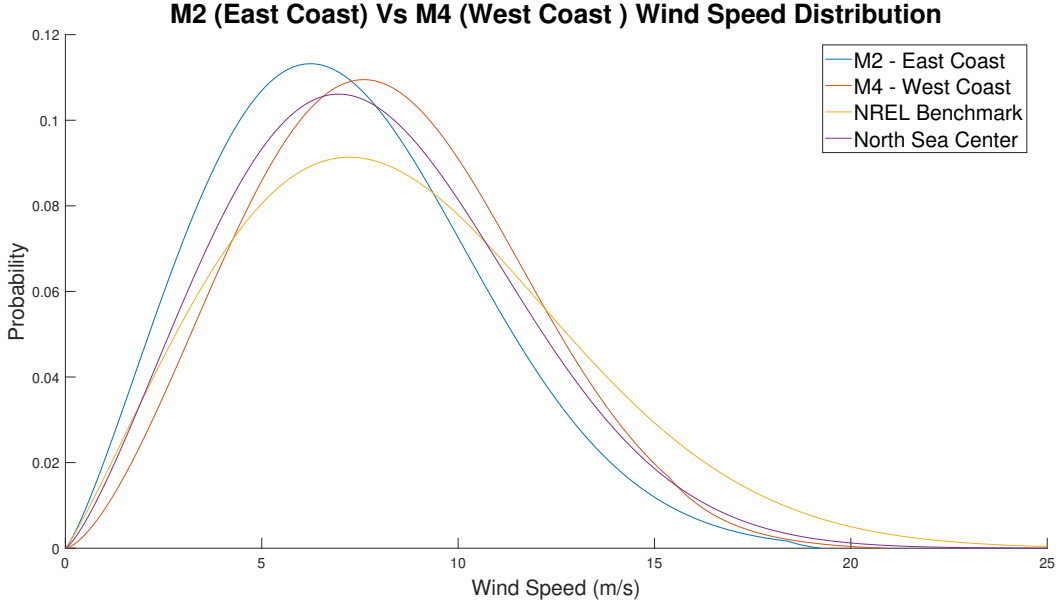


Figure 5.5: Comparison of wind speed probability distributions at M2 (east coast), M4 (west coast), NREL benchmark, and North Sea sites

These observations are quantified by the Weibull distribution parameters presented in Table 5.1. The notably higher scale parameter at the M4 site confirms that the west coast location experiences consistently greater mean wind speeds compared to M2. Furthermore, the M4 site exhibits a higher shape parameter indicating a narrower and more peaked distribution, signifying more uniform and stable wind conditions in comparison with M2. As anticipated, the North Sea parameters reflect the observations made above, with both the scale and shape parameters falling between those of the M2 and M4 sites. Conversely, the west coast of the United States displays a large-scale parameter and a relatively low shape parameter, reflecting higher average wind speeds and a broader distribution, indicating a powerful yet highly variable wind distribution.

The geographical location of each site and its prevailing wind direction, which strongly influence the local wind speeds, also have a significant impact on the wave climate. Table 5.2 presents the mean significant wave height for each wind speed

Table 5.1: Comparison of Weibull parameters for wind speed probability distributions at M2 (east coast), M4 (west coast), NREL benchmark, and North Sea sites

Site:	M2 (east coast)	M4 (west coast)	NREL	North Sea
λ_W	8.15	9.28	9.77	8.92
k_W	2.22	2.52	2.12	2.29

bin at the four sites. Consistent with the wind speed observations, the M4 site experiences considerably larger significant wave heights compared to the M2 site, resulting from its exposed location in the Atlantic Ocean. The uninterrupted wind and greater fetch over the Atlantic provide optimal conditions for wave development, allowing for the formation of consistently higher waves at M4 compared to M2. Following the same trend observed in the wind speeds, the North Sea experiences significant wave heights that lie between those of the sheltered M2 site and the exposed M4 site, reflecting its intermediate exposure conditions. In contrast, at NREL’s benchmark site on the west coast of the United States, the significant wave heights are notably lower than those at M4, despite its higher wind speeds. This can be attributed to the prevailing westerly wind at this location and the resulting limited fetch available for wave formation along the U.S. coastline. This acts to restrict the development of large waves even under stronger wind conditions.

Table 5.2: Comparison of mean significant wave height for a given wind speed at M2 (east coast), M4 (west coast), NREL benchmark, and North Sea sites

Wind Speed Bin (m/s)	0-2	2-4	4-6	6-8	8-10	10-12	12-14	14-16	16-18	18-20	20-22	22-24	24-26
H_s (M2) (m)	0.41	0.50	0.70	0.99	1.37	1.84	2.34	2.97	3.46	3.75	4.01	-	-
H_s (M4) (m)	1.79	1.98	2.12	2.52	3.10	3.86	4.65	5.45	6.02	6.51	6.55	7.26	6.85
H_s (NREL) (m)	-	-	1.10	1.18	1.32	1.54	1.83	2.18	2.59	3.06	3.61	4.02	4.52
H_s (North Sea) (m)	0.47	0.75	1.13	1.57	2.08	2.64	3.24	3.88	4.56	5.28	6.03	6.81	7.62

These trends are also evident in Table 5.3, which presents the mean peak spectral wave periods (T_p) for a given significant wave height at the four sites. The M4 site exhibits longer peak spectral periods than the M2 site, primarily due to the greater fetch of the Atlantic Ocean. The extended fetch allows waves to develop over greater distances and time, during which smaller, short-period waves gradually merge and transfer energy into longer waves. This process results in the formation of long-period waves characteristic of open ocean conditions. Stemming from this, the North Sea site again displays intermediate behaviour, with wave periods between

those of M2 and M4, consistent with its moderate exposure and fetch. Reflecting the trend observed in the significant wave heights, the NREL site on the U.S. west coast, despite boasting higher wind speeds, has its wave development restricted by the limited fetch associated with prevailing westerly winds, leading to shorter peak spectral periods than those observed at M4.

Table 5.3: Comparison of the mean peak spectral wave periods for a given mean significant wave height at M2 (east coast), M4 (west coast), NREL benchmark, and North Sea sites

H_s (m)	0-1	1-2	2-3	3-4	4-5	5-6	6-7	7-8
T_p (M2) (s)	4.95	5.85	7.13	8.31	9.47	9.99	11.43	-
T_p (M4) (s)	5.23	8.44	10.19	11.50	12.28	13.22	14.03	14.79
T_p (NREL) (s)	-	7.98	7.55	8.28	9.21	-	-	-
T_p (North Sea) (s)	5.94	6.78	7.51	8.22	8.88	9.54	9.86	10.52

Overall, this comparison of the wind and wave parameters across the four sites highlights distinct environmental characteristics driven largely by the prevailing wind direction, the exposure and the fetch of the sites. The M4 site, with its large fetch across the Atlantic Ocean, experiences the most energetic wave conditions with relatively high wind speeds, while the M2 remains comparatively sheltered. The North Sea site represents an intermediate environment, exhibiting wind and wave conditions between those of the two Irish locations. Conversely, the NREL site presents a unique case which despite experiencing strong winds, its limited fetch under prevailing westerly winds restricts wave development, resulting in less energetic seas.

5.2.6 Definition of Environmental Load Cases

Drawing from the statistical models introduced at the M2 and M4 sites, a range of environmental conditions are selected to generate stress histories for the fatigue analysis process. According to IEC standards, the specified met-ocean conditions must be chosen to adequately represent the full long-term distribution experienced by the wind turbine. To ensure compliance with these requirements while maintaining a manageable computational cost, 11 wind speeds were sampled, for each of these wind speeds, 5 significant wave heights were sampled, and finally, for each significant wave height, 5 peak spectral wave periods were sampled.

To select the environmental parameters from the distributions created, a suitable sampling technique is required. Sampling techniques such as the Monte Carlo approaches have been commonly used throughout the literature [69, 221, 240]. However, with a limited number of simulations, such an approach can yield inaccurate results. Consequently, a stratified sampling technique, namely Latin Hypercube Sampling (LHS), was selected to effectively capture the environmental parameter space. LHS has been proven to reduce the variance in comparison with standard Monte Carlo methods [241] and is widely implemented in fatigue analyses of wind turbine structures [100, 126]. In a similar manner to Chapters 3 and 4, the selected simulations were carried out as per IEC 61400-1 [49] guidelines for fatigue analysis, producing six 10-minute simulations with random seeds for each environmental condition. This resulted in 1,650 (11 wind speeds x 5 significant wave heights x 5 peak spectral wave periods x 6 seeds) simulations. Each simulation was run for a total time of 675 seconds using the MBD model, with the initial 75 seconds being removed to eliminate any initial transient effects. Throughout all simulations, the wind and wave loadings are assumed to be fully aligned.

These simulations were conducted using the MBD model of the IEA 15-MW reference wind turbine described in Chapter 3. While employing this turbine design at both the east and west coast Irish sites facilitates a direct comparison of environmental loading conditions, it is acknowledged that this approach does not reflect typical industry design practice. In reality, wind turbines are tailored to specific site conditions, and a dedicated design process would be carried out for each location. Consequently, the use of the generic design of the 15MW wind turbine will not reflect a true representation of the optimised performance of the 15MW turbine at the east and west coast sites and should be considered a limitation of this work.

5.3 Effect of Corrosion

As discussed in Section 5.1, the marine environment subjects steel structures to corrosive attacks. These corrosive attacks have been shown to directly impact the performance and fatigue life of offshore structures [19]. Recognising these effects, the DNV [101] emphasises that the effects of corrosion must be carefully considered in the design of connections, such as the ring flange connection explored in this work. However, within the existing framework, limited provisions are provided for considering corrosion wastage. In the DNV guidelines, the S-N curve for steel

exposed to seawater with cathodic protection is simply derived by modifying the corresponding S-N curve for steel in air. Specifically, for each stress level, the number of allowable cycles in air is divided by a factor of 2.5 to account for the influence of the marine environment [231].

A novel aspect of this study lies in the FE model's capability to simulate the effect of corrosion on the turbine structure once corrosion protection measures are assumed to have failed. This is achieved by examining the effective thinning of the turbine tower. The corrosion effects are incorporated by simulating material wastage over time, leveraging the parametric design of the turbine tower, ring flange, and monopile/transition piece to reduce the thickness of the components within the FE model. This material wastage reduces the effective area of the component, which gives rise to increased stresses and consequently causes the SCFs to vary with time.

Since corrosion protection measures are only expected to remain effective for a limited number of years [232] and with the repair of such coatings proving to be a difficult procedure in offshore conditions, the analysis is carried out considering a number of durations for which the corrosion protection is expected to remain effective. Upon failure of the corrosion protection coatings, the conventional approach of uniform corrosion is adopted. The rate at which the corrosion occurs is based on the exposure of the individual element. Specifically, a thinning rate of 0.1 mm/year for uncoated carbon steel is applied to the substructure in the atmospheric, submerged, and buried zones while a corrosion thinning rate of 0.3 mm/year is applied in the splash zone in accordance with ISO 19902 [242]. The corrosion zones, as illustrated in Figure 5.6, were defined in this study as per Okenyi et al. [139], which implemented the DNVGL-RP-0416 guidance [128] for the IEA 15-MW wind turbine. These zones were determined based on a constant mean sea level, thereby neglecting the influence of tidal variations. It should be noted that this simplification may affect the accuracy of corrosion predictions in regions near the splash and submerged zones, representing a limitation of the present study. Under these assumptions, the effects of corrosion were examined over the span of 50 years, significantly exceeding the typical 20-year service life of modern wind turbines.

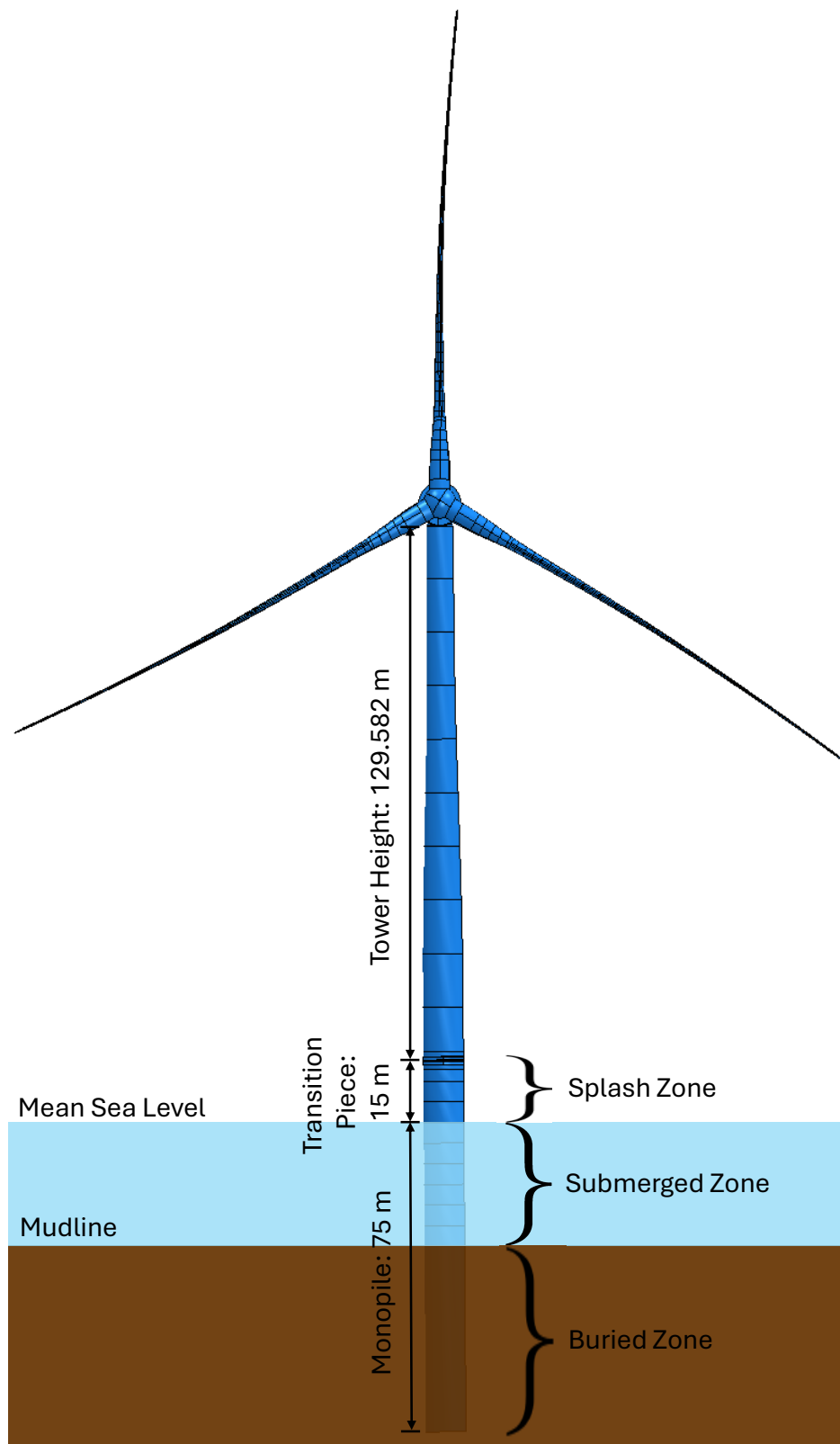


Figure 5.6: Corrosion zone classification for the 15-MW finite element model

The SCFs are calculated on an annual basis as the FE model is updated to account for corrosion-induced material loss. This process alters the substructure's geometry, leading to a global redistribution of stresses at the flange-tower connection. Using these time-varying SCFs, stress time histories for each simulation can be calculated for each individual year using Equation 5.9.

$$\sigma_{E,i}(t) = SCF_{Axial,i} \cdot \frac{-N}{A} + SCF_{Bending,i} \cdot \frac{M_{FA}(t) \cos(\theta)}{Z} + SCF_{Bending,i} \cdot \frac{M_{SS}(t) \sin(\theta)}{Z} \quad (5.9)$$

In this equation, $\sigma_{E,i}$ is the stress at the tower base at a time t during simulations conducted under environmental conditions E , subjected to i years of corrosion wastage. $SCF_{Axial,i}$ and $SCF_{Bending,i}$ are the axial and bending SCFs for each year considering the effects of the corrosion wastage. The remaining parameters remain the same as in Equation 4.60 in Section 4.6. By employing this approach, the study provides a thorough evaluation of how corrosion progressively affects the structural integrity of wind turbine components over time.

5.3.1 Modal Analysis

The impact of material wastage due to corrosion on the dominant first natural frequency of the turbine tower has been investigated through modal analysis. This investigation considers the failure of corrosion protection measures after a number of time intervals. The evolution of the side-to-side natural frequency of the wind turbine tower as it undergoes material wastage due to corrosion is presented in Table 5.4. The table provides the natural frequency values and their percentage change relative to the initial condition at installation, with corrosion protection failure simulated in increments of 5 years. The models used in this comparison include the ring flange connection to ensure consistency throughout.

Table 5.4: Variation in side-to-side natural frequency (%)

Corrosion Exposure (Years)	0	5	10	15	20	30	40	50
Natural Frequency (Hz)	0.170	0.169	0.168	0.167	0.166	0.164	0.161	0.158
Natural Frequency shift (%)	-	0.45	1.05	1.66	2.32	3.71	5.30	7.09

Considering that corrosion protection measures, such as hot-dip galvanised coatings, in the splash zone will typically have a life span of up to 5 years [243], the analysis indicates that over the standard 20-year life span of a wind turbine, the natural frequency of the tower decreases by 1.66%. This relatively small change

illustrates that corrosion wastage has a negligible impact on the natural frequency of the turbine, particularly when compared to potential frequency shifts caused by alternative influences such as soil-structure interaction [163]. This finding reinforces the validity of the composite model, which integrates the detailed FE model with responses from the MBD model. The MBD model calculates the response of the wind turbine tower, as discussed in Section 3.3, based on the modal summation method. This method relies heavily on the natural frequency and modal shapes of the wind turbine tower. As the impact of corrosion wastage is proven to have negligible effects on such parameters, the response of the wind turbine to external loads can be considered independent of corrosive effects. Therefore, a single set of dynamic simulations remains applicable and accurate across the 50-year service life assessed in this study.

5.3.2 Stress Concentration Factors

The geometric SCFs in this chapter are generated using the approach described in Peterson's book [103] discussed in Section 4.5.8. In this approach, geometric SCFs are calculated by comparing the stresses in a reference model with no corrosion or connection considered with the stresses in a model considering the corrosion wastage and the ring flange connection. As described in Section 4.5.8, the SCFs are evaluated at two specific locations, the flange and the tower wall, with the results of the tower wall being used in the fatigue analysis. The flange set contains all the elements within the flange thickness and the fillet radius, and the tower wall set consists of all the elements above the fillet radius and below the 0.8 meter limit. For the tower wall set, the governing SCF is determined by selecting the maximum SCF along its length in a bid to explore the worst-case scenario.

The SCFs for the flange connection, resulting from both axial and bending loads, are presented in Table 5.5. These values correspond to the first 50 years of service, during which corrosion protection is assumed to have failed, leading to corrosion wastage. The SCFs presented are the maximum values occurring exclusively within the flange section. This table clearly demonstrates the importance of considering the effects of corrosion on SCFs, with the axial SCF increasing from 1.18 to 1.96 over the span of 50 years. Over the required 20-year service life of the turbine, the effects of corrosion significantly impact both axial and bending SCFs. Specifically, the axial SCF increases from 1.18 to 1.40, while the bending SCF rises

from 1.16 to 1.37. This indicates that over 20 years, the stress amplification caused by SCFs more than doubles compared to the case where corrosion is neglected, emphasising the critical impact of long-term corrosion on stress concentrations in the flange connection. Under both axial and bending loads, and across the 50 years, the maximum SCFs are observed on the outer surface of the flange, as depicted in Figure 4.21. This location coincides with the area most affected by corrosion wastage, making it a critical point for inspection and maintenance strategies.

Table 5.5: Axial forces and bending moments SCF values in flange for years 1-50

Year	Axial SCF	Bending SCF	Year	Axial SCF	Bending SCF
1	1.1807	1.1579	26	1.4871	1.4503
2	1.1909	1.1677	27	1.5030	1.4659
3	1.2009	1.1771	28	1.5197	1.4821
4	1.2111	1.1870	29	1.5352	1.4974
5	1.2214	1.1968	30	1.5510	1.5129
6	1.2321	1.2071	31	1.5680	1.5296
7	1.2434	1.2176	32	1.5847	1.5462
8	1.2542	1.2280	33	1.6027	1.5638
9	1.2652	1.2385	34	1.6202	1.5810
10	1.2766	1.2493	35	1.6382	1.5986
11	1.2878	1.2599	36	1.6563	1.6164
12	1.3001	1.2715	37	1.6748	1.6349
13	1.3118	1.2826	38	1.6946	1.6542
14	1.3238	1.2940	39	1.7142	1.6736
15	1.3363	1.3057	40	1.7350	1.6941
16	1.3482	1.3169	41	1.7553	1.7140
17	1.3614	1.3292	42	1.7769	1.7352
18	1.3744	1.3415	43	1.7981	1.7563
19	1.3878	1.3540	44	1.8203	1.7782
20	1.4011	1.3666	45	1.8419	1.7995
21	1.4147	1.3792	46	1.8652	1.8224
22	1.4286	1.3929	47	1.8885	1.8456
23	1.4426	1.4065	48	1.9133	1.8701
24	1.4571	1.4209	49	1.9378	1.8942
25	1.4716	1.4351	50	1.9620	1.9181

Table 5.6 presents the maximum SCF occurring along the shell wall near the flange connection, independently subjected to axial and bending loads. These values again correspond to the first 50 years of service, during which corrosion protection is assumed to have failed, initiating corrosion wastage.

Table 5.6: Axial forces and bending moments SCF values in tower shell for years 1-50

Year	Axial SCF	Bending SCF	Year	Axial SCF	Bending SCF
1	1.0663	1.0632	26	1.3207	1.3170
2	1.0747	1.0717	27	1.3337	1.3300
3	1.0830	1.0798	28	1.3469	1.3432
4	1.0915	1.0883	29	1.3600	1.3563
5	1.1002	1.0970	30	1.3732	1.3695
6	1.1091	1.1060	31	1.3873	1.3835
7	1.1182	1.1149	32	1.4011	1.3975
8	1.1273	1.1240	33	1.4154	1.4118
9	1.1366	1.1332	34	1.4301	1.4265
10	1.1461	1.1427	35	1.4451	1.4415
11	1.1553	1.1518	36	1.4599	1.4562
12	1.1654	1.1619	37	1.4756	1.4721
13	1.1753	1.1719	38	1.4913	1.4878
14	1.1854	1.1819	39	1.5076	1.5043
15	1.1954	1.1920	40	1.5246	1.5211
16	1.2056	1.2021	41	1.5415	1.5380
17	1.2163	1.2128	42	1.5592	1.5557
18	1.2272	1.2237	43	1.5762	1.5730
19	1.2383	1.2347	44	1.5947	1.5915
20	1.2495	1.2458	45	1.6127	1.6096
21	1.2606	1.2570	46	1.6317	1.6286
22	1.2723	1.2688	47	1.6510	1.6481
23	1.2840	1.2803	48	1.6712	1.6683
24	1.2959	1.2924	49	1.6915	1.6887
25	1.3079	1.3044	50	1.7121	1.7095

The stress distribution patterns associated with these maximum SCFs are further analysed in Figures 5.7 and 5.8. These figures illustrate the SCF patterns for both axial and bending loads over the first 50 years of service, during which corrosion protection is assumed to have failed, highlighting the significant impact of corrosion wastage on SCF values. In these figures, the stress distribution pattern for the first year is represented by the bottom plot, with each subsequent year corresponding to the next plot above it. It is evident that as corrosion damage progresses, SCF values increase over time, however, the rate of increase is not uniform. A notably larger rise is observed between years 49 and 50 (uppermost curves) compared to the increase between years 1 and 2 (lowermost curves).

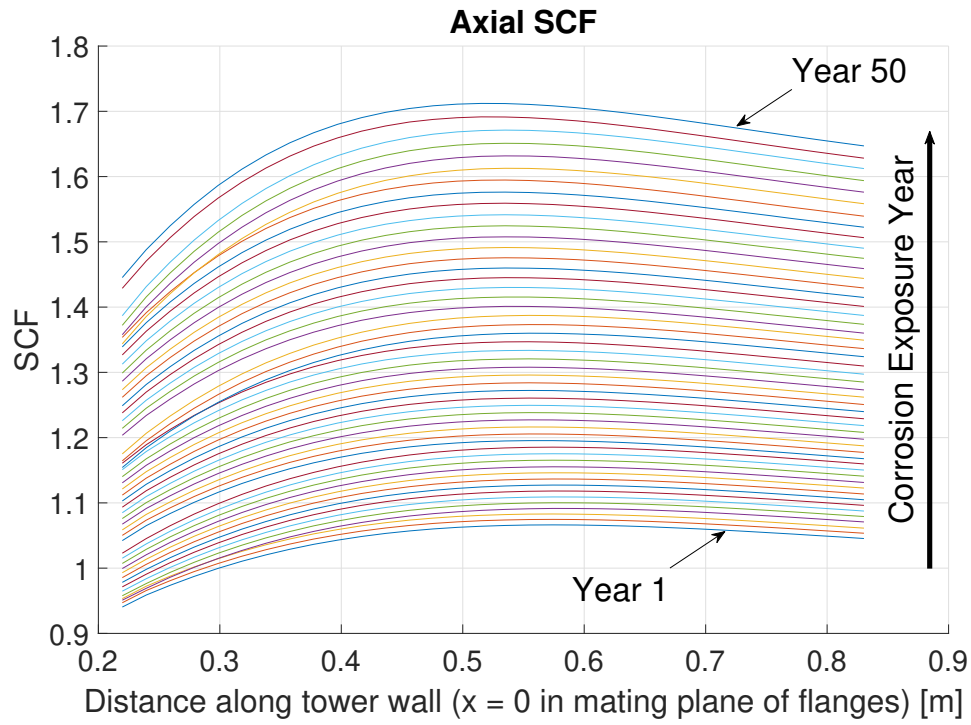


Figure 5.7: Axial force SCF distribution evolution over 50 years of corrosion exposure

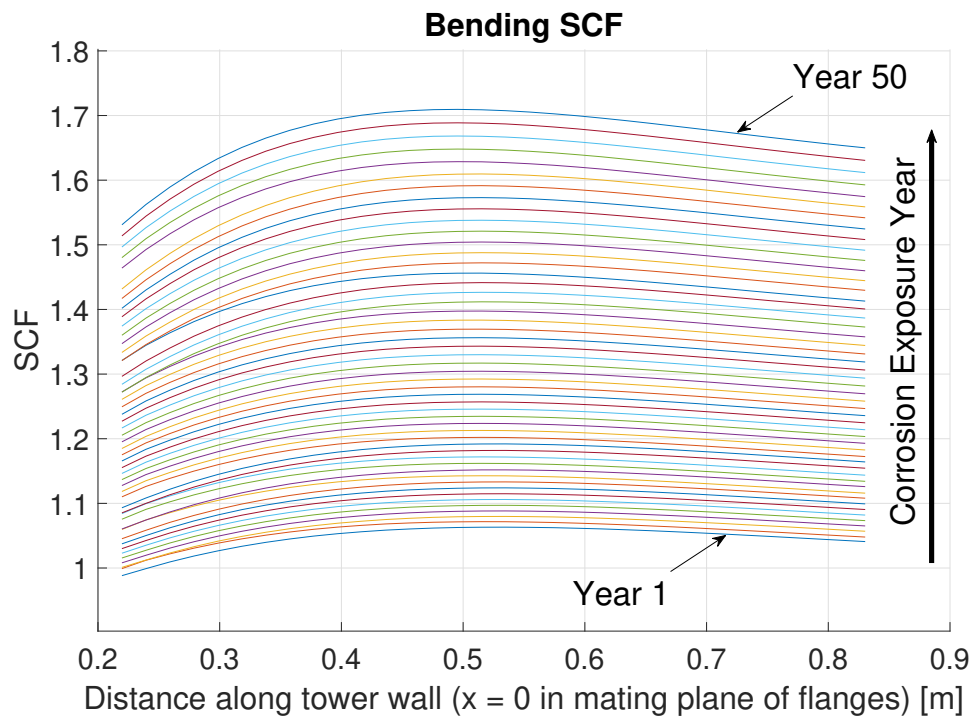


Figure 5.8: Bending moment SCF distribution evolution over 50 years of corrosion exposure

The rate at which the SCFs increase is further analysed in Figure 5.9, which plots the maximum SCF for each stress distribution against the corresponding year under both axial and bending loading conditions. The two plots closely align, highlighting the strong agreement between the SCF associated with axial and bending loads. This alignment would enable SCFs along the tower wall to be calculated purely based on either axial or bending loading in further works, reducing the computational cost of performing the FE simulations while having a minimal effect on the accuracy of the results.

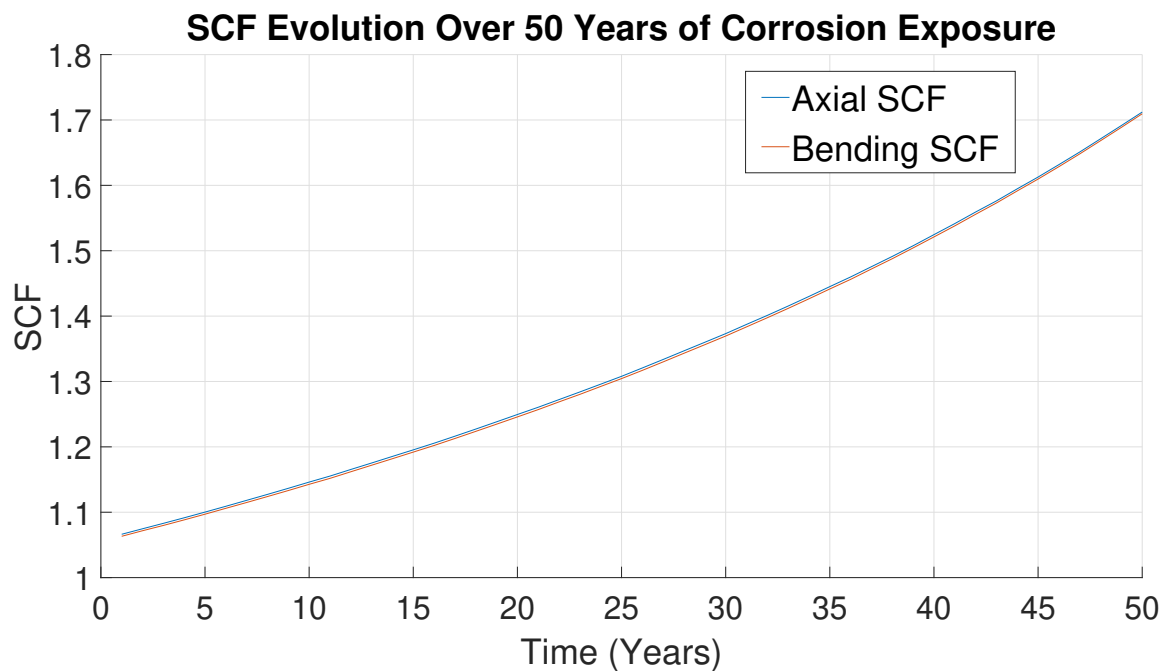


Figure 5.9: SCF evolution over 50 years of corrosion exposure, axial forces vs. bending moments

The variation of the SCFs over time follows a polynomial trend, highlighting the increasing impact of corrosion wastage throughout the wind turbine's lifespan. This relationship indicates that as material degradation progresses, the rate at which the SCFs increase accelerates, leading to more pronounced stress concentrations. This trend underscores the significance of accounting for corrosion wastage over the lifespan of the wind turbine, as even small changes in material thickness can significantly influence stress distributions and as a result, the fatigue life.

As a mode of comparison, the analytical Equation described in Section 4.5.7 is used to investigate the effects of corrosion by implementing the corrosion wastage

5.4. M2 (east coast) & M4 (west coast) Fatigue Analysis with Time-varying Stress Concentration Factors

parameters on the dimensions of the flange over a service life of 50 years. The resulting stress distributions are converted to SCFs and are illustrated in Figure 5.10. In this figure, the stress concentration distribution pattern for the first year is represented by the lowermost plot, with each subsequent year corresponding to the next plot above it. These distributions exhibit a pattern highly consistent with those produced in Figure 5.7, further validating the modelling approach and reinforcing confidence in the accuracy of the results.

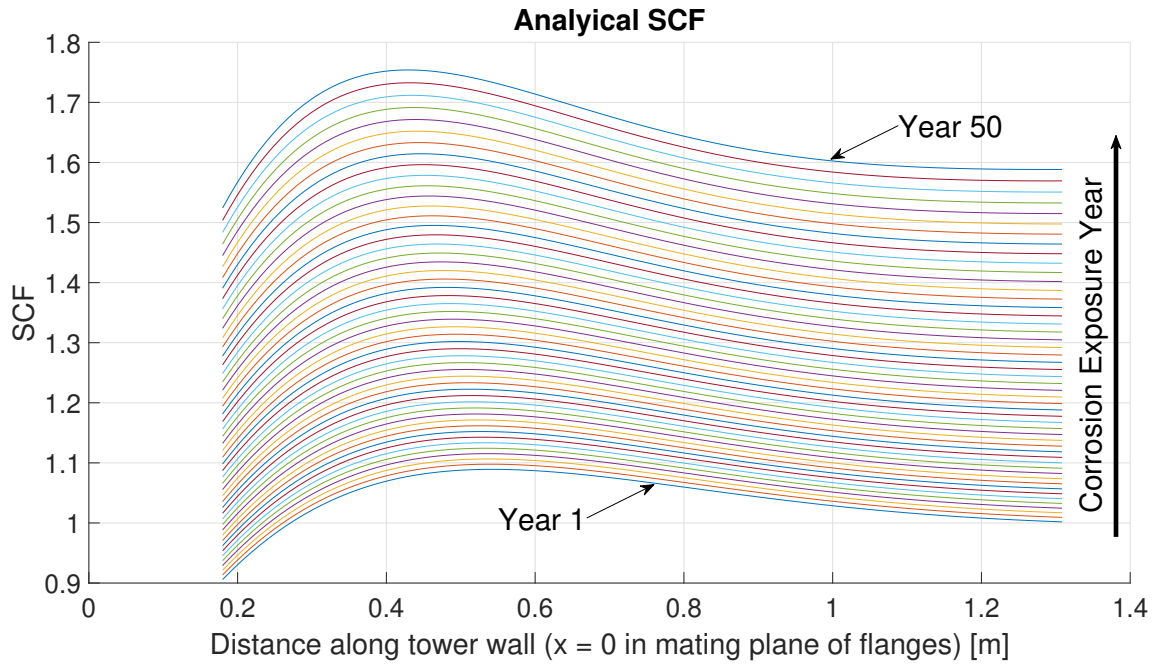


Figure 5.10: Analytical axial force SCF distribution evolution over 50 years of corrosion exposure

5.4 M2 (east coast) & M4 (west coast) Fatigue Analysis with Time-varying Stress Concentration Factors

Following the completion of the 3,300 (1650 x 2) simulations detailed in Section 5.2, a fatigue analysis was conducted in alignment with the methodology outlined in Chapter 4, with three notable modifications to enhance the analysis.

The first modification implemented in this analysis concerns the treatment of stress time history inputs. As in previous analyses, the post-processing of the

composite model results was conducted using a modified version of NREL's software package MLife, as described in Section 3.4.3. However, in this study, the fatigue analysis is performed on an annual basis to account for time-dependent corrosion effects. To achieve this, stress time histories are computed annually using Equation 5.9, allowing each time history to be directly associated with a specific level of corrosion wastage. By employing these stress time histories, the fatigue damage for one year with a given corrosion level can be calculated, providing a granular assessment of the fatigue damage rates.

The second modification incorporates refined environmental conditions to enhance the accuracy of the analysis. In Chapters 3 and 4, the fatigue analysis was conducted using environmental parameters based solely on the wind speed probability distribution, where a single representative met-ocean condition was selected for each wind speed. However, in this section, stemming from the detailed environmental assessment performed in Section 5.2, the environmental distributions of the wind speeds, significant wave heights and peak spectral wave periods are explicitly considered in the fatigue damage calculation. To facilitate this, 25 simulations considering a wide range of met-ocean conditions for each wind speed were performed. These simulations subsequently acted as inputs into the modified MLife software, where the joint probability distributions were used to estimate the frequency of the various environmental conditions over the operational life of the turbine.

The final modification involved the consideration of free corrosion effects on the butt weld connection through the application of S-N curves. Consistent with the approaches in Chapters 3 and 4, the DNV recommended practice standard class D S-N curves [101] were implemented for the butt weld connecting the ring flange and the respective component. This class includes three S-N curves, each corresponding to different environmental exposures: steel exposed to air, steel exposed to seawater with cathodic protection and steel exposed to seawater with free corrosion. These S-N curves are designed with a 97.7% probability of survival, and their properties are detailed in Table 5.7¹. In this chapter, both the S-N curves for steel exposed to seawater with cathodic protection and steel exposed to seawater with free corrosion were applied, depending on the status of the corrosion protection measure at the time in question. As these S-N curves are constructed based on a reference plate with a thickness of 25 mm, a thickness correction factor must be applied as detailed

¹ There is a discrepancy between the tables and charts in DNV GL, the chart values are deemed correct and thus used in this work.

5.4. M2 (east coast) & M4 (west coast) Fatigue Analysis with Time-varying Stress Concentration Factors

in Section 3.4.4.

Table 5.7: S-N curves for steel exposed to seawater with cathodic protection and steel exposed to seawater with free corrosion [101]

Condition	m1	$\log \bar{a}_1$	m2	$\log \bar{a}_2$	Fatigue Limit (MPa)	k
Air	3	12.164	5	15.606	52.63	0.2
Seawater (cathodic protection)	3	11.764	5	15.606	83.41	0.2
Seawater (free corrosion)	3	11.687	-	-	0.2	

As described in Section 3.4.3, the Palmgren-Miner's rule is based on the concept of partial damage accumulation. This principle allows the damage accrued for each year to be summed, providing the total fatigue damage experienced over the operational lifetime of the structure. Using the approach described above, fatigue rates for the wind turbine butt weld were calculated for each year of a 50-year period. This process was carried out under two conditions to account for the impact of corrosion protection effectiveness. The first condition assumes that the corrosion protection is fully effective. In this scenario, no material wastage is considered, and the S-N curve for steel exposed to seawater with cathodic protection is applied to calculate the fatigue rate. This condition represents an ideal case where the corrosion protection measures perform as intended, mitigating the effects of environmental degradation. The second condition considers the case where the corrosion protection measure has failed. In this case, corrosion wastage is applied annually, and the fatigue damage rate is calculated using the S-N curve for steel exposed to seawater with free corrosion.

By calculating the annual fatigue damage rates for these two conditions over 50 years, a detailed assessment of the fatigue damage experience by the wind turbine butt weld can be conducted, considering various durations under which the corrosion protection measures are considered to be effective. For example, the fatigue life of the component under the assumption of 5 years of effective corrosion protection can be determined by summing the fatigue rates for the first five years using the first scenario and subsequently adding the fatigue rates for the second scenario (beginning at year 1) until a cumulative fatigue damage of 1 is achieved.

This approach provides a comprehensive assessment of the structural integrity of the wind turbine under both optimal and degraded corrosion protection conditions, offering valuable insights into the long-term performance and durability of the structure. The framework established here acts as the foundation for the

subsequent fatigue life analysis, which is presented in the following section.

5.4.1 Fatigue Life Evaluation

Building on this established framework, the fatigue life of the butt weld connecting the tower base flange to the tower wall was evaluated. This analysis was carried out using the composite model employing the time-varying SCFs detailed in Section 5.3.2 for the two proposed site locations, M2 (east coast) and M4 (west coast). In order to provide a comparative evaluation, the analysis was additionally performed using SCFs equal to 1 for both the axial and bending stresses, enabling the fatigue life to be calculated in line with the traditional approach implemented in Chapter 3.

To illustrate the impact of the two approaches, Table 5.8 presents the fatigue life estimates for both the traditional and composite model approaches, incorporating the environmental conditions of the M2 site as detailed in Section 5.2. The fatigue lives in this table are calculated considering the DNV S-N curve for steel exposed to seawater with cathodic protection, assuming that the corrosion protective system remains fully functional for the full service life of the turbine. The results highlight the influence of the stress concentrations introduced by the internally bolted ring flange, which gives rise to a substantial reduction in the turbine tower's projected fatigue life. Although the reduction does not lower the life expectancy below the critical 20-year threshold, the expected lifespan is reduced by almost 24 years, representing a 23% decrease in service life.

Table 5.8: M2 (east coast) site: effect of internally bolted ring flange on tower lifespan with corrosion neglected

	Traditional Approach	Composite Model Framework
Lifespan (Years)	104.60	80.65

Table 5.9 presents the fatigue analysis results conducted under the same assumptions for the M4 site. Consistent with the findings for the M2 site, the presence of the internally bolted ring flange leads to a 22% reduction in fatigue life. Again, the projected lifespan remains above the 20-year minimum design requirement, however, it is reduced to 29.15 years, approaching the critical threshold with a much-reduced margin of safety. The results at the M2 and M4 sites echo those in Section 4.6 for

5.4. M2 (east coast) & M4 (west coast) Fatigue Analysis with Time-varying Stress Concentration Factors

the site on the east coast of the U.S, further emphasising the significant influence of the flange on the structural lifespan of the turbine and highlighting the importance of considering such effects in fatigue life assessments.

Table 5.9: M4 (west coast) site: effect of internally bolted ring flange on tower lifespan with corrosion neglected

	Traditional Approach	Composite Model Framework
Lifespan (Years)	37.45	29.15

A notable difference is observed in the expected fatigue lives between the M2 and M4 sites. The 15-MW wind turbine has a considerably shorter lifespan at the M4 site, with an estimated lifespan of 29.15 years compared to 80.65 years at the M2 site. This disparity arises from the contrasting environmental conditions at the two locations, as discussed in Section 5.2.5. Specifically, the M4 site, located on the west coast of Ireland, is exposed to harsher sea conditions from the Atlantic Ocean and stronger wind speeds driven by prevailing south-westerly winds. In contrast, the M2 site, situated on the sheltered east coast in the Irish Sea, experiences milder environmental conditions. The increased wind speeds and harsher met-ocean conditions at the M4 site result in higher fatigue loading, significantly reducing the turbine’s operational lifespan compared to the M2 site.

Considering that corrosion protection measures employed in the splash zones of wind turbines often rapidly degrade and become ineffective [232], a fatigue analysis was conducted considering varying durations of corrosion protection effectiveness, as discussed in Section 5.4. Tables 5.10 and 5.11 present the estimated fatigue lifespan of the wind turbine tower at the M2 and M4 sites considering various durations during which the corrosion protection measures remain effective. These tables evaluate five levels of corrosion protection, corresponding to durations of 0, 5, 10, 15, and 20 years of intact protection. During these periods, no material wastage is assumed, and the S-N curve for steel exposed to seawater with cathodic protection is applied. Given that protection measures in the splash zone typically have a limited lifespan, often not exceeding 5 years [243], the most realistic scenarios are represented by the 0 and 5-year protection durations, while the results for 10 years of corrosion resistance may be considered in cases where regular maintenance and inspection strategies are adapted.

Table 5.10: M2 (east coast) site: tower life span considering the effects of corrosion

Years protection	20	15	10	5	0
Estimated Life Span (Years)	36.42	32.56	28.67	24.75	21.14

Table 5.11: M4 (west coast) site: tower life span considering the effects of corrosion

Years protection	20	15	10	5	0
Estimated Life Span (Years)	23.33	20.07	16.75	13.38	10.11

In table 5.10, which presents the fatigue life estimates for the M2 site, it is demonstrated that the welded connection of the turbine tower surpasses the critical 20-year design life threshold in all the cases considered. However, the benefits of effective corrosion protection measures are clearly evident, with a difference of over 15 years between the unprotected structure and the fully protected structure. This translates to an average extension of 3.82 years in service life for every additional 5 years of corrosion protection, demonstrating how effective corrosion protection can significantly extend the tower’s operational lifespan.

Figure 5.11 illustrates the accumulation of fatigue damage in the butt weld connection over time, comparing the scenarios with various durations of corrosion protection at the M2 site. The figure emphasises the devastating effects of corrosion and underscores the necessity for robust corrosion protection measures. It can be observed that the failure of these protection measures, followed by the onset of corrosion, leads to a sharp increase in the fatigue damage rate, ultimately resulting in rapid failure. This demonstrates the profound influence of corrosion protection on the overall fatigue life of the connection and demonstrates how extended protection measures act to increase the fatigue life by delaying the onset of corrosion.

5.4. M2 (east coast) & M4 (west coast) Fatigue Analysis with Time-varying Stress Concentration Factors

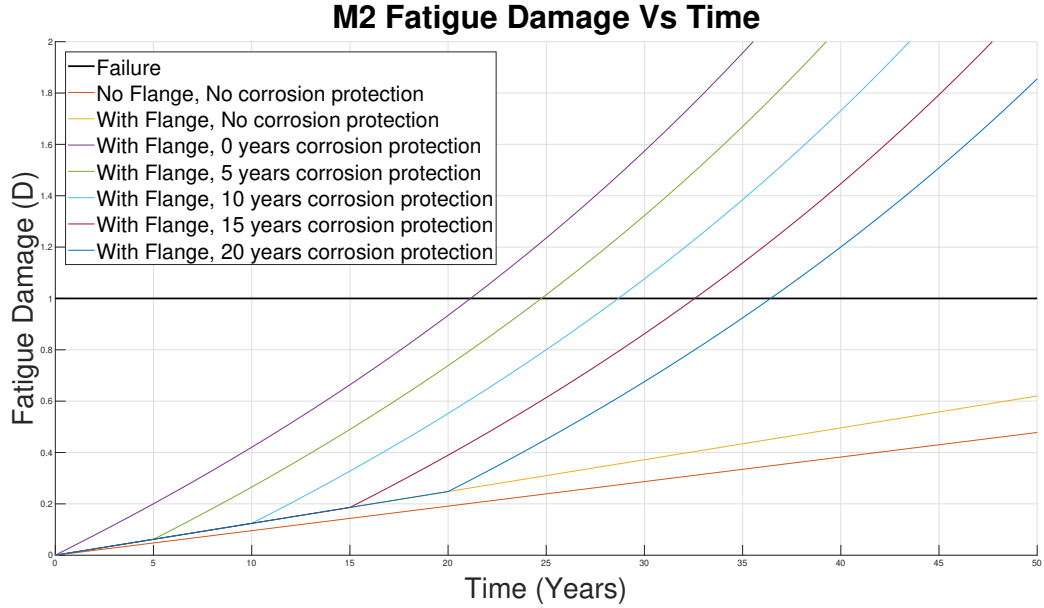


Figure 5.11: Fatigue damage accumulation at M2 site vs. time

The importance of incorporating corrosion wastage is further highlighted by comparing the fatigue life estimates in Table 5.10 with those in Table 5.8, which considers the effect of the flange connection without accounting for corrosion. When the effects of corrosion are neglected, the fatigue life is estimated to be 80.65 years. In contrast, the more realistic estimates incorporating 0 and 5 years of corrosion protection in Table 5.10 show a dramatic decrease to 21.14 and 24.75 years, respectively. These reductions of 74% and 69% underscore the significant impact of considering the effects of corrosion in structural assessments while also highlighting the critical importance of implementing effective corrosion protection measures to ensure long-term reliability.

The critical importance of accounting for corrosion effects is further emphasised by performing a similar comparison for the M4 site. When corrosion effects are neglected, the estimated fatigue life is 29.15 years, as presented in Table 5.9. However, when corrosion effects are incorporated, again considering realistic cases in which the projection measures last 0 and 5 years, the fatigue life is drastically reduced to 10.11 and 13.38 years, respectively. In both cases, the tower's life expectancy falls significantly below the critical 20-year threshold, indicating a structural failure within the component. Such a failure could compromise the integrity of the entire turbine, potentially leading to catastrophic collapse. Further exam-

ination of the data in Table 5.9 reveals that, on average, each additional 5-year period of corrosion protection extends the turbine’s life by approximately 3 years. Consequently, to achieve the minimum 20-year service life, a minimum of 14.9 years of effective corrosion protection would be necessary.

The critical role of corrosion mitigation is further demonstrated in Figure 5.12. Similar to Figure 5.11, the figure presents the accumulation of fatigue damage in the butt weld connection over time under varying durations of corrosion protection at the M4 site. The implementation of corrosion protection measures is shown to extend the fatigue life of the connection, however, the rate of fatigue damage accumulation, both with and without protection, is markedly higher than that observed at the M2 site. This reflects the more severe environmental loading conditions experienced at the M4 location, emphasising the heightened importance of corrosion protection in harsher offshore environments.

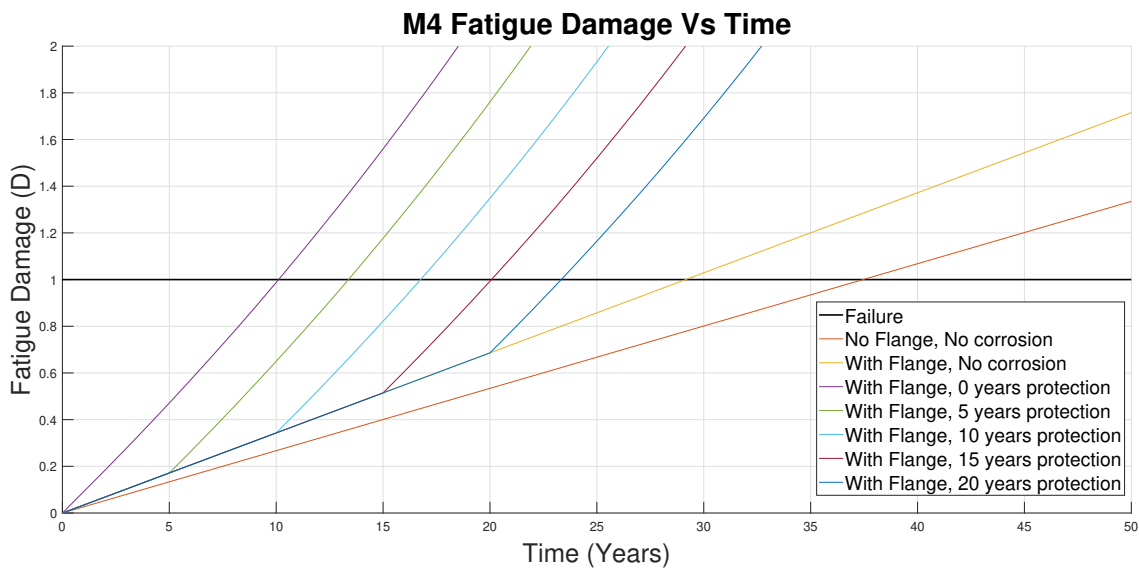


Figure 5.12: Fatigue damage accumulation at M4 site vs. time

5.5 Benchmark Fatigue Analysis with Time-varying Stress Concentration Factors

The fatigue analysis approach discussed in Section 5.4 was additionally applied to environmental conditions provided by NREL for a potential site on the east coast of the United States. In this approach, both the modifications made to incorpo-

5.5. Benchmark Fatigue Analysis with Time-varying Stress Concentration Factors

rate yearly stress time histories and the selection of S-N curves were implemented. However, to enable direct comparisons, the environmental parameters remained consistent with those outlined in Chapters 3 and 4.

For this site, the effect of considering the stress concentration arising from the presence of the ring flange connection was discussed in detail in Section 4.6.1. Compared to the traditional approach, where these additional stresses were not considered, the fatigue life decreased by 19.3%. This reduction is consistent with the $\approx 23\%$ decrease observed for both the M2 and M4 sites, as discussed above, highlighting the significant influence of localised stress concentrations on the fatigue performance of wind turbines.

In a similar manner to Tables 5.10 and 5.11, the expected fatigue lifespan of the butt weld at the tower base connection is summarised in Table 5.12 under different durations of effective corrosion protection. This table again highlights the critical impact of corrosion on lifetime estimation. The fatigue life neglecting such effects is found in Table 4.6.1 to be 24.57 years. However, under free corrosion conditions, where no protective measures are considered, this fatigue life is significantly reduced to 10.5 years. This represents a 57% reduction in fatigue life, leading to a lifespan of just over half of the desired service life. Implementing a typical corrosion protection system that remains effective for the first five years of the service life extends the fatigue life by 2.89 years, reducing the overall fatigue life reduction to 45%. However, even with this protection, the estimated lifespan remains significantly below the desired 20-year design life, indicating a high likelihood of premature weld failure. To prevent such a failure and ensure the cumulative damage remains below unity over a 20-year period, the welded component would require a minimum of 16.29 years of effective corrosion protection. The influence of corrosion protection measures is further illustrated in Figure 5.13, which, similar to Figures 5.11 and 5.12, shows the progression of fatigue damage in the butt weld connection over time for different durations of corrosion protection at the NREL benchmark site. This plot closely aligns with Figure 5.12, again presenting a steep increase in fatigue damage once corrosion protection measures are deemed to have failed. These results further underscore the critical need to account for corrosive effects in fatigue analysis and the importance of regular inspection and maintenance schedules to ensure the continued effectiveness of corrosion protection systems throughout a wind turbine's operational lifespan.

Table 5.12: NREL benchmark site: tower life span considering the effects of corrosion

Years protection	20	15	10	5	0
Estimated Life Span (Years)	22.08	19.27	16.37	13.39	10.50

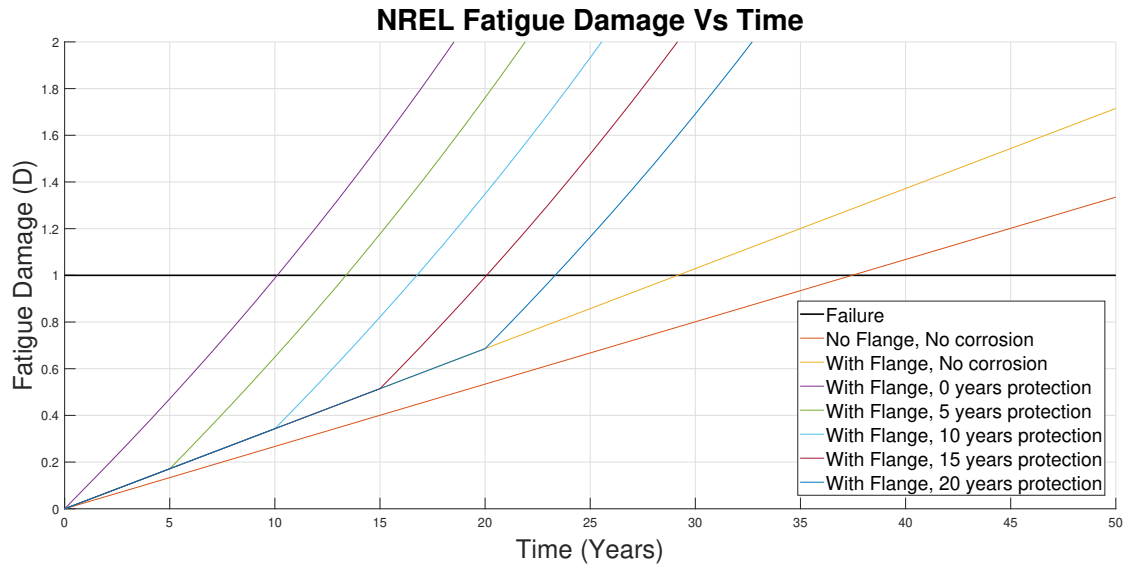


Figure 5.13: Fatigue damage accumulation at NREL benchmark site vs. time

5.6 Conclusion

In this work, the composite modelling framework, integrating an MBD model and a FE model of the IEA 15-MW offshore wind turbine, presented in Chapter 4, is employed to perform a detailed fatigue analysis considering the effects of corrosion. A novelty of this work arises in the ability of the FE model to update the SCFs of the fatigue-prone areas over time as corrosion-induced wastage alters the substructure's geometry, thereby inducing a global redistribution of stresses. Leveraging the parametric capabilities of the FE model within the composite model framework, SCFs are calculated to account for material wastage resulting from the corrosive marine environment in which wind turbines operate. Using these SCFs, the composite model can produce the time history of local stresses occurring in the tower base, considering the effects of material thinning of the components. Based on these stress time histories, which vary on an annual basis, the fatigue damage rate in the weld component of the wind turbine is calculated using S-N curves for steel exposed to seawater with cathodic protection and steel exposed to seawater with free corro-

sion, depending on the status of the corrosion protection measures implemented in the splash zone of the tower.

Using this detailed composite model, fatigue analyses are carried out at NREL's benchmark site on the east coast of the United States and at two sites off the east and west coasts of Ireland, named M2 and M4, respectively. The environmental conditions at NREL's site remained identical to the work in Chapter 4, while the environmental parameters for the M2 and M4 sites are estimated following the IEC 61400 [213] design guidelines and the EU-funded HIPERWIND project recommendations [233] with data obtained from the IMDBON. For both the M2 and M4 sites, the environmental conditions are extended to include the variation of met-ocean conditions at given wind speeds, which are incorporated into the fatigue analysis based on the joint probability distributions of the wind speed, significant wave heights and peak spectral wave periods. The two Irish sites selected have very different environmental characteristics, with the M4 site on the west coast of Ireland having higher wind speeds and larger wave loads due to its location in the Atlantic Ocean. This geographical contrast was reflected in the fatigue analysis results, with the M4 location having a significantly reduced predicted lifespan in comparison with the M2 site.

To validate the use of time-varying SCFs within the composite model framework, this study investigated the influence of corrosion thinning the natural frequency of the wind turbine tower. The modal analysis conducted concluded that material wastage has a negligible effect on the natural frequency of the turbine tower, enabling a single set of dynamic simulations to remain applicable and accurate across the 50-year service life assessed in this study.

Once validated, a fatigue analysis was carried out at each of the sites availing of the SCFs which vary annually. The effects of corrosion were investigated by comparing the lifespan of the wind turbine considering various levels of corrosion protection. Five levels of corrosion protection were considered: No protection, 5, 10, 15 and 20 years of protection. At the M2 site, the turbine tower was deemed safe across all 5 levels, however, at the M4 site, the turbine required at least 14.9 years of effective corrosion protection in order to survive the required 20-year lifespan. A similar result was echoed at the NREL site, with 16.29 years of effective corrosion protection required in order to achieve the desired service life. However, achieving corrosion protection for such durations is unlikely. For the more realistic case in

which the wind turbines' corrosion protection measures fail after 5 years, the life span of the wind turbine is 24.75, 13.38, and 13.39 years for the M2, M4 and NREL sites, respectively. Thus, for the sites at M4 and NREL, strategies aimed at extending the fatigue life must be investigated.

The findings of this study underscore the critical importance of accounting for corrosion effects when evaluating the fatigue life of structures operating in marine environments. In such corrosive conditions, component geometries degrade over time due to material wastage, leading to significant changes in stress distributions. This degradation can be observed in this study through time-varying SCFs which greatly increase the stresses experienced at the weld connection. Neglecting such effects will give rise to an underestimation of fatigue damage, however, it remains a factor currently overlooked in existing design standards [137]. Across all three sites, in the absence of corrosion protection measures, where the turbine tower is subjected to free corrosion, the fatigue lives of the butt weld are found to be less than half of the expected lifespan when corrosion effects are neglected. These findings highlight the severe consequences of neglecting corrosion in structural assessments, demonstrating its substantial influence on durability and operational reliability. To mitigate these detrimental effects, the study emphasises the necessity of implementing robust corrosion protection strategies alongside proactive maintenance programs.

Beyond corrosion effects, this study highlights the importance of conducting site-specific assessments, as identical wind turbines located at different sites off the coast of Ireland exhibit significantly different fatigue lives due to variations in local environmental conditions. By incorporating joint probability distributions of wind speed, significant wave height, and peak spectral wave period at the M2 and M4 sites, the analysis captures the variability in sea states associated with given wind speeds. This approach enables a more realistic representation of the loading conditions experienced by offshore wind turbines over their service life. These insights advocate the requirement for tailored fatigue assessments that reflect site-specific environmental loads, evolving local geometries, and corrosion effects. Neglecting such factors can lead to substantial inaccuracies in fatigue life estimation, potentially compromising the structural integrity and reliability of offshore wind turbine systems.

Chapter 6

Fatigue Life Enhancement via Damping Devices

6.1 Motivation

In Chapter 5, a detailed fatigue analysis was conducted investigating the effects of corrosion on a butt weld connection at the base of the IEA 15MW wind turbine tower. Under typical corrosion conditions, the predicted fatigue lives at the M4 site off the west coast of Ireland and the NREL benchmark site off the east coast of the United States fall significantly short of the standard 20-year design lifespan of modern wind turbines. Nevertheless, as discussed in Section 5.2.5, both sites possess strong energy production potential, making them highly attractive for wind energy development. Given their promising wind resources, there is a strong incentive to ensure the long-term viability of turbines in these locations. To achieve this and to extend the lifespan of wind turbines at these sites beyond the 20-year requirement, highly effective corrosion protection systems would be necessary, capable of maintaining performance for at least 16 years. In the absence of such systems, the fatigue loads on the turbine components must be sufficiently reduced to compensate for the degradation over time.

One potential approach for reducing fatigue loads and enhancing the longevity of wind turbines is the use of auxiliary damping devices. Among these, TMDs have become an increasingly popular solution for mitigating vibration-induced loads [146, 147, 148, 150, 154]. These devices are carefully tuned to the natural frequency of

the primary structure, allowing significant amounts of kinetic energy to be transferred to the tuned mass. The energy is then dissipated through the viscous damper, effectively reducing the vibrational response of the primary structure [244]. While conventional TMDs have demonstrated notable success in reducing structural vibrations, conventional TMDs can exhibit limited damping performance due to their narrow operational frequency range and the requirement for relatively large mass ratios. To overcome these limitations, recent research has proposed the TMDI, a damping device in which an inerter element is incorporated into the traditional TMD configuration. The inerter increases the effective mass of the system without adding significant physical mass, thereby enhancing vibration control across a broader frequency range.

In this chapter, both a TMD and TMDI are integrated into the composite model at the tower top of the IEA 15-MW wind turbine. Their respective effectiveness in mitigating tower vibrations and the resulting bending moments are evaluated with the goal of reducing fatigue damage and ultimately prolonging the operational lifespan of the wind turbine. This investigation is conducted at the benchmark NREL site to explore the potential for extending the wind turbines' lifespan beyond the typical 20-year service life.

6.2 Damping Device Integration with Kane's Dynamics

The damping devices are introduced to the composite model framework through the MBD model. Specifically, the devices are positioned at the top of the turbine tower, as illustrated in Figure 6.1. The inclusion of the damping device introduces an additional DOF, q_D , bringing the total number of DOFs in the MBD model to 23. The kinematics of the damper are defined and integrated into the MBD model, leveraging the flexibility of Kane's formulation to incorporate the additional DOF. By specifying the damper's position, velocity, and acceleration vectors, the generalised active and inertial forces associated with the damping device can be derived and included in the equations of motion. The position vector at the top of the turbine tower can be simply expressed as:

$$r^{OD} = \begin{cases} q_D \hat{\mathbf{b}}_3 & \text{coupled to side-to-side direction} \\ q_D \hat{\mathbf{b}}_1 & \text{coupled to fore-aft direction} \end{cases} \quad (6.1)$$

where D denotes the centre of mass of the damper, q_D is the displacement of the damper from its neutral position and $\hat{\mathbf{b}}$ is the coordinate system at the top of the tower. In the case of the TMD, the resulting forces generated by the spring and viscous damper are incorporated into the generalised active forces, as described in Equation 3.2, while the mass of the damping device is accounted for in the generalised inertia forces, as detailed in Equation 3.3. The resulting system of Equations 3.4 can be solved using the Runge-Kutta 4th order method, as previously described in Section 3.3.3.

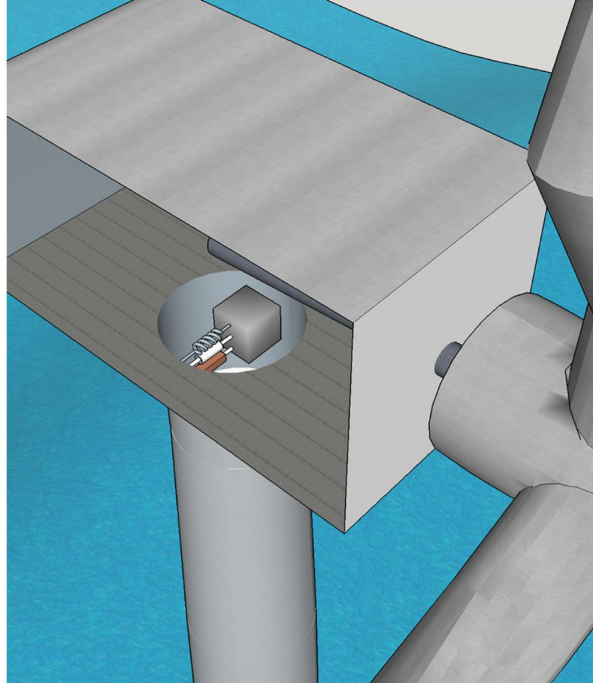


Figure 6.1: Offshore wind turbine with TMDI

6.2.1 Tuned Mass Damper Tuning Parameters

Given the inherent aerodynamic damping provided by the turbine blades in the fore-aft direction [245], the TMD is designed to operate exclusively in the less damped side-to-side direction ($q_D \hat{\mathbf{b}}_1 = 0$), where there is a possibility of a prolonged vibration response. To effectively mitigate lateral vibrations, the TMD is tuned to match

the turbine's first natural side-to-side frequency. There are many analytical and empirical expressions available in the literature to determine the optimal tuning parameters of a TMD such as those formulated by Ghosh and Basu [246], Hoang, Fujino, and Warnitchai [155] and Luft [247]. However, in this study, the stiffness and damping characteristics of the TMD are defined according to the classical approach proposed by Ormondroyd and DenHartog [248].

As the TMD is located at the top of the turbine tower, practical constraints limit the volume of the TMD mass that can be accommodated, preventing the use of excessively large masses. Consequently, a mass equivalent to 1% of the structure's total mass above the mudline is selected [244]. This corresponds to a total mass of 29 tonnes, which is equivalent to 3.7 cubic meters of steel. This mass ratio enables the optimal parameters to be determined using the expressions given in Equation 6.2, where K represents the spring stiffness, ω is the natural circular frequency of the structure, c is the viscous damping coefficient, ζ is the damping ratio and μ_{TMD} is the mass ratio of the TMD.

$$K = \omega^2 m, \quad c = 2\zeta\omega, \quad \zeta = \sqrt{\frac{(3\mu)}{8(1 + \mu)}} \quad (6.2)$$

6.2.2 Model Validation

Following the integration of the TMD into the MBD model, the system was validated through a code-to-code comparison with NREL's state-of-the-art offshore wind turbine simulator, OpenFAST. Figures 6.2-6.5 present a representative sample of the validation results in both time and frequency domains under steady state conditions. The analysis demonstrates strong agreement between the in-house 22-degree-of-freedom MBD model and OpenFAST, confirming the accuracy of the baseline model. This agreement extends to the 23-degree-of-freedom models incorporating the TMD, confirming the accuracy and reliability of the developed MBD model.

Additionally, a modal analysis was conducted using the detailed FE model to determine the influence of the TMD on the natural frequency of the turbine tower. Table 6.1 presents the side-to-side natural frequency of the turbine tower for the FE model, both with and without the TMD. The table demonstrates the negligible effect of the additional mass inducing a frequency shift of 0.56%.

6.2. Damping Device Integration with Kane's Dynamics

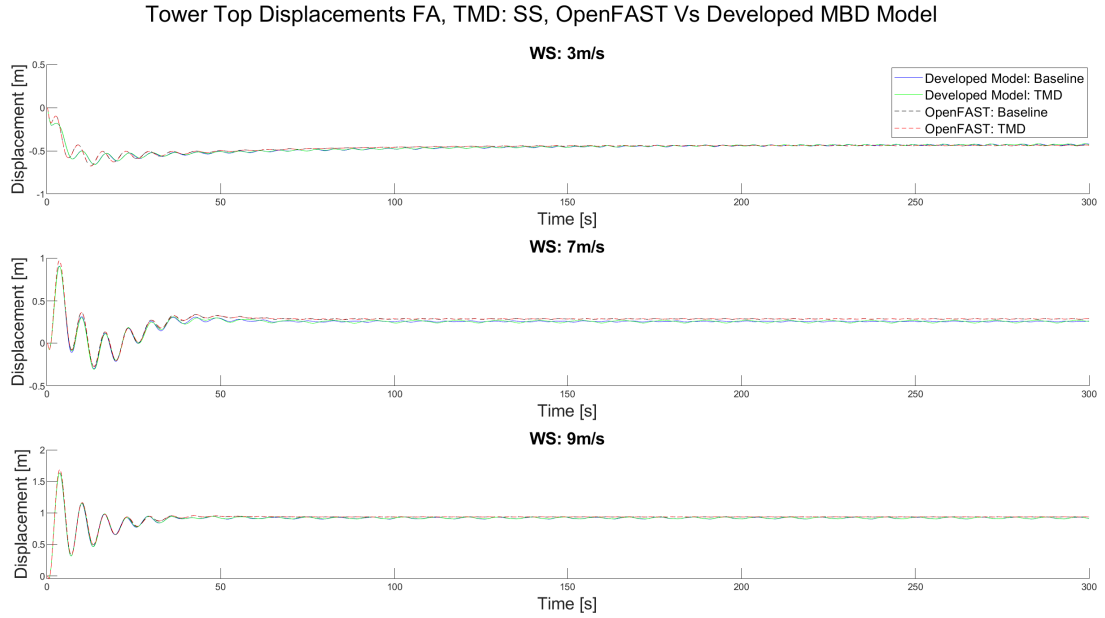


Figure 6.2: MBD model time history validation vs. OpenFAST in fore-aft direction

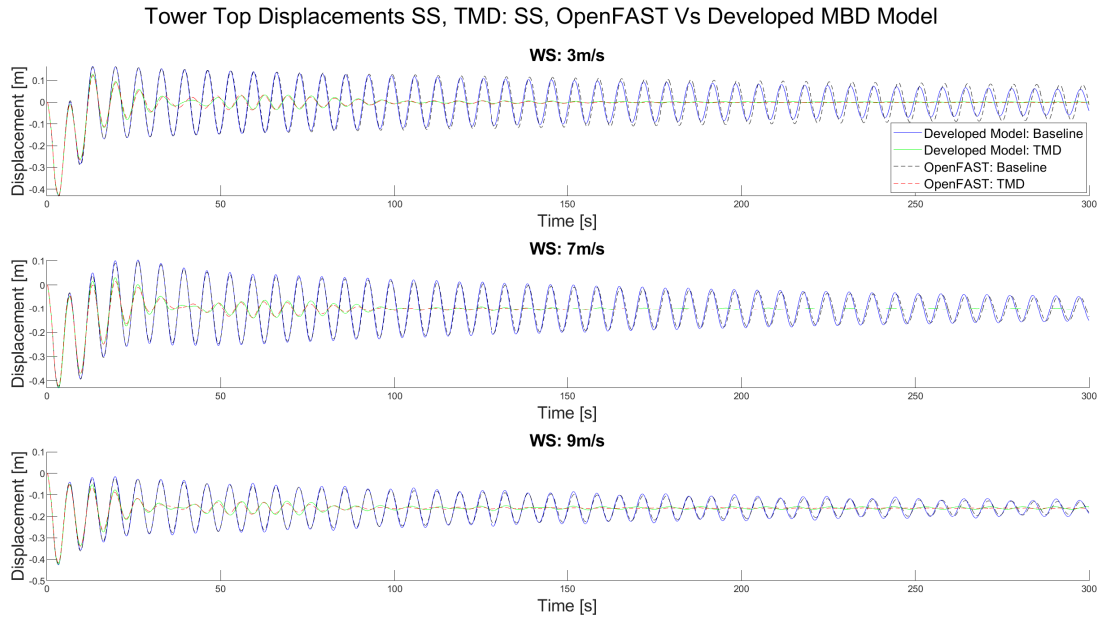


Figure 6.3: MBD model time history validation vs. OpenFAST in side-to-side direction

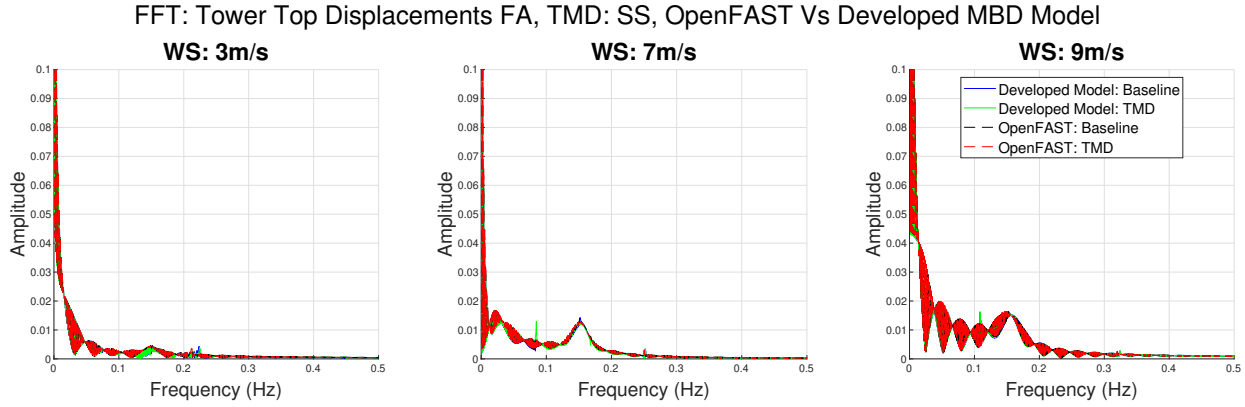


Figure 6.4: Fourier transform of tower top displacements in the fore-aft direction with a TMD vs. OpenFAST

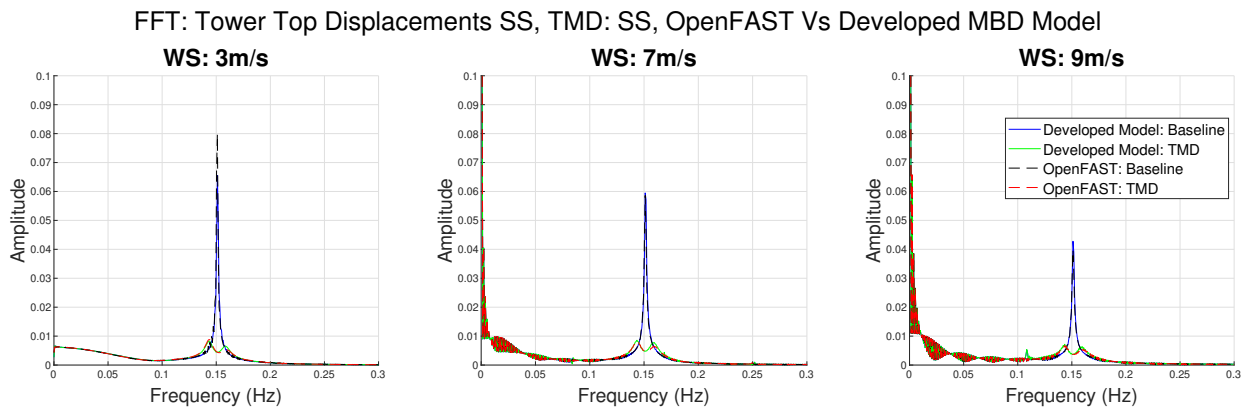


Figure 6.5: Fourier transform of tower top displacements in the side-to-side direction with a TMD vs. OpenFAST

Table 6.1: Effect of TMD on the tower side-to-side natural frequency

	Baseline	TMD
Natural Frequency (Hz)	0.17008	0.16912
Natural Frequency Shift (%)	-	0.56

6.2.3 Fatigue Life Evaluation with a Tuned Mass Damper

Once validated, a fatigue analysis was conducted following the same methodology and environmental conditions as outlined in Chapter 5, with the inclusion of the TMD. This analysis was performed for the benchmark site located on the east coast of the United States. This allowed the results from Chapter 5 to be used as a baseline (uncontrolled) case, enabling a direct evaluation of the damping device's effectiveness.

The addition of the TMD successfully mitigated the vibrational response of the wind turbine, particularly in the side-to-side direction, where significant reductions were observed throughout the simulations. This damping effect was evident across the full range of wind speeds, demonstrating the effectiveness of the control device. Figures 6.6 and 6.7 present sample time histories of the tower top displacements in the fore-aft and side-to-side directions respectively, for wind speeds below, at and above the rated wind speeds. As expected, the TMD's influence is particularly pronounced in the side-to-side direction, where the amplitude of the vibrational response is substantially reduced. The corresponding Fourier spectra, presented in Figures 6.8 and 6.9, demonstrate the impact of the TMD on the frequency content of the tower's response. As shown in Figure 6.9, the amplitude at the tower's natural frequency in the side-to-side direction is significantly reduced, demonstrating the TMD's effectiveness in mitigating resonance effects.



Figure 6.6: Fore-aft tower top displacements with TMD

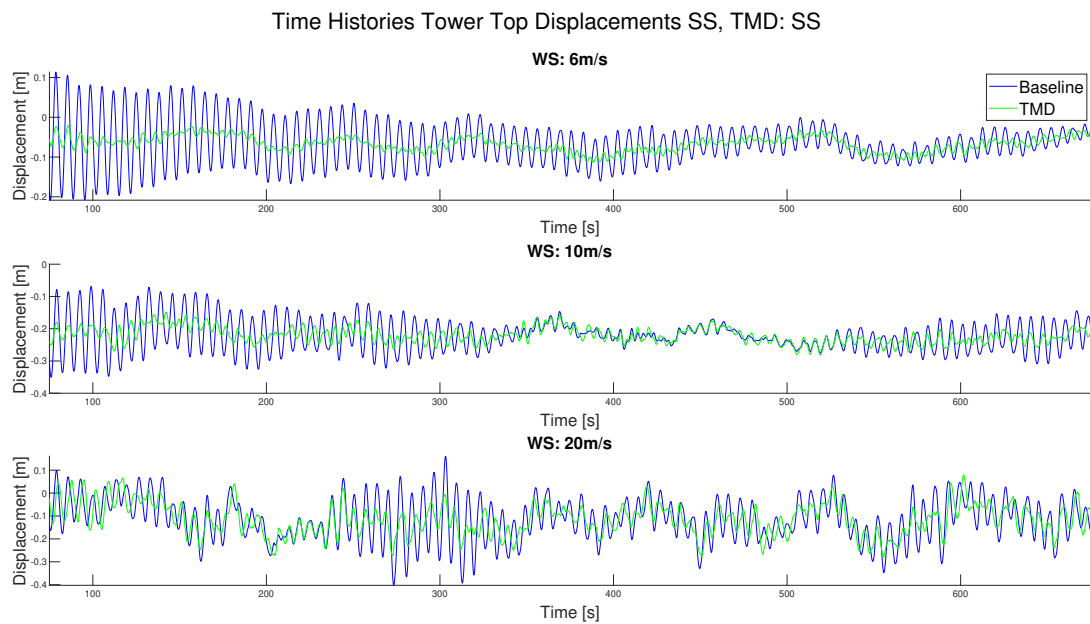


Figure 6.7: Side-to-side tower top displacements with TMD

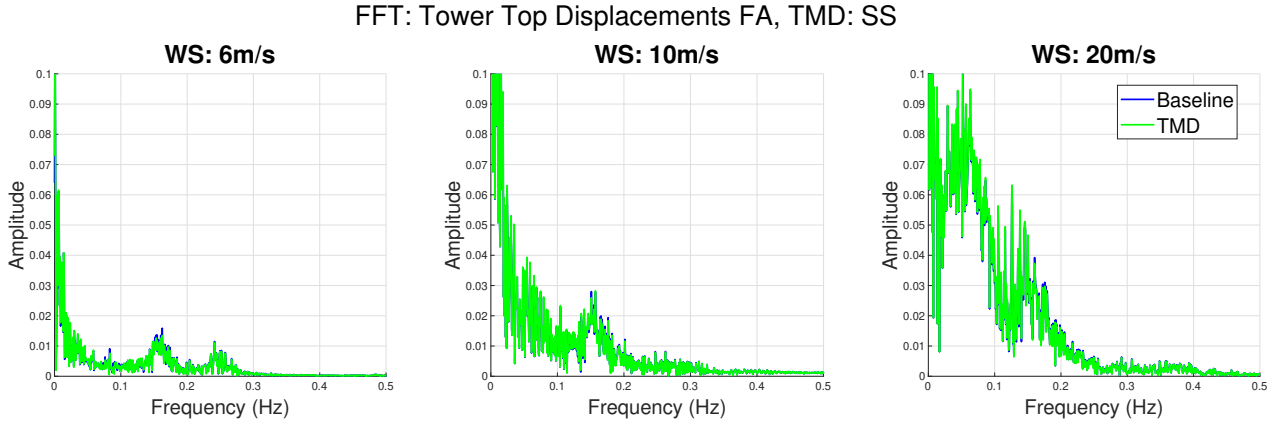


Figure 6.8: Fourier transform of tower top displacements in the fore-aft direction with TMD in the side-to-side direction

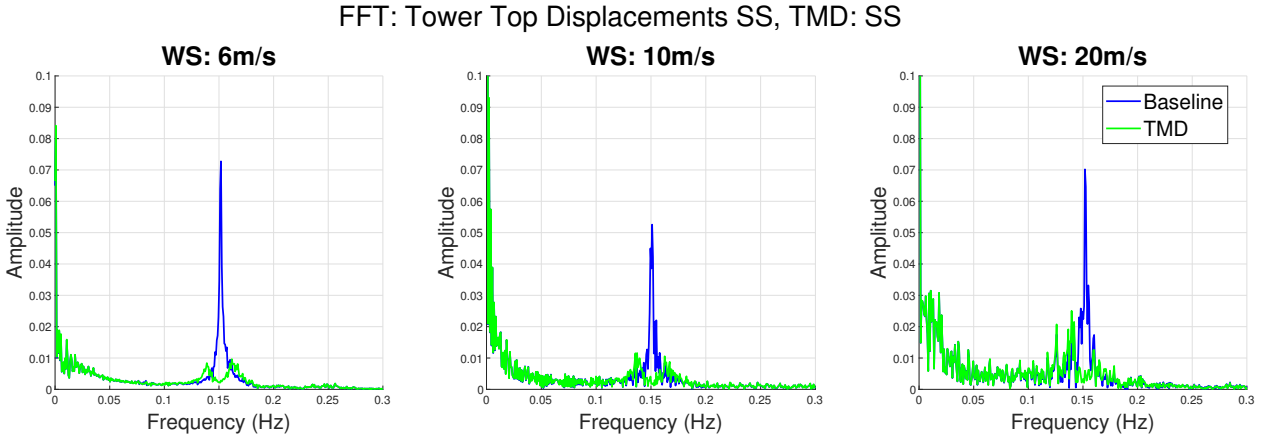


Figure 6.9: Fourier transform of tower top displacements in the side-to-side direction with TMD in the side-to-side direction

The impact of the TMD on the turbine's response was further evaluated by comparing the RMS values of the moments produced at the base of the turbine tower for the baseline condition against the RMS values after integrating the TMD. This comparison provides a quantitative measure of the system's dynamic response over time. The ratio of the RMS values of the tower top displacements for the baseline condition and the TMD-integrated model (with the TMD operating in the side-to-side direction) are presented in Tables 6.2 and 6.3 for the fore-aft and side-to-side directions, respectively. The mitigation capabilities of the TMD in the side-to-side direction are underscored by the consistent reduction of the RMS values across the full range of wind speeds and seeds. However, in the fore-aft direction, many of

the RMS values exhibit a slight increase, as indicated by the highlighted red values. This increase can be attributed to the additional mass of the TMD at the top of the tower, which raises the overall inertia of the system without providing direct damping in the fore-aft direction. To further support the validity of these results, identical simulations were carried out using NREL's openFAST, which confirmed the observed behaviour.

Table 6.2: RMS ratio of fore-aft response: Baseline vs. TMD

Wind Speed [m/s]	Seed 1	Seed 2	Seed 3	Seed 4	Seed 5	Seed 6
4	0.9925	1.0181	1.0042	1.0094	0.9690	0.9840
6	1.0442	1.0196	1.0038	1.0002	0.9874	1.0144
8	1.0211	1.0086	0.9916	1.0069	0.9950	1.0101
10	1.0026	1.0015	1.0107	1.0012	1.0015	1.0154
12	1.0087	1.0016	1.0017	0.9941	0.9966	1.0017
14	1.0052	1.0008	1.0183	1.0055	1.0232	1.0050
16	0.8841	1.2054	1.0021	1.0115	1.0072	1.0022
18	1.0115	1.0858	1.0047	1.0069	1.0041	1.0063
20	1.0044	1.0117	1.0054	1.0065	1.0076	1.0046
22	0.9189	0.9460	1.0106	1.0082	1.0045	1.0043
24	1.0610	1.0569	1.1001	1.0051	1.0045	1.0733

Table 6.3: RMS ratio of side-to-side response: Baseline vs. TMD

Wind Speed [m/s]	Seed 1	Seed 2	Seed 3	Seed 4	Seed 5	Seed 6
4	0.1395	0.1385	0.2542	0.2098	0.2460	0.2252
6	0.6882	0.6289	0.5898	0.6251	0.6791	0.6864
8	0.8906	0.9130	0.8801	0.8437	0.8417	0.8425
10	0.9437	0.9482	0.9339	0.9411	0.9257	0.9618
12	0.9466	0.9556	0.9516	0.9426	0.8339	0.9670
14	0.9313	0.8692	0.9486	0.9666	0.8675	0.9028
16	0.6555	0.7026	0.8810	0.7809	0.8672	0.7777
18	0.9473	0.7077	0.8743	0.9216	0.8055	0.8732
20	0.8212	0.8566	0.7212	0.9061	0.8406	0.8573
22	0.7158	0.7984	0.7847	0.8511	0.9066	0.8788
24	0.7208	0.7299	0.8323	0.7128	0.6789	0.8400

While the magnitude of the increase in fore-aft displacements due to the TMD is relatively small, its impact on the total stresses at the turbine base can be significant. Figures 6.10 and 6.11 illustrate the relative contributions from the fore-aft and side-to-side moments which give rise to the stresses at the tower base. It can be observed that as the wind speed approaches the rated wind speed, the moments resulting from the fore-aft moments become the dominant factor in the induced stresses. Conversely, at lower wind speeds near the cut-in wind speed and at higher wind speeds approaching the cut-out limit, the influence of side-to-side moments increases and the influence from the two components level out. As a result, while the TMD effectively reduces side-to-side displacements, the slight increase in fore-aft moments may offset these benefits under certain wind conditions. In particular, as wind speeds approach the rated wind speed, where fore-aft moments dominate, the additional loading introduced by the TMD can lead to an overall increase in the induced stresses.

This trend of fore-aft dominance in wind turbine loading aligns with the previously identified critical wind speed range of 6 m/s to 14 m/s, as discussed in Section 3.5. The combination of high aerodynamic loading, giving rise to large fatigue damage rates, and the frequent occurrence of the wind speeds within this range means that they contribute significantly to the overall fatigue damage experienced by the turbine. As this critical range coincides with the fore-aft dominance range, any increase in fore-aft moments at these wind speeds can have a pronounced effect on the cumulative fatigue damage. This has critical implications regarding the effectiveness of the TMD, ultimately determining whether the TMD provides a net positive benefit to the structure with respect to fatigue damage. This highlights the importance of considering the effects of damping devices on both the controlled and uncontrolled directions when evaluating their potential for increasing fatigue life.

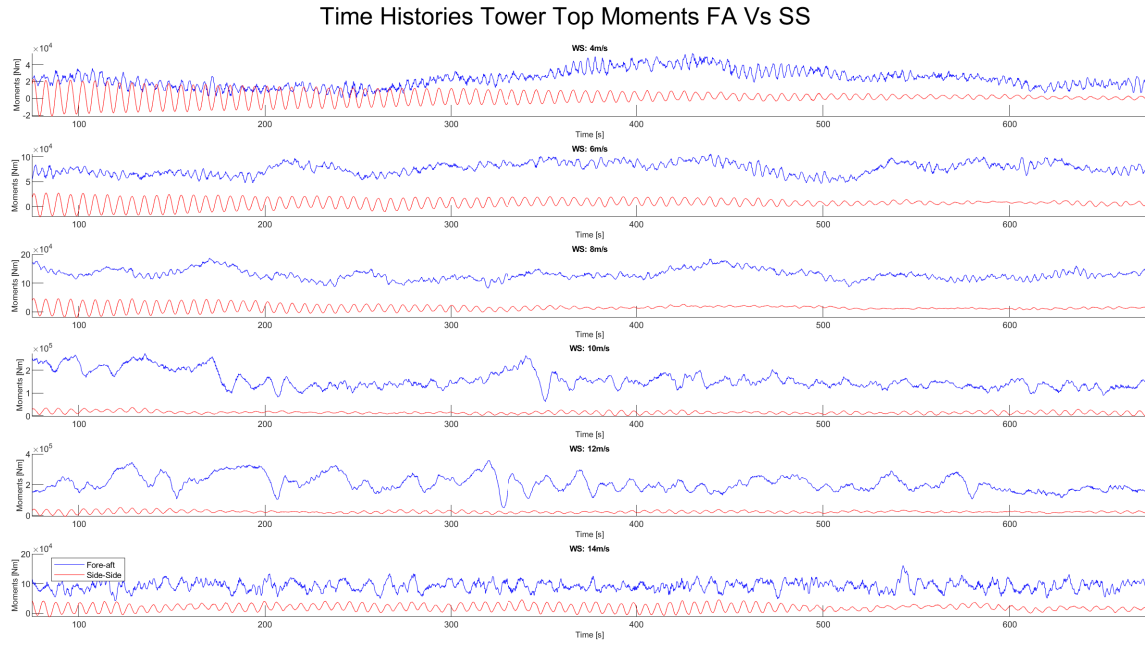


Figure 6.10: Time histories of tower base moments in fore-aft and side-to-side directions (wind speeds 4 m/s - 16 m/s)

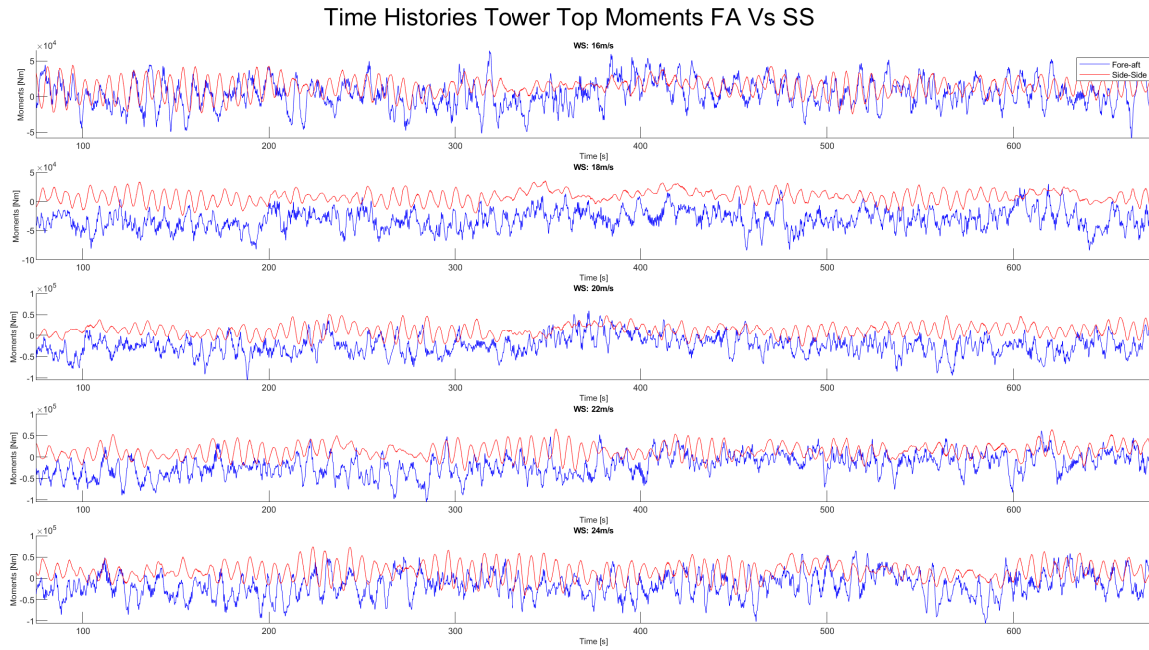


Figure 6.11: Time histories of tower base moments in fore-aft and side-to-side directions (wind speeds 18 m/s - 24 m/s)

As an initial analysis, the fatigue life of the butt weld was examined assuming that the corrosion protective system remains fully functional for the full-service life of the turbine. The computed fatigue damage rates, presented in Table 6.4, indicate that the incorporation of the TMD leads to a reduction in the fatigue damage rate by 10.8%. This reduction contributes to an extension of the fatigue life by just over three years, corresponding to a 12.1% increase, as shown in Table 6.5.

Table 6.4: Fatigue rates Baseline vs. TMD: NREL

	Baseline	TMD
Damage (D)	0.0407	0.0363

Table 6.5: Fatigue life Baseline vs. TMD: NREL

	Baseline	TMD
Fatigue Life (Years)	24.57	27.55

Table 6.6 extends the fatigue analysis, presenting the estimated fatigue lifespan of the wind turbine tower. The analysis considers effective corrosion protection measures for durations of 0, 5, 10, 15, and 20 years, both with and without the addition of a TMD. The inclusion of the TMD results in a modest improvement in fatigue life across all five cases, increasing the predicted lifespan by just over one year. This marginal gain is ultimately insufficient to address the fatigue challenges identified at the end of Chapter 5. The result is particularly underwhelming in the most realistic scenarios, where the TMD fails to extend the fatigue life close to the 20-year requirement. To prevent a premature failure and ensure the cumulative damage remains below unity over the 20-year period, the welded component would require a minimum of 14.21 years of effective corrosion protection, a minor reduction in comparison with the 16.29 years identified in Chapter 5.

Table 6.6: NREL conditions: tower life span with TMD

Years Protection	20	15	10	5	0
Life Span (Years): Baseline	22.08	19.27	16.37	13.39	10.50
Life Span (Years): TMD	23.37	20.49	17.53	14.48	11.55

While the addition of the TMD device resulted in a net gain in fatigue life, its overall effectiveness was limited. This limitation stems from the slight increase in the fore-aft moments, as discussed in Section 6.3.2. In several critical cases, particularly under environmental conditions where wind speeds approached the rated

wind speed of the turbine, this unintended secondary effect of the TMD in the fore-aft direction significantly hindered its potential benefits. To address these limitations, an alternative solution, namely a TMDI, is explored, offering the potential for improved response control and enhanced fatigue life for the wind turbine tower.

6.3 Tuned Mass Damper Inerter

The effectiveness of a TMD in mitigating vibrations is highly dependent on its inertial properties, with larger mass ratios generally enhancing performance [244]. However, for wind turbines, simply increasing the damper's mass to attain improved performance is not a viable solution, as excessive mass at the top of the tower can compromise stability and induce additional vibrations [185]. To address this limitation, researchers have explored alternative damping devices, including the TMDI, a novel variation of the classical TMD. The TMDI works by including an inerter device in parallel to the spring and damper in a classical TMD. The inerter is a mechanical device which amplifies the effect of the mass of the TMD. The presence of the inerter virtually increases the mass of the damper without the addition of the physical mass, leading to greater vibration control capabilities. This enables improved vibration control using a lighter damper, which is particularly beneficial for large flexible structures such as wind turbines [185].

The TMDI is integrated into the MBD model in a similar manner to the TMD discussed in section 6.2, introducing an additional DOF, q_D , to the baseline model. Like the TMD, the TMDI is mounted at the top of the wind turbine tower and is constrained to lateral displacements exclusively. The mass of the TMDI contributes to the generalised inertia forces, while the spring, damper, and inerter contribute to the generalised active forces of the entire system. The force exerted by the inerter element on the tower structure is proportional to the relative acceleration between the structure and the TMDI mass, and is expressed as:

$$F_{\text{TMDI}} = b(\ddot{x}_{\text{TMDI}} - \ddot{x}_t) \quad (6.3)$$

Where: b is the inertance, $\ddot{x}_t$ is the acceleration of the point at which the inerter is connected to the tower and $\ddot{x}_{\text{TMDI}}$ is the acceleration of the TMDI mass.

6.3.1 Tuned Mass Damper Inerter Tuning Parameters

Consistent with the TMD, the TMDI is deployed in the side-to-side direction to target the less-damped vibration mode of the turbine tower. Following the same principles applied to the TMD, the TMDI is restricted to lateral displacements in the side-to-side direction ($\hat{\mathbf{b}}_1 = 0$) and is tuned to match the turbine's first natural side-to-side frequency. The optimal tuning parameters for the TMDI in this work are determined using the closed-form expressions proposed by Sarkar and Fitzgerald [185]. These expressions were derived from a reduced-order model of a fixed-base wind turbine tower with a TMDI installed at the tower top, subjected to white noise excitation. In this simplified model, the mass of the blades, hub, and nacelle were lumped at the top of the turbine tower, as illustrated in Figure 6.12, and no structural damping was applied to the primary structure. These tuning parameters have been demonstrated to be highly effective in dissipating energy associated with the tower's natural frequency, making them highly effective for vibration control [193].

Within these closed form expressions, μ the mass ratio and β the inerter ratio are defined as:

$$\mu = \frac{m_d}{m_0} \quad \beta = \frac{b}{m_0} \quad (6.4)$$

where m_d is the TMDI mass and b is the inertance [181], which has the units of mass [181, 187, 244]. The tuning ratio and damping ratio of the damper are defined as

$$\omega_r = \frac{\omega_d}{\omega_t} \quad \zeta_d = \frac{c_d}{2m_d\omega_d} \quad (6.5)$$

where ω_t and ω_d are the natural frequencies of the tower and the TMDI respectively, and c_d is the damping coefficient of the TMDI. The normalised (normalised to 1) displacement of the tower at a height H_I from its base (illustrated in Figure 6.12) can be obtained directly from the primary mode shape as:

$$\phi = \phi_t(H_I) \quad \text{where } 0 \leq \phi \leq 1 \quad (6.6)$$

where ϕ_t is the primary mode shape of the tower.

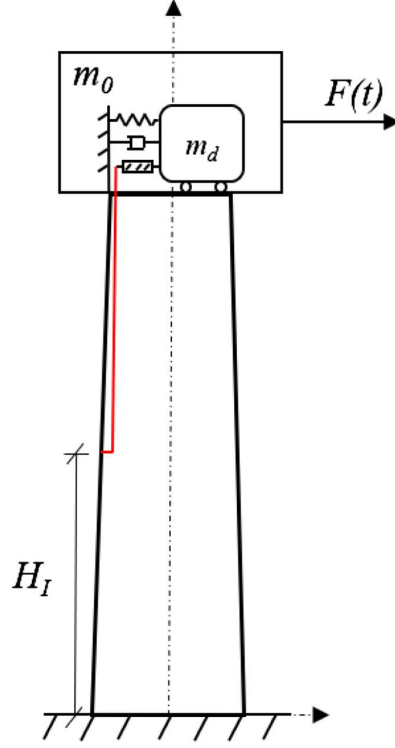


Figure 6.12: Schematic diagram of a tower with a TMDI placed on the top

Based on these definitions, the optimum tuning parameters for this system can be obtained, where the optimal tuning ratio is calculated as:

$$\omega_r = \sqrt{\frac{-B + \sqrt{B^2 - 4AC}}{2A}} \quad (6.7)$$

where

$$\begin{aligned} A &= 3\mu^2(\mu^2 + 2\mu\phi^2\beta - 4\mu\phi\beta + 2\mu\beta + 2\mu + \phi^4\beta^2 - 4\phi^3\beta^2 + 6\phi^2\beta^2 + 2\phi^2\beta - 4\phi\beta^2 \\ &\quad - 4\phi\beta + \beta^2 + 2\beta + 1) \\ B &= \mu(4\mu^2\zeta_d^2 - \mu^2 + 4\mu\phi^2\zeta_d^2\beta - 2\mu\phi^2\beta - 8\mu\phi\zeta_d^2\beta + 2\mu\phi\beta + 4\mu\zeta_d^2\beta + 4\mu\zeta_d^2 - 2\mu\beta \\ &\quad - 2\mu - \phi^2\beta^2 + 2\phi\beta^2 \\ &\quad - \beta^2 - 2\beta) \\ C &= -(\mu^2 + 2\mu\beta + \beta^2) \end{aligned}$$

The optimal damping ratio is given by:

$$\zeta_d = \frac{1}{2\mu\omega_r} \sqrt{\frac{D}{E}} \quad (6.8)$$

where

$$\begin{aligned} D = & \mu^4\omega^4 + 2\mu^3\phi^2\beta\omega_r^4 - 4\mu^3\phi\beta\omega_r^4 + 2\mu^3\beta\omega_r^4 + 2\mu^3\omega_r^4 - \mu^3\omega_r^2 + \mu^2\phi^4\beta^2\omega_r^4 \\ & - 4\mu^2\phi^3\beta^2\omega_r^4 + 6\mu^2\phi^2\beta^2\omega_r^4 + 2\mu^2\phi^2\beta\omega_r^4 - 2\mu^2\phi^2\beta\omega_r^2 - 4\mu^2\phi\beta^2\omega_r^4 \\ & - 4\mu^2\phi\beta\omega_r^4 + 2\mu^2\phi\beta\omega_r^2 + \mu^2\beta^2\omega_r^4 + 2\mu^2\beta\omega_r^4 - 2\mu^2\beta\omega_r^2 + \mu^2\omega_r^4 - 2\mu^2\omega_r^2 + \mu^2 \\ & - \mu\phi^2\beta^2\omega_r^2 + 2\mu\phi\beta^2\omega_r^2 - \mu\beta^2\omega_r^2 - 2\mu\beta\omega_r^2 + 2\mu\beta + \beta^2 \\ E = & \mu + \phi^2\beta - 2\phi\beta + \beta + 1 \end{aligned}$$

In this work, a mass ratio of 1% is selected in line with the TMD and the TMDI is assigned an inertance of $\beta = 0.4$ as it has been shown that the vibration control performance of TMDIs becomes saturated at inerter ratios greater than 0.4 [185]. The inerter device was attached to the turbine tower at $H_I = 0.5$ at a height of 87.29 meters above the mudline in line with Sarkar and Fitzgerald [185]. Using these three parameters, an iterative approach was employed to calculate optimal tuning and damping ratios. Notably, in the above expressions, when $\beta = 0$, the tuning equations reduce to the optimal parameters for a conventional TMD, demonstrating their consistency with well-established tuning approaches.

6.3.2 Fatigue Life Evaluation with a Tuned Mass Damper Inerter

To evaluate the relative effectiveness of the TMDI, the fatigue analysis was conducted following the same methodology and environmental conditions outlined in Chapter 5, enabling a direct comparison with the baseline case and the TMD.

The performance of the implemented controllers is initially assessed in Figures 6.13 and 6.14. These figures present sample time histories of the tower top displacements in the fore-aft and side-to-side directions, comparing the baseline case and the two control devices. These figures illustrate the system's response at wind speeds below, at, and above the rated wind speed. In Figure 6.14, both the TMD and TMDI can be seen to significantly reduce and smooth the vibrational

response, demonstrating their effectiveness in mitigating side-to-side motion.

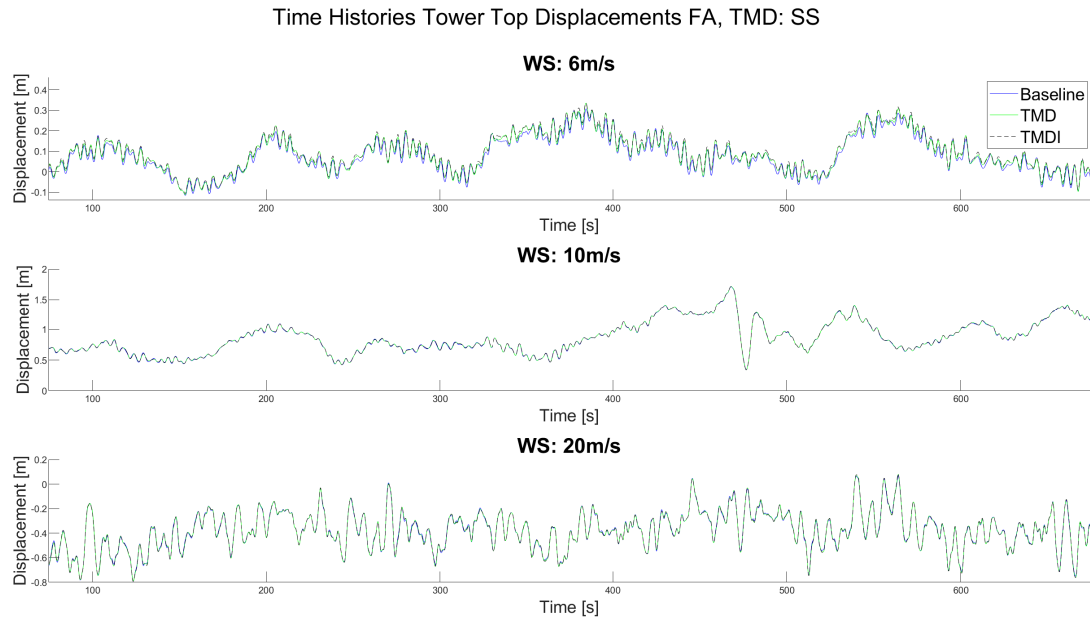


Figure 6.13: Fore-aft tower top displacements with TMDI

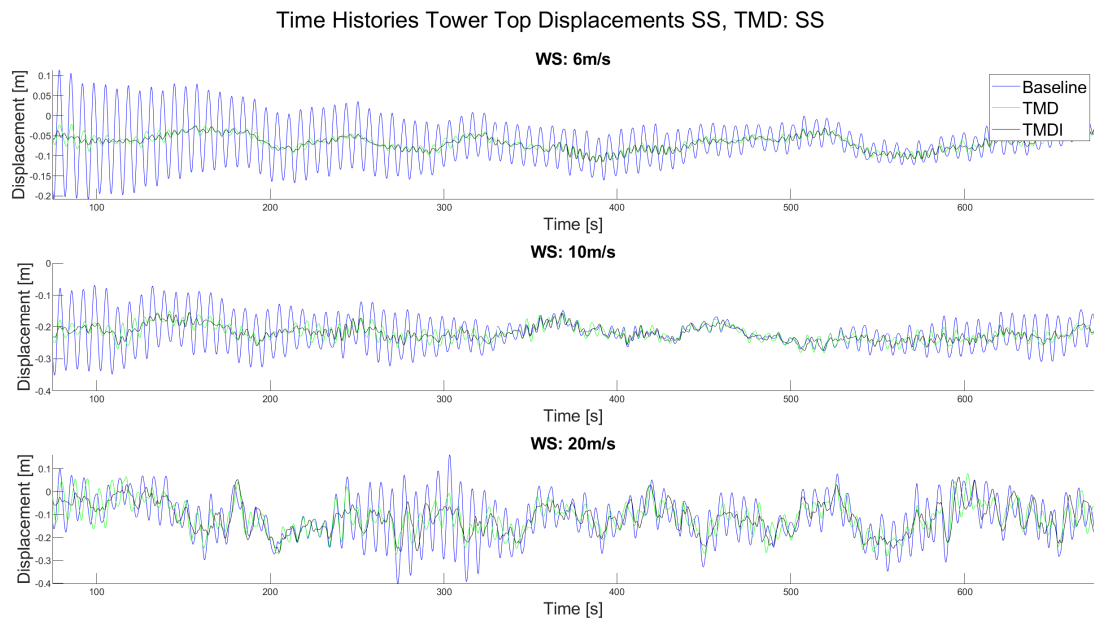


Figure 6.14: Side-to-side tower top displacements with TMDI

The superior mitigation capabilities of the TMDI are further highlighted in Figure 6.15, which provides a detailed comparison of the two control devices in the side-side direction, illustrating the same time histories as Figure 6.14 exclusively for the TMD and TMDI. Owing to its increased effective mass, achieved through the incorporation of an inerter, the TMDI can be observed to consistently outperform the traditional TMD, delivering significantly enhanced vibration mitigation for the same damper mass. This advantage becomes particularly evident under intensified dynamic conditions, where the TMDI exhibits enhanced vibration control.

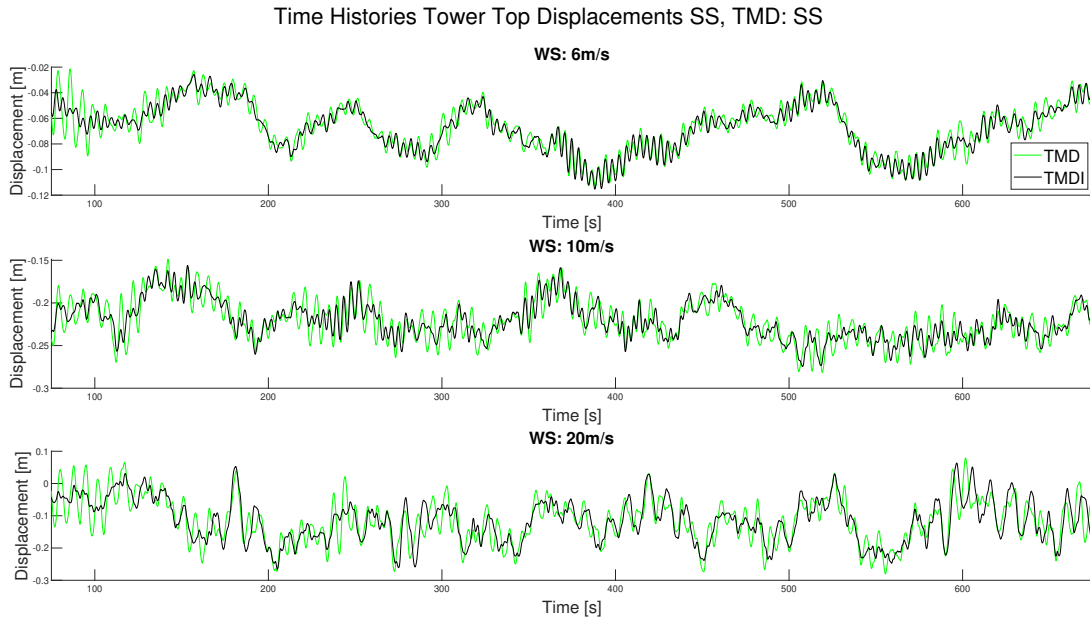


Figure 6.15: Side-to-side tower top displacements with TMDI, TMD vs. TMDI

Figures 6.16 and 6.17 present the Fourier spectra of the tower side-to-side and fore-aft displacements respectively, for the three cases. In Figure 6.16, which depicts the response spectrum for the fore-aft direction, no significant changes are observed across the three cases. In contrast, in Figure 6.17, which represents the side-to-side direction, the sharp peak associated with the natural frequency of the structure is significantly reduced. This reduction is particularly pronounced in the case of the TMDI, which outperforms the traditional TMD due to its amplified effective mass, enabling greater energy dissipation across a broader frequency range. As a result, the TMDI achieves a smoother and more controlled spectrum, making it a more effective solution for vibration control.

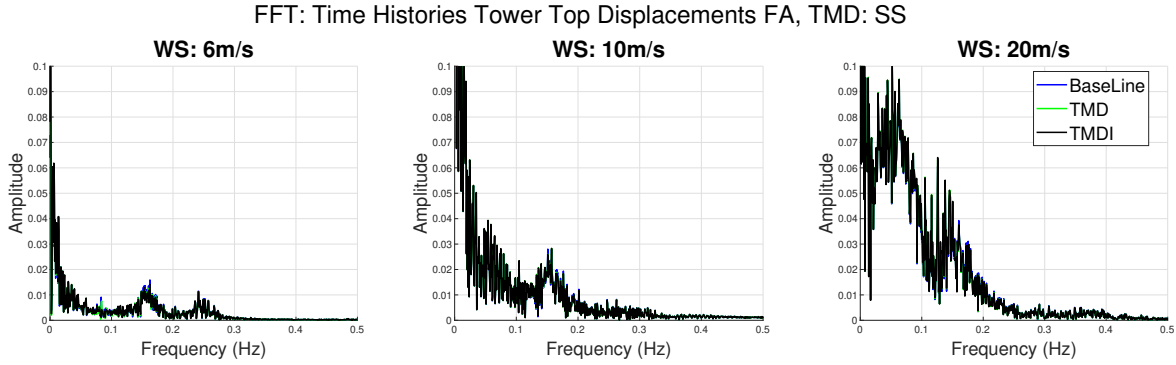


Figure 6.16: Fourier transform of tower top displacements in the fore-aft direction with TMDI in the side-to-side direction

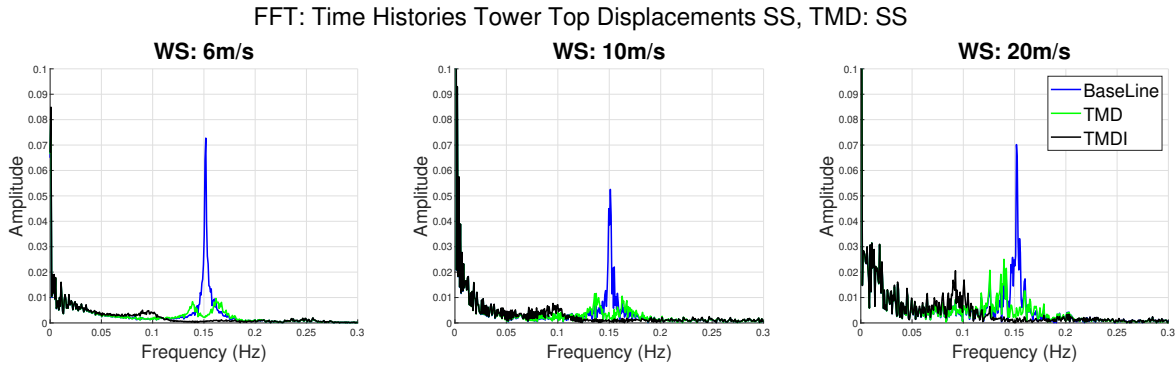


Figure 6.17: Fourier transform of tower top displacements in the side-to-side direction with TMDI in the side-to-side direction

Given the limited space available inside a wind turbine tower, the stroke of the damping device is a critical factor when selecting control devices and optimising their parameters. A key advantage of incorporating an inerter into the damping system is its ability to reduce the required stroke. Figure 6.18 highlights this benefit, showing that the stroke of the TMDI is significantly smaller than that of a conventional TMD. This enhancement becomes especially pronounced under intensified dynamic conditions at higher wind speeds, where the inerter is capable of reducing the stroke by an order of magnitude in comparison with the traditional TMD. This secondary benefit of the TMDI allows for a more compact installation, facilitating easier integration within the restricted internal space of the tower, making it a more practical and viable solution.

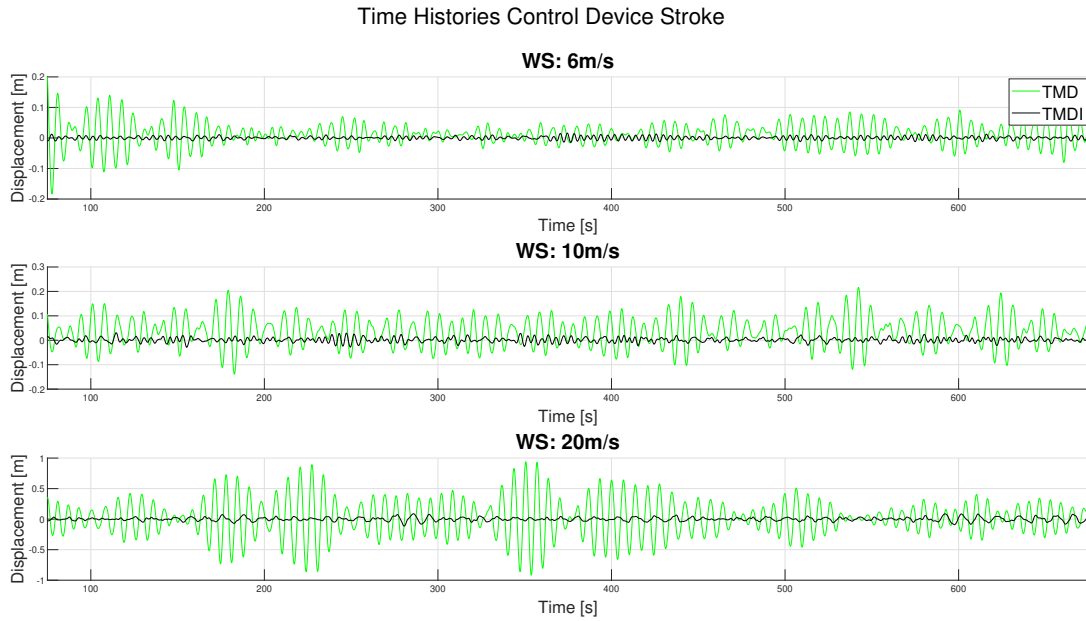


Figure 6.18: Control device stroke in the side-to-side direction, TMD vs. TMDI

A detailed assessment of the TMDI's effectiveness in vibration mitigation is conducted by comparing the RMS values of the moments produced at the base of the turbine tower for the baseline condition with those after the installation of the TMDI. The ratio of these RMS values for both the fore-aft and side-to-side moments are presented in Tables 6.7 and 6.8. The strong vibration mitigation capabilities of the TMDI are evident in Table 6.8, where RMS values show a significant reduction across the full range of wind speeds and seed cases. However, similar to the TMD, a slight increase in fore-aft moments is observed across the majority of the wind speeds and seeds.

Table 6.7: RMS ratio of fore-aft response: Baseline vs. TMDI

Wind Speed [m/s]	Seed 1	Seed 2	Seed 3	Seed 4	Seed 5	Seed 6
4	0.9913	1.0034	0.9862	0.9806	0.9660	0.9475
6	1.0428	1.0312	1.0227	1.0003	0.9790	1.0011
8	0.9923	1.0083	0.9907	1.0084	1.0041	0.9921
10	1.0081	1.0015	1.0107	0.9821	1.0016	1.0265
12	0.9994	1.0082	0.9915	0.9904	0.9944	1.0100
14	1.0103	1.0007	1.0007	1.0302	1.0730	1.0227
16	0.8709	1.0056	1.0045	1.0117	1.0302	1.0170
18	1.0689	0.8886	1.0117	1.0073	1.1047	1.1271
20	1.1364	1.0132	1.0047	1.1299	1.1811	1.0043
22	0.9199	0.9456	1.1975	1.0076	1.0947	1.0054
24	1.0653	1.0508	1.2752	1.0042	0.9829	1.0012

Table 6.8: RMS ratio of side-to-side response: Baseline vs. TMDI

Wind Speed [m/s]	Seed 1	Seed 2	Seed 3	Seed 4	Seed 5	Seed 6
4	0.1152	0.1284	0.2379	0.1928	0.2253	0.2040
6	0.6804	0.6283	0.5955	0.6193	0.6649	0.6753
8	0.8614	0.9111	0.8779	0.8390	0.8495	0.8218
10	0.9402	0.9441	0.9316	0.9232	0.9210	0.9637
12	0.9414	0.9546	0.9475	0.9360	0.8309	0.9652
14	0.9311	0.8642	0.9454	0.9560	0.8571	0.8906
16	0.6093	0.6444	0.8755	0.6494	0.8630	0.7685
18	0.9051	0.6806	0.8586	0.8451	0.7756	0.8387
20	0.7664	0.8413	0.6803	0.8569	0.8081	0.8226
22	0.6836	0.7478	0.7394	0.8181	0.8332	0.8241
24	0.6530	0.6396	0.7490	0.6742	0.6318	0.7489

To compare the two damping devices, the ratio of the RMS values of the TMDI and TMD are presented in Tables 6.9 and 6.10. The superior vibration suppression capability of the TMDI in the side-to-side direction is confirmed in Table 6.10, where the TMDI consistently provides reductions in the side-to-side bending moments. However, in Table 6.9, the TMDI is shown to induce increased fore-aft moments compared to the TMD across many of the cases. This increase stems from the enhanced inertial effects introduced by the inerter, which amplifies the apparent mass of the damping device. Due to the inherent coupling between the fore-aft and side-to-side modes, the increased effective inertia in the side-to-side direction also influences the fore-aft direction. As a result, the fore-aft moments

are elevated compared to those of the TMD, even when the same mass ratio is maintained.

Table 6.9: RMS ratio of fore-aft response: TMD vs. TMDI

Wind Speed [m/s]	Seed 1	Seed 2	Seed 3	Seed 4	Seed 5	Seed 6
4	0.9988	0.9855	0.9821	0.9714	0.9969	0.9629
6	0.9986	1.0113	1.0188	1.0001	0.9916	0.9868
8	0.9718	0.9997	0.9991	1.0016	1.0092	0.9822
10	1.0054	1.0000	1.0000	0.9809	1.0001	1.0109
12	0.9908	1.0067	0.9898	0.9963	0.9978	1.0084
14	1.0051	0.9999	0.9826	1.0246	1.0487	1.0176
16	0.9851	0.8343	1.0024	1.0002	1.0229	1.0147
18	1.0567	0.8184	1.0070	1.0004	1.1002	1.1200
20	1.1315	1.0015	0.9993	1.1226	1.1723	0.9998
22	1.0012	0.9995	1.1850	0.9994	1.0898	1.0011
24	1.0041	0.9943	1.1592	0.9992	0.9784	0.9328

Table 6.10: RMS ratio of side-to-side response: TMD vs. TMDI

Wind Speed [m/s]	Seed 1	Seed 2	Seed 3	Seed 4	Seed 5	Seed 6
4	0.8258	0.9274	0.9359	0.9190	0.9157	0.9057
6	0.9886	0.9990	1.0097	0.9907	0.9791	0.9838
8	0.9673	0.9979	0.9975	0.9945	1.0093	0.9755
10	0.9963	0.9957	0.9975	0.9810	0.9949	1.0020
12	0.9945	0.9989	0.9956	0.9930	0.9964	0.9981
14	0.9998	0.9942	0.9967	0.9890	0.9880	0.9864
16	0.9294	0.9172	0.9938	0.8315	0.9952	0.9882
18	0.9555	0.9617	0.9820	0.9170	0.9629	0.9605
20	0.9333	0.9821	0.9433	0.9457	0.9613	0.9595
22	0.9550	0.9366	0.9423	0.9612	0.9191	0.9377
24	0.9059	0.8764	0.8999	0.9459	0.9305	0.8915

While the superior mitigation of side-to-side vibrations is evident, the unintended amplification of fore-aft moments raises concerns regarding the net benefit of the inerter within the damping device. This concern is particularly relevant given that fatigue damage is highly dependent on the critical wind speed range of 6 m/s to 14 m/s, where fore-aft moments dominate the induced stresses. Understanding whether this increase in fore-aft loading leads to accelerated fatigue degradation is essential for assessing the overall effectiveness of the TMDI.

To evaluate this impact further, the fatigue damage rates at the butt weld connecting the tower base flange to the tower wall, considering the additional damping provided by both the TMD and TMDI, are presented in Table 6.11. It can be seen that while the TMDI decreases the damage rate in comparison with the baseline case, the introduction of the inerter acts to increase the damage rate experienced by the weld. Despite the additional damping that is observed in the side-to-side direction provided by the TMDI, the additional moments induced in the fore-aft direction act to increase the fatigue damage rate by 2.5% in comparison with the traditional TMD.

Table 6.11: Fatigue damage rate with TMDI

	Baseline	TMD	TMDI
Damage (D)	0.0407	0.0363	0.0372

Using this damage rate and the assumption that the corrosion protective system remains fully functional for the full-service life of the turbine, the fatigue life is calculated in Table 6.12. The increased damage rate arising from the inerter results in a fatigue life reduction of just over 8 months. While this is a small reduction, the inerter was introduced with the primary goal of increasing the fatigue life via its enhanced response mitigation. This finding emphasises the need to carefully weigh the TMDI's benefits in side-to-side damping against the potential drawbacks of increased fore-aft moments, particularly in scenarios where fore-aft loading dominates fatigue damage.

Table 6.12: Tower fatigue life with TMDI

	Baseline	TMD	TMDI
Lifespan (Years)	24.57	27.55	26.88

Figure 6.19 presents the damage rates at individual wind speeds for the uncontrolled and controlled cases and further highlights the critical wind speeds identified in Section 3.5. At wind speeds of 8, 10 and 12 m/s, corresponding to the peak of the damage rates, the TMD can be seen to outperform the TMDI. In this wind speed range, the fore-aft moments have a significant influence on the resulting fatigue rate. As a result of the increased fore-aft moments caused by the amplified mass, the effectiveness of the damping device is decreased in this region of critical importance. This wind speed range, due to its frequent occurrence and high loading,

dominates the overall fatigue life, ultimately leading to the net negative impact of the TMDI on fatigue life.

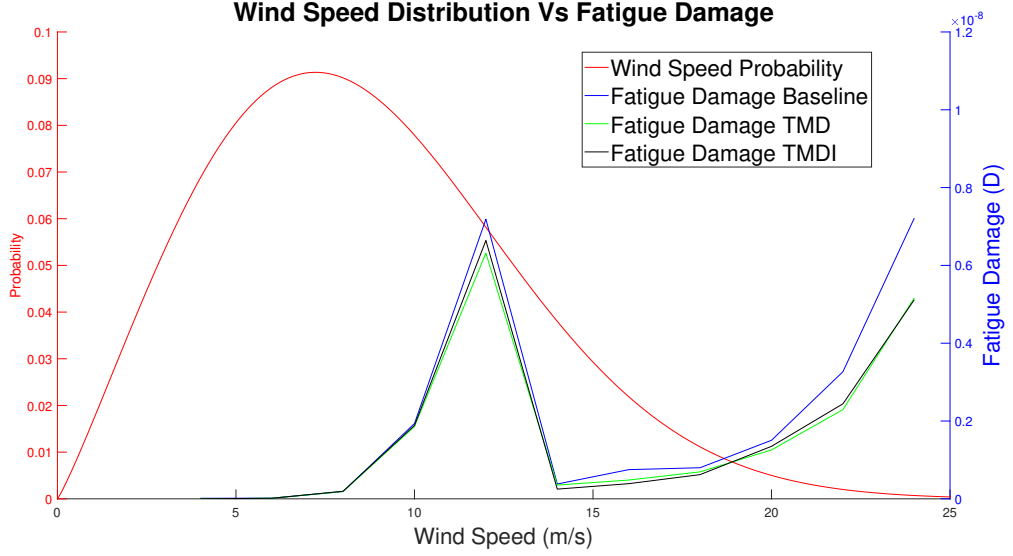


Figure 6.19: Fatigue damage rate vs. wind speed with control devices

The impact of this negative influence considering the effects of corrosion is investigated in Table 6.13, which presents the total fatigue life given the various levels of corrosion protection considered. The negative influence of the TMDI in comparison with the TMD is observed to be insignificant across the full range of corrosion protection measures. While the inerter does not provide the desired increase in fatigue life, the TMDI may still be beneficial for overall turbine operation. Due to its enhanced vibration mitigation capabilities, the TMDI will provide additional damage protection to acceleration-sensitive equipment typically housed in the nacelle while having a minimal influence on the fatigue life of the turbine.

Table 6.13: Tower life span with TMDI and corrosion effects

Years Protection	20	15	10	5	0
Life Span (Years): Baseline	22.08	19.27	16.37	13.39	10.50
Life Span (Years): TMD	23.37	20.49	17.53	14.48	11.55
Life Span (Years): TMDI	23.15	20.33	17.41	14.41	11.52

Although the TMDI effectively mitigates excessive vibrations that could impact other components, it fails to achieve its primary objective in this study by

accelerating fatigue damage to the tower compared to the TMD. This highlights the need to consider the impact of the TMDI on the long-term structural performance, as its vibration reduction capability comes at the cost of increased fatigue damage.

6.4 Conclusions

This study employs a composite model to conduct a local fatigue analysis of a fatigue-critical component at the base of the state-of-the-art IEA 15-MW offshore wind turbine tower. This fatigue analysis is conducted specifically to investigate the influence of damping devices on the fatigue life of offshore wind turbines. The composite model utilises a blended modelling approach integrating a holistic MBD model and a detailed FE model, leveraging the strengths of both approaches to deliver an efficient and accurate fatigue analysis of the turbine, mitigating the inherent weaknesses of each individual model.

The MBD model developed using Kane's Dynamics provides a flexible framework for incorporating damping devices. This approach allows for seamless integration of the damping devices within the composite model. The developed MBD model, fully coupled with 23 DOFs, is validated using a code-to-code comparison with NREL's state-of-the-art wind turbine simulator OpenFAST. This MBD model accounting for the dynamics associated with the damping devices is employed within this composite model to simulate nominal stress time histories of the wind turbine subjected to environmental loading, considering the effects of both hydrodynamic and aerodynamic loading. SCFs arising at the butt weld of the ring flange connection at the turbine tower base are coupled with these nominal stresses to generate local stress time histories. These SCFs are calculated using a detailed FE model of the wind turbine, considering the effects of local geometries and corrosion wastage. Using these results, a comprehensive fatigue assessment is performed of the 15-MW monopile-based wind turbine, taking into account met-ocean conditions specific to NREL's benchmark offshore wind turbine site off the east coast of the United States.

Control devices were introduced to mitigate the vibration response of the wind turbine in the side-to-side direction and were tuned to the natural frequency of the turbine tower. Both of the controllers displayed large reductions in the response of the wind turbine tower across the range of environmental conditions examined,

demonstrating the effectiveness of the control devices. The TMDI outperformed the traditional TMD across the full range of wind speeds and loading conditions, producing significantly increased vibration mitigation for the same damper mass. Additionally, the introduction of the inerter greatly reduced the stroke of the mass of the damping device, providing the TMDI with an extremely beneficial and practical advantage over classical TMDs, given the limited space available inside wind turbine towers.

While the damping devices successfully mitigate vibrations in their intended direction, their influence on other directions must be carefully evaluated to ensure balanced performance across all critical modes of vibration. While both the TMD and TMDI successfully mitigate vibrations in their intended direction, it was demonstrated that the additional mass and coupling of the side-to-side and fore-aft responses induce an increase in the fore-aft response. This negative impact on the response of the turbine in the fore-aft direction, which dominates the induced stresses, constrained the benefits of the damping devices regarding the lifespan of the wind turbine. This negative influence was particularly pronounced in cases where the TMDI was implemented due to the mass amplification effect induced by the inerter. Despite this negative influence, the damping devices did provide some benefits in terms of fatigue life extension. When considering the effects of corrosion, the introduction of the TMD provided one additional year of service life to the weld, while the introduction of the inerter device reduced this improvement marginally. However, these limited improvements were insufficient to offset the cumulative fatigue damage experienced by the wind turbine over a 20-year lifespan.

In summary, the TMDI demonstrates great potential for vibration mitigation in the side-to-side direction for offshore wind turbine towers. This novel damper significantly outperforms the traditional TMD for the same damper mass, effectively reducing the tower's vibration response in its designed direction. However, as a result of the additional mass of the damping devices and the inherent coupling, the introduction of the damping devices induces a marginal increase in the fore-aft response which hinders the fatigue life improvements which can be achieved via the damping devices. The mass amplification effect, which enhances the TMDI's damping capability in the side-to-side direction, simultaneously acts to increase this negative influence in the fore-aft direction, causing the TMDI to underperform compared to the TMD in terms of fatigue reduction. Therefore, despite their strong

ability to mitigate undesired vibrations, neither the TMD nor the TMDI prove to be particularly effective in reducing fatigue damage in offshore wind turbine towers.

Chapter 7

Conclusions

7.1 Summary and Conclusions

In an endeavour to begin competing with conventional fossil fuels, wind turbine technology has made significant strides in enhancing efficiency and cost-effectiveness. In pursuit of this, wind turbines have increased dramatically in size, and in more recent times, are being deployed in offshore locations. The increased scale and offshore deployment of modern turbines has unlocked the potential to harness the abundant wind resources of open seas, allowing them to capture higher wind speeds with improved airflow quality. However, these advancements come with significant challenges. Modern offshore wind turbines are subject to harsh environmental loading, exposed to highly fluctuating wind speeds, volatile sea states, and a corrosive marine environment. The operation of these large-scale turbines under such conditions generates substantial dynamic responses, with the rotating mass of the turbine blades-within an atmospheric boundary layer influenced by wind shear-imposing additional cyclic, gravitational loads. As a result of these amplified loading conditions, modern wind turbines and their constituent components are particularly vulnerable to fatigue damage. Accurately quantifying this fatigue damage is a vital aspect of offshore wind turbine design and analysis. It enables the optimisation of maintenance schedules, end-of-life strategies, and the informed selection of component dimensions and materials. This, in turn, facilitates the development of wind energy systems that achieve long-term structural integrity, promoting increased sustainability and economic viability.

However, the accurate estimation of fatigue damage is inherently challenging due to the complex and highly variable nature of the environmental loading conditions, the nonlinear dynamic behaviour of wind turbine systems, and the intricate geometries of fatigue-critical components such as welds, bolts, and flanges. Traditional MBD modelling approaches can effectively capture the global dynamic response of wind turbines under operational conditions, considering the effects of both hydrodynamic and aerodynamic loading. However, these models often rely on simplified geometric representations that fail to capture localised structural effects or account for time-dependent material degradation. Conversely, high-fidelity FE models can resolve local stress concentrations with greater accuracy, however, their application is constrained by the substantial computational cost of dynamic simulations—particularly in fatigue analysis, where a large number of simulations are required to capture the stochastic nature of environmental conditions.

Recognising the essential role of fatigue analysis and the need for accurate and efficient evaluation, this research developed a computationally efficient approach for enhanced fatigue damage quantification. To achieve this, a composite modelling framework was proposed in Chapter 4. This framework integrates an MBD model with a detailed FE model, blending the MBD model’s ability to capture the global response of the wind turbine under both aerodynamic and hydrodynamic loading with the FE model’s capacity to characterise local stress distributions caused by geometric features. The stress time histories generated by the MBD model are refined through the incorporation of SCFs from the FE model, enabling accurate consideration of local geometric effects that are otherwise neglected in the MBD approach. This framework was employed to investigate the life span of a critical butt weld connection at the base of the wind turbine tower, selected due to its documented history as the origin point for fatigue-induced failure of several costly turbine collapses [43, 44]. This developed framework was employed in Chapter 4 to assess the fatigue life of the butt weld, considering the effects of a specifically designed internally bolted ring flange connection. This flange connection acts to connect the base of the wind turbine tower to the transition piece of the IEA 15-MW monopile-based offshore wind turbine. The fatigue damage at the weld location was evaluated under real metocean conditions representative of a site off the east coast of the United States. These conditions are provided by NREL as a standardised set of environmental parameters intended to serve as a benchmark for comparison within the research community.

The vital importance of obtaining accurate stress time histories, which consider the geometric effects arising from such components, was highlighted in Chapter 4 by comparing the lifespan of the butt weld both with and without such effects. In Chapter 3, the fatigue life was calculated to be 31.45 years, neglecting the influence of the proposed ring flange connection and corrosion. However, in Chapter 4, considering the SCF arising due to the stiffening effect of the flange, the fatigue life was reduced by 19.3%, yielding a more realistic estimate of 24.57 years. These findings reinforce the necessity of accounting for detailed structural features in fatigue assessments, as neglecting these factors can result in conservative fatigue damage estimations, potentially resulting in unexpected failures, heightened maintenance demands, and diminished economic performance of offshore wind assets. In addition to quantifying the impact of geometric effects, the fatigue analyses also identified a critical wind speed range between 6 and 14 m/s. Owing to the combination of the high fatigue damage rates and the frequent occurrence of these conditions, these conditions were found to have a significant influence on the overall fatigue life of the IEA-15MW wind turbine.

Building on the importance of considering local geometric effects, Chapter 5 extends the fatigue analysis to consider the effects of the corrosive marine environment. These effects were accounted for within the composite model framework through the use of time-varying SCFs, which reflect the effects of material degradation over time. By considering these effects, the fatigue life of the butt weld was reduced to just 13.39 years, falling significantly short of the expected design life of 20 years that is typical in modern offshore wind turbines. This critically low fatigue life underscores the importance of accounting for both local geometric effects and corrosion, highlighting the need for robust corrosion protection strategies combined with proactive maintenance programs.

In addition to NREL's benchmark site, the fatigue life of the weld was assessed under environmental loading conditions for two offshore locations off the east and west coasts of Ireland. These sites exhibit inherently different environmental characteristics due to their geographical positions, one located in the Atlantic Ocean and the other in the Irish Sea. Environmental parameters for both locations were derived from data provided by the Irish Marine Data Buoy Observation Network, and the resulting fatigue life predictions were compared to assess how site-specific metocean conditions influence the structural performance. A large variation in the

fatigue life was noted at the two sites, with the M2 site located on the east coast of Ireland having a fatigue life of almost double that of the M4 site located on the west coast. This difference is largely attributed to the M4 site experiencing higher wind speeds and greater significant wave heights, contributing to heightened load demands. These findings highlight the importance of conducting site-specific assessments, as identical wind turbines are shown to exhibit significantly different fatigue lives when subjected to geographically varying environmental conditions.

While turbines located at the M4 (west coast) and NREL benchmark sites fall significantly short of the standard 20-year design lifespan of modern wind turbines, the strong energy production potential makes these sites highly attractive for wind energy development. Given the promising wind resources at these locations, there is a clear incentive to ensure the long-term viability of turbine installations, emphasising a need to develop strategies to enhance fatigue resistance. Motivated by their strong vibration mitigation capabilities, auxiliary damping devices, namely a TMD and TMDI, were investigated with respect to their influence on the fatigue life of the welded connection. In Chapter 6, these devices were integrated into the MBD model within the composite modelling framework, allowing their dynamic influence on structural response to be captured under realistic operating conditions. Both the TMD and TMDI demonstrated strong mitigation capabilities, providing significant damping in their targeted direction. The inclusion of the inerter enabled the TMDI to achieve superior mitigation capabilities with a significantly reduced stroke, allowing it to operate in spatially constrained environments. However, while these enhancements were achieved in their intended direction, their influence on all modes of vibration must be carefully evaluated to ensure balanced performance. It was found that as a result of the additional mass of the damping device and coupling of the side-to-side and fore-aft responses, a slight increase in the structural response was observed in the direction perpendicular to that targeted by the device. This adverse effect on the turbine's fore-aft response proved to be particularly pronounced within the critical wind speed range identified in Chapter 3, where fore-aft moments dominate the resulting stress. The inclusion of the inerter further intensified this negative influence through its mass amplification effect, leading to an increased fore-aft response compared to the TMD. This secondary effect significantly hindered the fatigue life improvements that might otherwise be achieved through the use of damping devices. This emphasises the importance of evaluating cross-directional effects when implementing damping strategies in offshore wind turbine

structures. Ultimately, both the TMD and the TMDI contributed to an increase in the fatigue life of the weld, with the TMD demonstrating superior performance. However, the improvements achieved by both devices were marginal. Therefore, despite their proven ability to mitigate vibrations, neither the TMD nor the TMDI can be considered as particularly effective for reducing fatigue damage in offshore wind turbine towers, indicating the need for alternative or complementary strategies in future design approaches.

7.2 Suggestions for Future Work

The fatigue assessment framework developed in this work follows the procedures outlined in IEC 61400 [49] and incorporates site-specific wind and wave conditions. However, it is important to acknowledge certain limitations concerning the scope of the loading conditions considered in the analysis. Within this thesis, the fatigue analyses are restricted to DLC 1.2, representing normal, power-production operating conditions. While this is a fundamental and often dominant contributor to the total cumulative fatigue damage, this current scope excludes several non-operational and transient states. These states, including start-up, shut-down, idling, fault and storm conditions are notable, as the loads arising from such events have been proven to be significant [209]. To achieve a more comprehensive fatigue assessment in accordance with the full IEC 61400 recommendations, these additional fatigue load cases should be incorporated into the analysis. An extension of the current framework, following approaches similar to those in [249, 250] would enable a more precise quantification of the influence of these conditions on the 15-MW wind turbine, thereby providing fatigue life predictions with enhanced accuracy.

The consideration of storm events and their impact on fatigue damage is becoming increasingly important as the effects of climate change, discussed in Chapter 1, continue to intensify. With the progression of global warming, severe weather events such as storms are expected to become more frequent and intense, leading to increased structural loading during these events. Such changes in global weather patterns will not only affect extreme events but will additionally alter typical operational conditions. Some regions may experience stronger and more turbulent wind regimes, while others may encounter reduced wind speeds. In addition to variation in wind flow characteristics, rising sea levels may modify coastal currents, increase water depths around turbine foundations, and amplify wave energy, all of which

contribute to higher hydrodynamic loads. Although the influence of evolving environmental conditions is beyond the scope of the present work, future studies should incorporate these factors to further improve the accuracy and robustness of the fatigue assessment framework presented here. Accounting for the increasing frequency and severity of extreme weather, alongside broader environmental changes, will be essential for ensuring reliable long-term turbine performance.

Future works could additionally address the limitations regarding the loading assumptions by exploring the influence of wind-wave misalignment on fatigue damage accumulation. Investigating the effects of misaligned loading would provide greater confidence in fatigue assessments by ensuring that a broader range of dynamic scenarios are considered, including those that may give rise to potential instabilities in the turbine's response. However, accounting for misaligned wind and wave loading significantly increases the number of required simulations, resulting in substantial computational demands.

To address this challenge, machine learning approaches could be explored as a means of reducing computational effort. In particular, algorithms such as Long Short-Term Memory (LSTM) networks are particularly well-suited for this application, as their ability to learn temporal dependencies from limited high-fidelity simulation data enables accurate prediction of stress time histories for complex dynamic systems. Physics-Informed Neural Networks (PINNs) also show strong potential in this field, such as the work of Baisthakur and Fitzgerald [207], as they incorporate governing physical constraints directly into the training process. This enables PINNs to maintain physical consistency even when data is sparse under certain operating conditions, ensuring that predictions remain physically realistic. Integrating such models to approximate the output of the MBD model could dramatically reduce the computational burden of the framework. This would enable researchers to perform a greater number of simulations across a broader range of loading scenarios, thereby improving the robustness and scalability of future fatigue assessments.

The integration of machine learning techniques to reduce computational effort would not only support extended fatigue analyses but also enable a broader investigation of fatigue mitigation strategies. While the auxiliary damping devices examined in this work are limited in their capacity to reduce fatigue damage, alternative or complementary approaches may be considered. Advanced control al-

gorithms, for instance, may be developed to maximise power output while simultaneously minimising fatigue loads. These control strategies could be explored both independently and in combination with auxiliary damping devices, potentially offering a more adaptive and effective approach to fatigue mitigation in offshore wind turbine systems. In addition, alternative damping devices could be considered in future studies to further enhance vibration mitigation and reduce fatigue damage. These could include passive systems such as pendulum TMDs, tuned liquid dampers (TLDs), tuned liquid column dampers (TLCDs), as well as variations of semi-active and active damping devices. Furthermore, alternative configurations involving multiple damping devices along the tower could be investigated to optimise performance across different vibration modes and improve overall fatigue reduction.

With regard to the FE model, this work adopted the conventional approach of uniform corrosion to account for the effects of corrosion over time. However, further refinement could be achieved through localised corrosion modelling strategies, such as pitting corrosion models or stochastic approaches. This could offer enhanced insights into non-uniform material loss and its impact on stress concentrations and crack initiation. Future work could also consider coupling the FE model with advanced corrosion-fatigue deterioration frameworks, such as the model employed by Zhang and colleagues [42], to account for the interaction between time-dependent corrosion processes and cyclic loading. Incorporating such methods would provide a more realistic representation of long-term degradation, thereby enhancing the accuracy of fatigue life predictions for offshore structural components.

While the parametric nature of the FE model in this work was initially developed to account for material degradation effects such as corrosion, this framework can be readily extended to support design optimisation studies. Key structural parameters, including the shell thickness of the wind turbine tower, the geometric dimensions of flange connections, and associated material properties, can be systematically varied to evaluate their influence on fatigue performance. Such an approach would allow for the identification of fatigue-resilient design configurations, balancing structural integrity with material efficiency and cost-effectiveness. By integrating this capability within the modelling framework, future work could explore optimal design strategies tailored to site-specific loading conditions and long-term operational requirements.

Finally, as discussed in Section 3.3, soil structure interactions can influence

the overall dynamics of the wind turbine and, consequently, the stresses experienced by components such as the ring flange connection. However, in this work, a fixed base condition is assumed in the composite model framework. Future work should extend the current approach to account for SSI effects, this could be implemented in both the MBD model and the FE model through the inclusion of appropriate soil spring representations. In the MBD model, the soil–structure interaction can be represented through a linearised 6×6 stiffness and damping matrix applied to the platform, capturing the coupled translational and rotational flexibility of the foundation. In the FE model, the soil response could be incorporated by applying a series of distributed nonlinear spring elements along the embedded length of the monopile to represent lateral, axial, and tip soil resistance. Employing these techniques with site-specific geotechnical parameters would allow for a more accurate estimation of foundation stiffness, modal properties, and fatigue damage at critical connections such as the ring flange.

Appendix A

Kane's Dynamic Equations: Generalised Inertial and Active Forces Parameter Definitions

Generalised equilibrium equations for a simple holonomic multi-body system:

$$F_r + F_r^* = 0 \quad \text{for } r = 1, 2, \dots, N \quad (\text{A.1})$$

The generalised inertial forces are:

$$F_r^* = F_r^*|_X + F_r^*|_T + F_r^*|_N + F_r^*|_H + F_r^*|_{B1} + F_r^*|_{B2} + F_r^*|_{B3} + F_r^*|_G \quad (\text{A.2})$$

where

F_r^* - Generalised inertial forces

$F_r^*|_X$ - Inertial forces of the platform

$F_r^*|_T$ - Inertial forces of the tower

$F_r^*|_N$ - Inertial forces of the nacelle

$F_r^*|_H$ - Inertial forces of the hub

Appendix A. Kane's Dynamic Equations: Generalised Inertial and Active Forces Parameter Definitions

$F_r^*|_{B1}$ - Inertial forces associated with blade 1

$F_r^*|_{B2}$ - Inertial forces associated with blade 2

$F_r^*|_{B3}$ - Inertial forces associated with blade 3

$F_r^*|_G$ - Inertial forces of the generator

The generalised active forces are:

$$\begin{aligned}
 F_r = & F_r|_{\text{AeroT}} + F_r|_{\text{AeroB1}} + F_r|_{\text{AeroB2}} + F_r|_{\text{AeroB3}} + F_r|_{\text{GravT}} \\
 & + F_r|_{\text{GravN}} + F_r|_{\text{GravH}} + F_r|_{\text{GravB1}} + F_r|_{\text{GravB2}} + F_r|_{\text{GravB3}} \\
 & + F_r|_{\text{YawStiff}} + F_r|_{\text{YawDamp}} + F_r|_{\text{Gen}} + F_r|_{\text{Brake}} + F_r|_{\text{ElasticT}} \\
 & + F_r|_{\text{ElasticB1}} + F_r|_{\text{ElasticB2}} + F_r|_{\text{ElasticB3}} + F_r|_{\text{DampT}} + F_r|_{\text{DampB1}} \\
 & + F_r|_{\text{DampB2}} + F_r|_{\text{DampB3}} + F_r|_{\text{ElasticDrive}} + F_r|_{\text{DampDrive}}
 \end{aligned} \tag{A.3}$$

Where

F_r - Generalised active forces

$F_r|_{\text{AeroT}}$ - Aerodynamic forces acting on the tower

$F_r|_{\text{AeroB1}}$ - Aerodynamic forces acting on blade 1

$F_r|_{\text{AeroB2}}$ - Aerodynamic forces acting on blade 2

$F_r|_{\text{AeroB3}}$ - Aerodynamic forces acting on blade 3

$F_r|_{\text{GravT}}$ - Gravitational force acting on the tower

$F_r|_{\text{GravN}}$ - Gravitational force acting on the nacelle

$F_r|_{\text{GravH}}$ - Gravitational force acting on the hub

$F_r|_{\text{GravB1}}$ - Gravitational force acting on blade 1

$F_r|_{\text{GravB2}}$ - Gravitational force acting on blade 2

$F_r|_{\text{GravB3}}$ - Gravitational force acting on blade 3

$F_r|_{\text{YawStiff}}$ - Restoring moment generated by the yaw stiffness at the nacelle-tower interface

$F_r|_{\text{YawDamp}}$ - Damping torque resisting nacelle yaw motion

$F_r|_{\text{Gen}}$ - Generator torque

$F_r|_{\text{Brake}}$ - Braking torque applied to the drivetrain

$F_r|_{\text{ElasticT}}$ - Elastic forces within the tower

$F_r|_{\text{ElasticB1}}$ - Elastic forces within blade 1
 $F_r|_{\text{ElasticB2}}$ - Elastic forces within blade 2
 $F_r|_{\text{ElasticB3}}$ - Elastic forces within blade 3
 $F_r|_{\text{DampT}}$ - Structural damping forces within the tower
 $F_r|_{\text{DampB1}}$ - Structural damping forces in blade 1
 $F_r|_{\text{DampB2}}$ - Structural damping forces in blade 2
 $F_r|_{\text{DampB3}}$ - Structural damping forces in blade 3
 $F_r|_{\text{ElasticDrive}}$ - Elastic restoring torque in the drivetrain due to torsional flexibility
 $F_r|_{\text{DampDrive}}$ - Damping in the drivetrain

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