FAILURE ANALYSIS OF THE PROPELLER SHAFT OF A TWIN SCREW PASSENGER FERRY

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A PROJECT SUBMITTED TO THE DEPARTMENT OF MARINE ENGINEERING, FACULTY OF ENGINEERING, UNIVERSITY OF BENIN, BENIN CITY IN PARTIAL FULFILMENT OF THE REQUIREMENTS FOR THE AWARD OF BACHELOR OF ENGINEERING (B.ENG) DEGREE IN MARINE ENGINEERING.

FEBRUARY, 2025

CERTIFICATION

This project was carried out by Oriyomi Anjuwon Oluwapelumi Mathew, Enabulele Osaivbie Emmanuel, Agbonfoh Eromosele Emmanuel, Alegieuno Omoghena Leonard the department of Marine Engineering, Faculty of Engineering, University of Benin, Benin City, and is hereby certified.

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DEDICATION

This project work is dedicated to God, the Almighty, giver of Wisdom, Knowledge and Understanding for providing the enabling supports and the necessary assistance for us successfully complete this program.

ACKNOWLEDGEMENT

We extend our heartfelt gratitude to each member of our dedicated team **Anjuwon Oluwapelumi Mathew, Enabulele Osaivbie Emmanuel, Agbonfoh Eromosele Emmanuel, Alegieuno Omoghena Leonard** for their unwavering commitment, tireless effort and exceptional expertise in successfully completing this project on Failure analysis of a propeller shaft in a twin screw passenger ferries. The collective efforts and collaboration of every team member have been instrumental in achieving our goals and delivering a high-quality outcome.

A special acknowledgement goes to our supervisor, **Engr. Dr. I.B. Owunna,** whose invaluable guidance, support and mentorship have been pivotal in steering us in the right direction throughout this project. Their expertise and encouragement have inspired us to push our boundaries and strive for excellence. We are deeply appreciative of the individuals and organizations who generously provided resources, data and valuable insights that enriched our research and analysis. Their contributions have significantly enhanced the depth and breadth of our project, enabling us to explore new avenues and make meaningful discoveries in the field of Failure analysis of a propeller shaft in a twin screw passenger ferries. Furthermore, we express our sincere thanks to our families and friends for their unwavering support, understanding and encouragement during the course of this project. Their patience, belief in our abilities, and words of encouragement have been a source of strength and motivation, propelling us forward even during challegingtimes. In conclusion, we extend our heartfelt thanks to all those who played a role in the successful completion of this project. Your collective efforts, support and contributions have been invaluable, and we are truly grateful for the opportunity to work alongside such dedicated and talented individuals.

ABSTRACT

The propeller shaft is a critical component in marine propulsion systems, transmitting power from the engine to the propellers. In twin-screw passenger ferries, failure of the propeller shaft can lead to severe operational disruptions, safety hazards, and costly repairs. This study presents a comprehensive failure analysis of a propeller shaft from a twin-screw passenger ferry to determine the root cause of failure and suggest mitigation strategies. The investigation includes visual inspection, non-destructive testing (NDT), metallurgical analysis, and finite element analysis (FEA) to evaluate material properties, fatigue characteristics, and stress distribution. The findings indicate that fatigue failure, corrosion-assisted cracking, misalignment, and improper lubrication are potential contributing factors. Based on the results, recommendations for improved maintenance, material selection, and design modifications are proposed to enhance the reliability and longevity of propeller shafts in marine vessels. This study provides valuable insights into preventing similar failures in future maritime applications.

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FEA	Finite Element Analysis
ANSYS	Analysis System
IMO	International Maritime Organization
CAD	Computer-Aided Design
FOS	Factor of Safety
NDT	Non-Destructive Testing
kW	Kilowatts
T	Torque
P	Power
F	Rotational Frequency
N	Newton
Rad/s	Radian per Second
Rpm	Revolution per Minute
M	Meters
Pa	Pascals

CHAPTER ONE

INTRODUCTION

1.1 Background of the Study

The propeller shaft is an essential component in marine vessels, responsible for transmitting power from the engine to the propeller and facilitating propulsion. Given the demanding mechanical loads and the harsh marine environment, propeller shafts are susceptible to failure modes such as fatigue, corrosion, and wear, which can result in vessel downtime, costly repairs, and even catastrophic failures at sea. Understanding and mitigating these failure mechanisms are critical in improving the safety and operational efficiency of marine propulsion systems (Pantazopoulos & Papaefthymiou, 2015).

Marine propeller shafts endure cyclic loading conditions that introduce complex stress variations over time, including bending, torsional, and axial stresses. Research has shown that these repetitive stress cycles lead to fatigue failure, where micro-cracks gradually form and propagate within the shaft material, ultimately resulting in complete fracture. Corrosion, particularly in seawater, exacerbates this degradation through corrosion fatigue, where saltwater accelerates material weakening by creating localized pits that act as stress concentrators. These pits increase mechanical stress, significantly shortening the shaft's service life (Zerbst et al., 2019; Pantazopoulos & Papaefthymiou, 2015).

Studies indicate that fatigue cracks in propeller shafts often originate at stress concentration points, such as the keyway or other surface flaws introduced during manufacturing or maintenance. Techniques like burnishing, which improves surface quality by inducing compressive residual stresses, have been employed to reduce crack initiation and extend the fatigue life of shafts. For example, burnishing decreases surface roughness, which helps resist crack formation under cyclic loading, especially for shafts exposed to high-cycle fatigue (Dyl & Dolny, 2010; Chandra Mohana Reddy, 2020).

Finite Element Analysis (FEA) has become a valuable method in marine engineering for analyzing these stress and load distributions. FEA simulation allows engineers to identify critical areas within the shaft that may be prone to stress concentrations, guiding material selection and structural improvements to withstand operational stresses. Such modeling has led to the optimization of component designs, improving fatigue resistance and overall durability, especially in components exposed to high torsional and bending stresses.

The current study investigates the failure mechanisms of a propeller shaft in a marine vessel by analyzing stress distributions, critical failure points, and fatigue behavior under operational conditions. This analysis aims to provide insights into potential failure modes and guide design improvements that could extend the operational life of propeller shafts, ultimately contributing to safer and more reliable marine vessel operations.

1.2 Statement of the Problem

The propeller shaft of a marine vessel, especially in twin-screw passenger ferries, is a critical component that experiences significant mechanical stresses during operation. As the primary means of power transmission from the engine to the propellers, this shaft endures cyclic loading, bending, and torsional stresses, which can lead to various failure modes. Among these, *fatigue failure* from repeated cyclic loading and *bending or torsional failure* under heavy loads are particularly concerning.

Marine environments exacerbate these stresses, as propeller shafts must withstand fluctuating forces from turbulent water conditions, speed variations, and frequent shifts in load. These factors, combined with high power requirements in modern vessels, heighten the risk of failure, posing safety risks for passengers and potential operational downtime. Current design standards often struggle to accommodate these elevated loads, particularly in scenarios where the power demands exceed traditional thresholds, such as 10,000 kW and 20,000 kW.

Without thorough analysis and optimization, the likelihood of shaft failure increases, potentially leading to costly repairs, safety hazards, and environmental risks. Therefore, it is crucial to identify and understand the failure mechanisms of the propeller shaft under these conditions to develop more robust designs that can withstand the extreme forces and power levels involved. This study seeks to address this problem by analyzing the failure modes of the propeller shaft under cyclic loading and varying power scenarios using simulation tools, with the goal of proposing solutions for enhanced shaft durability and reliability..

1.3 Aim and Objectives of the Study

To analyze and understand the failure mechanisms of the propeller shaft in a twin-screw passenger ferry, focusing on fatigue, bending, and torsional stresses under cyclic loading and varying power conditions, to enhance design robustness and operational safety.

The following objectives will be pursued to achieve the aim of the study:

1. To conduct a detailed simulation analysis using Ansys Mechanical to assess the impact of cyclic loading on the propeller shaft.

2. To evaluate the effects of bending and torsional stresses on the shaft under high power scenarios (10,000 kW and 20,000 kW).

3. To identify critical stress points and likely failure zones within the shaft, particularly under the combined effects of cyclic, bending, and torsional loading.

4. To compare failure risks between different power scenarios and establish correlations between power levels and shaft durability.

5. To propose design recommendations aimed at minimizing failure risks, including potential material and structural modifications, for enhanced performance and longevity of marine propeller shafts.

1.4 Scope of Study

This study focuses on analyzing the failure mechanisms of the propeller shaft in a twin-screw passenger ferry, specifically under the influence of fatigue, bending, and torsional stresses. The scope includes:

1. **Simulation Analysis**: Utilizing Ansys Mechanical to simulate the mechanical behavior of the propeller shaft under cyclic loading, as well as under two power scenarios—10,000 kW and 20,000 kW. This simulation will help assess the effects of increased power on stress distribution and potential failure points.

2. **Failure Modes**: The study concentrates on key failure modes relevant to marine propeller shafts, namely:

    i. Fatigue failure due to cyclic loading.

ii. Bending and Torsional failure resulting from external heavy loads and high torque levels.

3. **Load Scenarios**: The study will analyze the shaft under two distinct power conditions to simulate operational variability, exploring how different load levels impact the shaft's structural integrity.

4. **Identification of Critical Stress Zones**: Through simulation, the study aims to locate areas of high stress and potential points of failure on the shaft, particularly those likely to initiate fatigue or experience maximum deformation under combined loading conditions.

5. **Recommendations for Design Improvement**: Based on the findings, the study will suggest potential design modifications, material changes, or structural reinforcements to improve the durability and performance of marine propeller shafts.

This scope is confined to the theoretical and simulation-based analysis of the propeller shaft, without experimental testing. The insights gained will be applicable to similar vessels and may inform future studies aimed at optimizing marine propeller shaft design for enhanced safety and longevity.

1.5 Significance of the Study

The study on the failure analysis of a marine vessel's propeller shaft is significant for several reasons:

1. **Enhancing Safety and Reliability**: By understanding the primary failure mechanisms affecting the propeller shaft, this study contributes to

the safety and operational reliability of marine vessels. Identifying fatigue, bending, and torsional stress points helps reduce the risk of unexpected failures, which can jeopardize passenger and crew safety.

2. **Optimizing Design and Performance**: The insights gained from this analysis can inform improvements in propeller shaft design, material selection, and structural integrity. Recommendations for design modifications can lead to enhanced durability, allowing the shaft to better withstand cyclic loading and high-power conditions.

3. **Advancing Simulation-Based Failure Analysis**: The study demonstrates the application of advanced simulation tools (Ansys Mechanical) for failure analysis, which can serve as a model for similar research in other marine engineering contexts. It highlights the value of simulation in predicting failure modes without needing extensive physical testing.

CHAPTER TWO

LITERATURE REVIEW

2.1 Brief History of Marine Propeller Shafts

Marine vessels, before the 19th century, were primarily powered by oars, sails, or paddle wheels. Propeller shafts were unnecessary until the invention of the screw propeller, which transformed propulsion technology. Although early concepts of screw-based propulsion can be traced back to Leonardo da Vinci's sketches in the 15th century, it was not until the 1830s that practical applications emerged. This breakthrough was largely credited to John Ericsson and Sir Francis Pettit Smith, who developed and patented screw propellers that could be driven by a rotating mechanism beneath the waterline, necessitating the use of propeller shafts to transmit engine power efficiently.

The screw propeller saw initial applications in small vessels, but its success quickly caught the attention of the Royal Navy and the U.S. Navy. The new propulsion technology required durable and reliable propeller shafts, capable of withstanding water pressure, load, and rotational forces. Early shafts were typically made from wrought iron, which limited their strength and led to frequent failures, motivating engineers to experiment with materials and designs that could support the increased demands of larger ships.

The advent of steam engines in the mid-19th century marked a new era in shaft design. With steam engines producing higher power, shafts needed to efficiently handle greater torque and rotational forces. Twin-screw designs also became popular, especially in larger steamships, to enhance propulsion efficiency and improve navigability. However, these designs required dual shafts, which placed further demands on shaft engineering. To meet these needs, steel began to replace iron in shaft construction, offering superior strength and durability, which were essential for withstanding the loads and stresses associated with steam -powered vessels.

As the 20th century approached, marine propulsion advanced further with the introduction of diesel and gasoline engines. These engines provided higher power output, fuel efficiency, and speed, which placed even more demands on propeller shaft design. During this time, international organizations like the International Maritime Organization (IMO) began to establish standards to ensure the safety and reliability of propulsion systems. Advances in metallurgy during this period also led to the development of alloyed steel shafts, such as stainless steel and nickel-chromium alloys, which offered both strength and corrosion resistance, making them suitable for the harsh marine environment.

The latter half of the 20th century saw further refinement in shaft technology with the growth of high-speed ferries, container ships, and naval vessels. Propeller shafts were engineered to handle greater power, withstand severe cyclic loading, and operate more efficiently. The emergence of finite element analysis (FEA) and computer-aided design (CAD) transformed shaft engineering, allowing engineers to simulate stress, fatigue, and failure modes with high precision. Tools like Ansys Mechanical enabled designers to analyze critical points on shafts and optimize designs to balance performance with safety and efficiency. In recent years, advanced materials like carbon fiber composites have begun to find their way into propeller shaft design. Composite shafts offer the potential for reduced weight and improved resistance to corrosion, making them ideal for high-speed applications. Additionally, hybrid shafts that combine metals with composite materials have been developed to enhance both strength and flexibility. Smart shafts equipped with sensors to monitor stress, temperature, and load in real time have also emerged, providing data for predictive maintenance and reducing the likelihood of in-service failures. Some researchers are even exploring 3D printing for specific shaft components, which could lower manufacturing costs and enable greater customization.

2.2 Overview of Marine Propeller Shaft

The marine propeller shaft transmits the rotational power generated by the engine to the propeller, converting engine energy into thrust to propel the vessel. It is designed to handle high loads, resist corrosion, and maintain efficiency under various operational conditions.

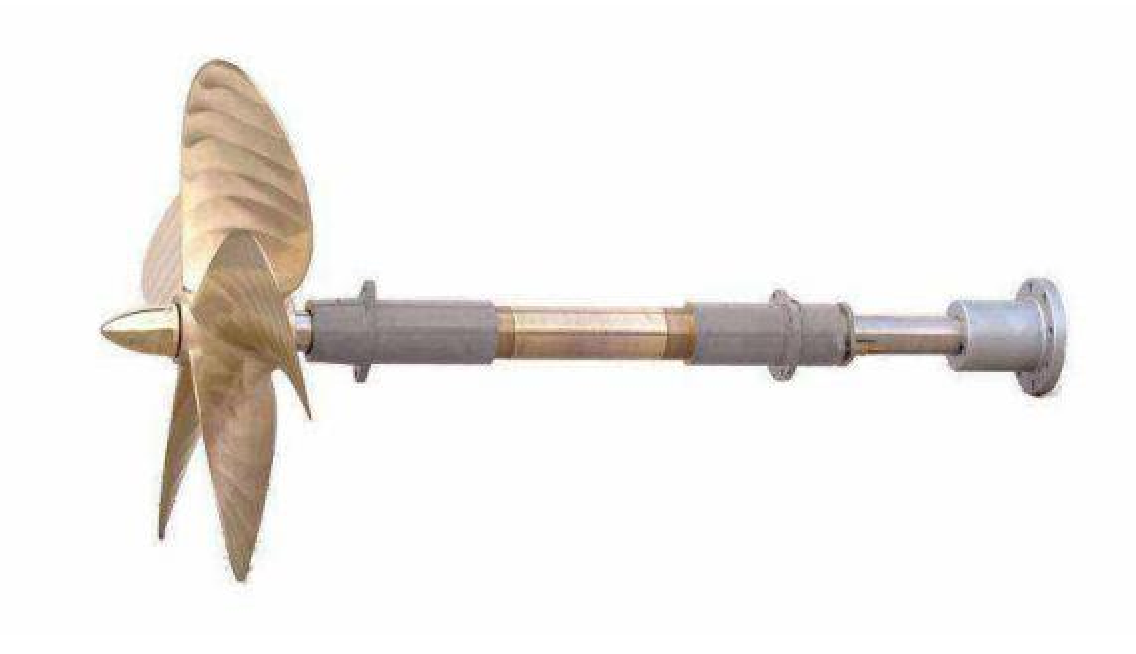

Figure 2.1: Marine Propeller Shaft

2.3 Main Parts of the Propeller Shaft System

The marine propeller shaft system consists of several parts, each with a specific role in ensuring smooth and efficient power transmission.

1. **Main Shaft**: The central part of the propeller shaft system, extending from the gearbox to the propeller. This shaft is typically long, slender, and made of high-strength materials to handle the torque from the engine.

2. **Intermediate Shaft**: In larger vessels, an intermediate shaft may be placed between the engine and the propeller shaft to extend the power transfer distance. It provides flexibility in aligning the engine and propeller position and helps distribute load more evenly.

3. **Tail Shaft**: Also called the *propeller shaft*, the tail shaft is the final part in the power transmission line, connecting directly to the propeller. The tail shaft passes through a stern tube that is sealed and lubricated to protect it from seawater.

4. **Couplings**: Shaft couplings join different sections of the shafting system (like the main shaft to the intermediate shaft) or connect the shaft to the gearbox. Common types include flexible couplings, which accommodate slight misalignments, and rigid couplings for precise, secure connections.

5. **Stern Tube**: The stern tube houses the tail shaft where it exits the hull, protecting it from seawater intrusion and supporting the tail shaft with bearings. Inside the stern tube, seawater-lubricated or oil-lubricated bearings are used.

6. **Bearings**: Bearings support the shaft and reduce friction during rotation. Marine propeller shafts commonly use thrust bearings to manage axial loads

and journal bearings to handle radial loads. These bearings are essential for stable and efficient rotation, helping to maintain alignment and prevent wear.

7. **Seals**: Seals, particularly around the stern tube, prevent seawater from entering the hull. They also retain lubrication, especially in oil-lubricated stern tubes. Seals are critical for protecting the shaft and maintaining the vessel's integrity.

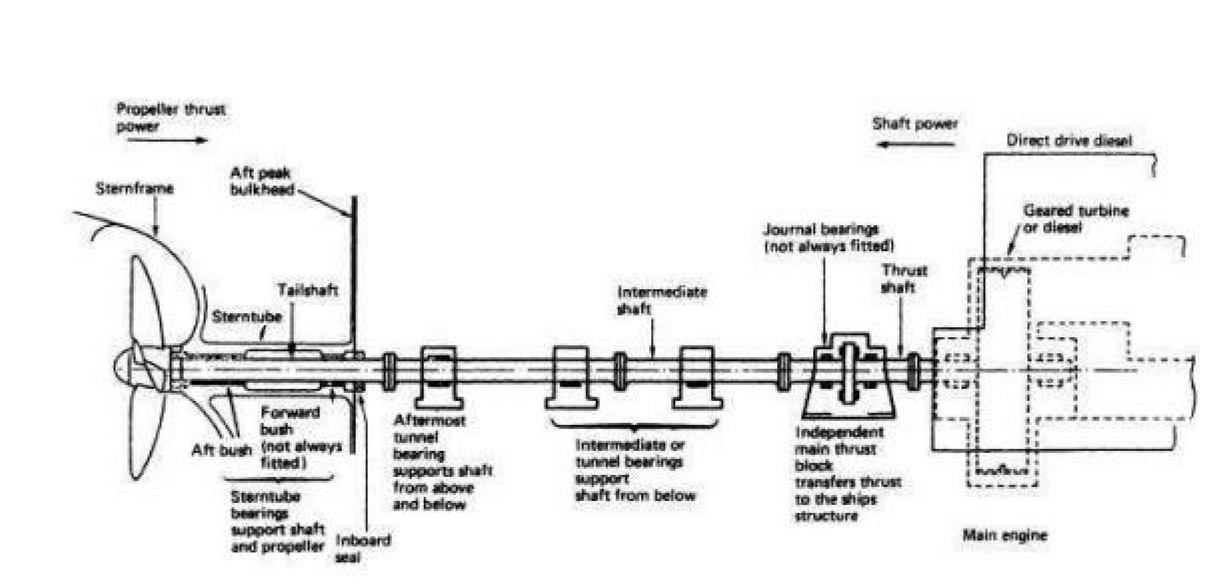

Figure 2.2: Labelled Diagram of Marine Propeller Shaft.

2.4 Types of Propeller Shafts

Marine propeller shafts vary based on their design, material, and application. The main types are:

1. **Solid Shafts**: These are the most common type, made from a single piece of material (usually alloy steel or stainless steel)

and are very robust. They are preferred for vessels that require high strength and durability.

2. **Hollow Shafts**: Hollow shafts are used in high-performance applications where weight reduction is crucial. They are ighter than solid shafts but still offer adequate strength and are used in fast-moving vessels and military applications.

3. **Flexible Shafts**: These shafts are designed to accommodate slight misalignments and absorb vibrations, which helps in reducing noise and wear on the shaft and bearings. They are useful in vessels where minor movement between components may occur.

4. **Composite Shafts**: Made from materials like carbon fiber or fiberglass, composite shafts are lightweight and corrosion-resistant, making them ideal for high-speed vessels. However, they are more expensive and generally limited to specialized applications..

2.5 Materials Used in Marine Propeller Shafts

Marine environments are highly corrosive, and propeller shafts must withstand a range of mechanical and environmental stresses. Common materials include:

1. **Stainless Steel**: Offers corrosion resistance and good strength, commonly used for medium-sized vessels.
2. **Alloy Steel**: Known for high strength and durability, alloy steel is commonly used in commercial and military vessels where robust performance is necessary.
3. **Duplex Stainless Steel**: Combines excellent strength and corrosion

resistance, suitable for harsh environments.

4. **Nickel-Chromium Alloys**: Provide superior fatigue resistance, corrosion resistance, and durability, ideal for high-performance vessels.

2.6 Shaft Alignment and Balancing

Proper alignment and balancing of the propeller shaft are crucial for reducing vibrations, minimizing wear on bearings, and optimizing fuel efficiency. Shaft alignment ensures that the shaft rotates on a true axis, preventing lateral or angular misalignment, which can lead to excessive wear, noise, and power loss. Balancing involves distributing the mass evenly along the shaft to reduce vibrations and ensure smooth operation..

2.7 Review of Existing Literature

The study of marine propeller shaft failures is critical to ensuring the reliability and safety of marine propulsion systems. Propeller shafts, which transmit power from the engine to the propellers, endure various mechanical stresses during operation. Understanding the factors leading to their failure can significantly enhance design practices and maintenance strategies.

The investigation of marine propeller shafts is crucial due to their significant role in propulsion systems. Various studies highlight the challenges of failure modes, materials, and design considerations for these components. Research has shown that material properties greatly affect fatigue life, with simulations revealing the importance of selecting suitable materials to enhance durability. Additionally, analyses of bending and torsional stresses have emphasized the need for robust designs and regular maintenance to prevent failures. Operational factors, such as alignment and lubrication, are critical in extending the service life of shafts. Recent advancements in computational techniques also shed light on the dynamic responses of shafts under hydrodynamic forces, underscoring the complexity of analyzing failure modes. Collectively, these findings advocate for ongoing research to improve safety and reliability in marine applications.

In a comprehensive study, Van et al. employed fault tree analysis to evaluate the reliability of marine propulsion systems, identifying common failure modes such as fatigue and bending under heavy loads. Their analysis emphasized the importance of understanding the interactions between different components within the propulsion system for predicting potential failures. This foundational

work guides the development of more resilient propulsion designs by pinpointing critical areas where failures are most likely to occur.

The classification rules established by Det Norske Veritas are vital for the integrity of marine vessels, as they outline stringent requirements for the design and maintenance of shaft systems. Compliance with these regulations enhances safety by ensuring propulsion shafts are engineered to withstand the varying loads encountered in operational settings. Failure to meet these standards can lead to catastrophic incidents at sea.

Han's research on fatigue failure specifically examined the impact of torsional vibrations on keyways in reduction gear input shafts, demonstrating how these vibrations significantly increase stress concentrations, leading to premature failure. This underscores the necessity of analyzing vibration patterns in marine environments, which are critical for the longevity of propulsion systems.

In further research, Han et al. conducted a parametric study to investigate the causes of high torsional vibrations in propulsion shafts, identifying factors such as engine misalignments and load variations that contribute to increased vibration levels. Recognizing these elements allows marine engineers to design systems that mitigate these stresses, enhancing overall reliability.

Sitthipong et al. performed a failure analysis on metal alloy propeller shafts, finding that material selection, particularly the choice of alloy and manufacturing processes, profoundly affects the fatigue life of shafts. This

highlights the importance of selecting appropriate materials to minimize fatigue failure risk over time, especially in harsh marine environments.

Guidelines set forth by organizations such as the International Association of Classification Societies and the American Bureau of Shipping outline specific design dimensions and permissible torsional vibration stresses for propulsion shafts, serving as benchmarks for designers. These regulations ensure that shafts are constructed to withstand the demanding conditions of marine operation while adhering to safety standards.

Through these collective insights, it is clear that addressing the multifaceted challenges associated with marine propeller shafts is essential for improving safety, enhancing design practices, and ensuring the longevity of marine propulsion systems. Continued research and adherence to established guidelines will play a pivotal role in reducing the occurrence of shaft-related failures and advancing marine engineering practices.

2.8 Operational Loads Acting on a Marine Propeller Shaft

In the operation of a marine propeller shaft, various forces act upon the shaft due to the dynamic environment and loads it experiences during operation.

Understanding these forces is crucial for ensuring the structural integrity and performance of the shaft.

First and foremost, **torque** is one of the primary forces acting on the shaft. As the engine generates power, this rotational force is transmitted through the shaft to drive the propeller. The magnitude of torque depends on the engine's output.

power and the rotational speed of the shaft, as described by the equation: Torque (T) = Power (P) / Angular Velocity (ω). This torque results in shear stress on the shaft material, and the design of the shaft must accommodate this stress to prevent failure.

In addition to torque, **bending forces** are also significant. When the propeller is engaged and producing thrust, the resulting force from the water can create bending moments along the shaft. This bending force arises from the reaction of the water against the propeller blades, which can exert a lateral load on the shaft. The bending moment is typically highest near the propeller and decreases along the length of the shaft. It is essential for the shaft to be designed with adequate stiffness to resist these bending moments without excessive deflection, which could lead to misalignment and increased wear on bearings and seals.

Another critical force is **axial force** or thrust, which occurs as the propeller pushes against the water to generate forward motion. This force acts along the shaft's length and is primarily caused by the propeller's action. When the propeller rotates, it not only creates thrust in the forward direction but also generates a corresponding axial load on the shaft, requiring the incorporation of thrust bearings to absorb these forces. The magnitude of axial force can vary with changes in speed and load conditions and is typically greatest during full power operations.

Furthermore, **torsional vibrations** can occur within the shaft, especially under variable load conditions. These vibrations result from oscillations in torque due to fluctuations in engine power or changes in propeller loading. Torsional

vibrations can lead to resonance phenomena, potentially causing fatigue and failure of the shaft material if not properly managed. Engineers often analyze the natural frequencies of the shaft and its supporting structures to mitigate these risks.

Lastly, **dynamic loads** resulting from the vessel's motion, including wave action and changes in speed or direction, can also affect the shaft. As the vessel navigates through varying sea conditions, these dynamic forces can impose additional stresses on the shaft, compounding the effects of static loads. The shaft must be designed to accommodate these variable conditions, ensuring that it can withstand the combined effects of static and dynamic forces over its operational lifespan.

CHAPTER THREE

METHODOLOGY

3.1 Overview of Materials

In the failure analysis of the marine vessel propeller shaft, two primary software materials were employed: **SolidWorks** for 3D design and assembly of the shaft components and **ANSYS** for simulation of the shaft's behavior under various loading conditions. Together, these tools enabled precise modeling and analysis of the shaft's performance under operational stresses, providing insights into potential failure modes.

3.2 Overview of SolidWorks

SolidWorks is a powerful 3D computer-aided design (CAD) software widely used in engineering and product design. Developed by Dassault Systèmes, SolidWorks enables designers and engineers to create, simulate, and analyze complex models across various industries, including automotive, aerospace, and marine engineering. Its intuitive interface and extensive range of tools make it particularly effective for designing mechanical parts, assemblies, and detailed technical drawings

3.2.1 Key Features and Capabilities

1. **3D Modeling**:

SolidWorks offers robust 3D modeling tools that enable users to build intricate part geometries and assemblies with precision. It supports various modeling techniques, including parametric, direct, and surface modeling, allowing flexibility in design. Parametric modeling in SolidWorks lets users define relationships and constraints between features, making it easier to adjust dimensions and automatically update dependent features.

2. **Assembly Design**:

With SolidWorks, multiple parts can be combined to form assemblies, where users can define how components interact and fit together. This feature is crucial for simulating mechanical systems, as it enables designers to visualize part interactions, identify potential interference, and check the alignment. Advanced mating options allow users to simulate motion, apply constraints, and test the functionality of moving parts, helping ensure that complex assemblies operate as intended.

3. **Simulation and Analysis**:

SolidWorks integrates basic simulation capabilities for initial testing and verification, allowing designers to perform stress, thermal, and motion analyses directly within the software. For more comprehensive simulations, users often export their models to specialized simulation software like ANSYS for advanced analysis, as was done in this project. Its integrated simulation tools support preliminary assessments of design strength and durability, aiding in identifying weak points or areas prone to deformation.

4. **Export and Interoperability**:

SolidWorks supports a range of file formats, such as **.iges**, **.step**, **.stl**, and **.dxf**, which facilitate compatibility with other CAD and simulation platforms. This versatility ensures that designs can be easily transferred to other software like ANSYS for further analysis or 3D printing. For this project, the completed shaft assembly was exported in **.iges format** to ensure smooth integration with ANSYS Mechanical, enabling efficient and accurate simulations.

5. **Technical Drawings and Documentation**:

SolidWorks includes powerful tools for generating 2D technical drawings, providing detailed views, annotations, and dimensional information essential for manufacturing. These drawings can include part dimensions, tolerances, and assembly instructions, creating a complete documentation package.

6. **User Interface and Workflow**:

The SolidWorks interface is designed for ease of use, with a customizable workspace that allows users to access tools efficiently. Features like drag- and-drop functionality, context-sensitive menus, and real-time feedback simplify the design process. Its workflow is intuitive, allowing designers to progress from concept sketches to detailed models and assemblies quickly.

SolidWorks was selected for the design and assembly of the propeller shaft and its components due to its advanced modeling capabilities and flexibility in managing complex geometries. It allowed for the creation of a detailed 3D CAD model that accurately represented the physical properties and dimensions of the shaft system.

Each component of the shaft, including the **tail shaft**, **intermediate shaft**, **thrust shaft**, and their corresponding **bearings**, was modeled individually. Once the parts were designed, they were assembled within SolidWorks to ensure proper alignment and interaction between components. The completed assembly was then saved and exported in **.iges format** for compatibility with ANSYS, facilitating an efficient transfer to the simulation phase.

3.3 Overview of ANSYS

ANSYS is a leading engineering simulation software used extensively in industries such as automotive, aerospace, electronics, and marine engineering. Developed by ANSYS, Inc., it offers a range of advanced simulation tools that allow engineers to model, analyze, and optimize product performance under various physical conditions. ANSYS is particularly valuable for simulating complex interactions of physical phenomena, including structural mechanics, fluid dynamics, thermal analysis, and electromagnetics.

3.3.1 Key Features and Capabilities

1. **Multiphysics Simulation**:

ANSYS stands out for its **multiphysics capabilities**, which enable users to simulate interactions between different physical forces in a single environment. Engineers can assess how structural, thermal, and fluid forces interact within a product, allowing for comprehensive analysis. This capability is critical for complex engineering applications where products encounter multiple simultaneous stresses, such as thermal and mechanical loads.

2. **Finite Element Analysis (FEA)**:

ANSYS employs **finite element analysis (FEA)** to divide models into smaller elements, making it possible to analyze the behavior of complex shapes under stress. This method enables precise calculation of stress, strain, deformation, and failure points. FEA is essential for studying structural behavior, particularly in applications like the propeller shaft analysis in this project, where understanding stress distribution and deformation is critical for preventing mechanical failure.

3. **Specialized Modules for Various Engineering Disciplines**:

ANSYS offers a variety of specialized modules for different engineering analyses, including **ANSYS Mechanical** for structural analysis, **ANSYS Fluent** for fluid dynamics, **ANSYS Maxwell** for electromagnetics, and **ANSYS CFD** for computational fluid dynamics. In this project, **ANSYS Mechanical** was used for the structural analysis of the marine vessel propeller shaft, examining the effects of bending, torsion, and cyclic loading to evaluate potential failure modes.

4. **Material Properties Library**:

ANSYS includes an extensive **material library** with detailed properties for thousands of engineering materials, such as metals, polymers, ceramics, and composites. This library allows users to select materials with specific mechanical properties to simulate realistic loading conditions. For accurate results, engineers can also customize materials by inputting specific material properties, which is useful for tailored analyses in industries where materials vary in composition and behavior.

5. **Advanced Meshing Capabilities**:

ANSYS provides robust **meshing tools** that allow users to create optimized meshes for accurate simulations. Its meshing capabilities include automatic and manual controls to refine the mesh where high accuracy is needed, balancing precision with computational efficiency. By refining the mesh around critical areas, such as high-stress points, ANSYS ensures that simulations accurately capture stress concentrations and potential points of failure.

6. **Analysis and Results Interpretation**:

ANSYS outputs a wide range of results, including **stress distributions, total deformation, fatigue life, and factor of safety**, providing engineers with comprehensive data to interpret the structural integrity of components. The software also provides visualization tools to create contour plots, animations, and graphs that illustrate the results, allowing engineers to understand how forces impact the structure and identify areas at risk of failure.

ANSYS was used to perform detailed simulations on the shaft assembly, examining how it would respond under specific loading scenarios. Its simulation capabilities allowed for the evaluation of stress distributions, deformation, and fatigue life, providing critical insights into the structural integrity of the shaft.

After importing the **.iges** model into ANSYS, material properties were assigned based on the shaft's specifications, and appropriate boundary conditions were established. The simulation parameters included axial and torsional loads, which were applied to replicate real-world conditions during operation

3.3.2 Analysis System: Static Structural Analysis

The **Static Structural Analysis** system within ANSYS was used to evaluate the propeller shaft's response to bending and fatigue under applied loads. This analysis system is particularly suited to assessing the effects of both static and cyclic loading on the structural components, making it ideal for examining two key failure modes:

1. **Bending and Torsional Failure Analysis**: The static structural module was used to simulate bending and torsional stresses when the shaft operated under high power inputs (10,000 kW and 20,000 kW scenarios at 300 RPM). This analysis determined the equivalent von Mises stress, total deformation, directional deformations, and factor of safety under applied loads, providing insights into potential bending or torsional failures.

2. **Fatigue Failure Analysis**: For fatigue analysis, the static structural system evaluated the shaft's performance under cyclic loading

conditions. The analysis produced results for fatigue life, damage accumulation, safety factor, and fatigue sensitivity, essential for assessing the shaft's endurance and identifying high-risk areas prone to fatigue failure.

3.4 Overview of Methods Used

This section provides an outline of the primary methods employed in the failure analysis of the propeller shaft of a marine vessel, focusing on **Geometric Modeling** and **Mathematical Modeling**. These methods were integral to accurately representing the physical structure and simulating the mechanical behavior of the shaft under operational loads.

3.4.1 Geometric Modeling

Geometric modeling involves creating a precise 3D representation of the propeller shaft and its components, which includes the **tail shaft**, **intermediate shaft**, **thrust shaft**, and their associated bearings. In this project, **SolidWorks** was utilized to design and assemble the shaft system. The following steps outline the geometric modeling process:

1. **Component Design**:

Each component of the propeller shaft system was individually designed in SolidWorks, with specific attention to details such as dimensions, shape, and material properties. This approach allowed for a high degree of accuracy in the final assembly.

2. **Assembly of Shaft Components**:

After each part was modeled, the components were assembled to ensure proper alignment and interaction, reflecting the real-world configuration of the propeller shaft. This assembly process is critical for simulating interactions between parts and evaluating how they respond under load.

3. **Exporting the Model**:

The complete assembly was exported in **.iges format** to facilitate compatibility with ANSYS. This format preserves the geometrical integrity of the model, enabling it to be imported into simulation software for further analysis. The geometric model developed in SolidWorks served as the foundation for load simulations and structural analysis in ANSYS, providing a realistic representation of the shaft system that allowed for accurate assessment of potential failure modes.

3.4.2 Mathematical Modeling

Mathematical modeling was employed to calculate the loads and stresses acting on the propeller shaft under different operational conditions. This process involved determining the torque and axial thrust applied to the shaft, based on the power input scenarios specified (10,000 kW and 20,000 kW). Key aspects of the mathematical modeling include:

1. **Torque Calculation**:

   The torque acting on the shaft was calculated using the formula:

   $T=P/2\pi f$

   where:

   T is the torque,

   P is the power (10,000 kW or 20,000 kW),

   f is the rotational frequency (300 RPM converted to rad/s).

This calculation provided the torque values for each power scenario, which were then applied as input loads in the ANSYS simulation.

2. **Axial Thrust and Propeller Weight**:

In addition to torque, the shaft experiences an axial thrust and a load due to the propeller's weight. These values, measured at **48069 N** for axial thrust and **10791 N** for propeller weight, were applied as forces in the model to represent real operating conditions.

3. **Static Structural Analysis for Load Application**:
   The calculated torques and forces were applied in ANSYS's **Static Structural module** to simulate the bending and torsional stresses on the shaft. This module allowed for a detailed assessment of the stress distributions and deformation under each load scenario.

Together, the geometric and mathematical modeling techniques provided a comprehensive basis for the simulation, allowing for the precise evaluation of structural integrity and fatigue life of the propeller shaft under realistic load conditions.

3.5 Simulation Flow Process for Failure Analysis of a Marine Vessel Propeller Shaft

This project involved an in-depth analysis of the failure modes of a marine propeller shaft used in a twin-screw passenger ferry. The primary objective was to evaluate the shaft's structural integrity under two theoretical loading conditions and to identify potential failure modes, particularly focusing on fatigue and bending or torsional failure.

3.5.1 CAD Model Preparation

SolidWorks was utilized to create a detailed CAD model of the propeller shaft assembly. The assembly included key components such as the tail shaft, intermediate shaft, thrust shaft, and the associated bearings.

Each component was designed based on specified dimensions to replicate the actual shaft assembly.

Upon completing the assembly, the model was exported in **.iges format** to ensure compatibility with **ANSYS Mechanical** and to preserve the geometry for further analysis.

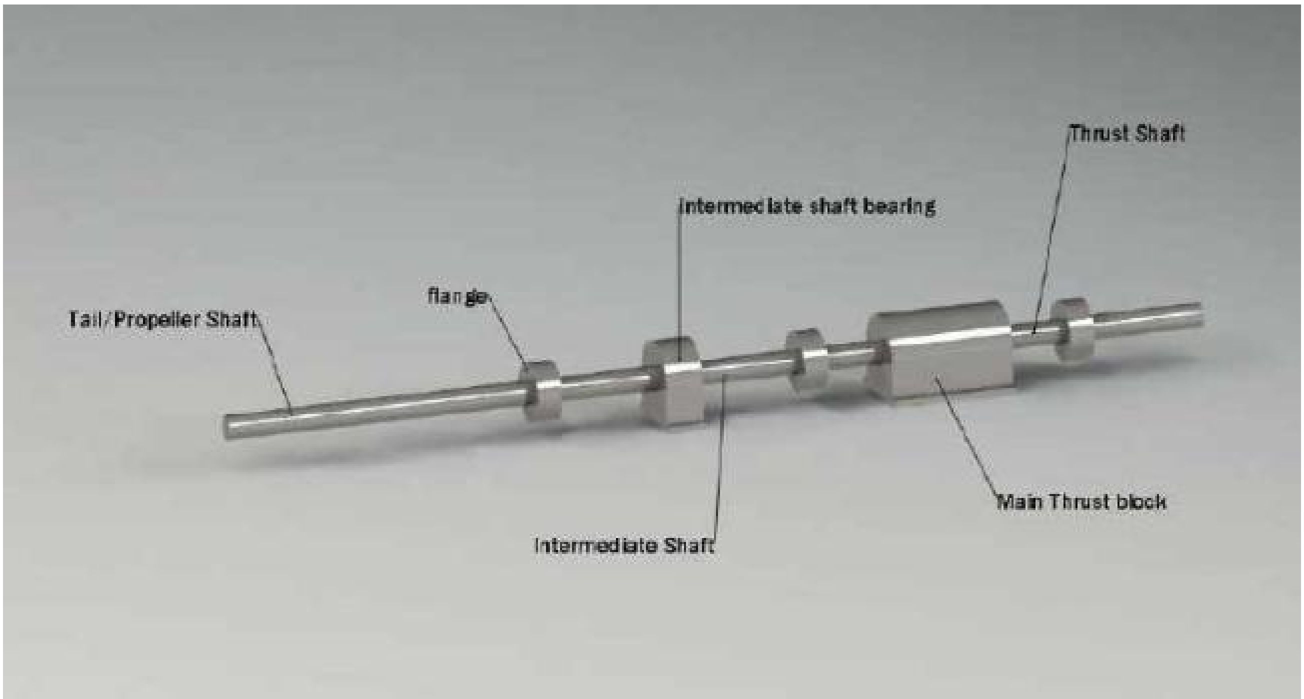

Figure 3.1: CAD model in Solidworks

3.5.2 Simulation Setup in ANSYS Mechanical

The **.iges** file was imported into ANSYS Mechanical. The imported geometry was then validated to confirm that all components were accurately represented.

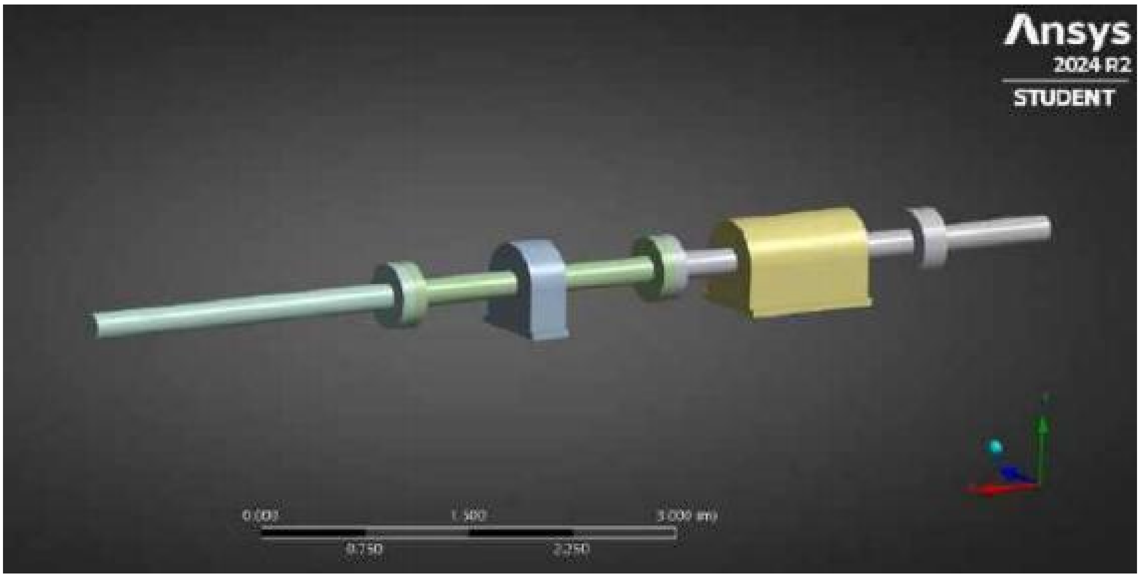

Figure 3.2: CAD model in Ansys (.iges format)

Structural Steel was used as the engineering material for the propeller shaft and the material properties came pre-installed with the Ansys software..

3.5.3 Material Data

The material data are as seen below

1. **Constants**

Density	7850 kg m^-3
Coefficient of Thermal Expansion	1.2e-005 C^-1
Specific Heat	434 J kg^-1 C^-1
Thermal Conductivity	60.5 W m^-1 C^-1
Resistivity	1.7e-007 ohm m

Table 3.1: Constants

2. **Compressive Yield Strength**

Compressive Yield Strength Pa
2.5e+008

Table 3.2: Compressive Yield Strength

3. Tensile Yield Strength

Tensile Yield Strength Pa
2.5e+008

Table 3.3: Tensile Yield Strength

4. Tensile Ultimate Strength

Tensile Ultimate Strength Pa
4.6e+008

5. Isotropic Secant Coefficient of Thermal Expansion

Zero-Thermal-Strain Reference Temperature C
22

Table 3.6 Structural Steel > S-N Curve

Alternating Stress Pa	Cycles	Mean Stress Pa
Alternating Stress Pa	Cycles	Mean Stress Pa
2.827e+009	20	0
1.896e+009	50	0
1.413e+009	100	0
1.069e+009	200	0
4.41e+008	2000	0

2.62e+008	10000	0
2.14e+008	20000	0
1.38e+008	1.e+005	0
1.14e+008	2.e+005	0
8.62e+007	1.e+006	0

Table 3.7: Strain-Life Parameters

Strength Coefficient Pa	Strength Exponent	Ductility Coefficient	Ductility Exponent	Cyclic Strength Coefficient Pa	Cyclic Strain Hardening Exponent
9.2e+008	-0.106	0.213	-0.47	1.e+009	0.2

Table 3.8: Isotropic Elasticity

Young's Modulus Pa	Poisson's Ratio	Bulk Modulus Pa	Shear Modulus Pa	Temperature C
2.e+011	0.3	1.6667e+011	7.6923e+010	

3.5.4 Application of Loads and Boundary Conditions

1. **Torque Calculation**:

Two loading scenarios were defined, one with a power input of **10,000 kW** and the other with **20,000 kW**, both at an operational speed of **300 RPM**. The corresponding torque values for each scenario were calculated using the formula: T=P/2πN where T represents torque, P is the power (10,000 kW or 20,000 kW), and N is the rotational speed of 300 RPM.

2. **Load Application**:

Calculated torques were applied along the shaft's axis for each scenario. Additionally, a **propeller weight of 10,791 N** was applied to simulate the propeller's weight, along with an **axial thrust of 48,069 N** to replicate the thrust generated during operation.

3. **Boundary Conditions**

The base of the bearings was fixed using **fixed support** constraints to simulate the actual anchoring points. **Revolute joints** were applied between the shaft and the round openings of the bearings, allowing rotation while restricting linear displacement, thus accurately replicating bearing behavior under operational conditions.

Table 3.9: Contact Regions

Object Name	*Bonded - drive To Thrust shaft*	*Frictionless - Intermediate shaft bearing To Intermediate shaft*	*Bonded - Intermediate shaft To Tail shaft*	*Bonded - Intermediate shaft To Thrust shaft*	*Frictionless - Stern tube housing To Tail shaft*	*Frictionless - Main thrust block To Thrust shaft*
State	Fully Defined	Suppressed	Fully Defined		Suppressed	
Scope						
Scoping Method	Geometry Selection					

Contact	1 Face	2 Faces	1 Face		No Selection	2 Faces
Target	1 Face	2 Faces	1 Face		2 Faces	
Contact Bodies	drive	Intermediate shaft bearing	Intermediate shaft		Stern tube housing	Main thrust block
Target Bodies	Thrust shaft	Intermediate shaft	Tail shaft	Thrust shaft	Tail shaft	Thrust shaft
Protected	No					
Definition						
Type	Bonded	Frictionless	Bonded		Frictionless	
Scope Mode	Automatic					
Behavior	Program Controlled					
Trim Contact	Program Controlled					
Trim Tolerance	2.0276e-002 m					
Contact APDL Name						
Target APDL Name						
Suppressed	No	Yes	No		Yes	
Display						
Element Normals	No					
Advanced						

Formuln	Program Controlled			
Sm all Slidi ng	Program Controlled			
etectio n Meth od	Program Controlled			
Penetrati on Toleran ce	Program Controlled			
Elastic Slip Toleran ce	Progra m Control le d		Program Controlled	
Nor mal Stiffn ess	Program Controlled			
Upd ate Stiffn ess	Program Controlled			
Pinb all Regi on	Program Controlled			
Stabiliza tio n Dampin g Factor		0.		0.
Time Step Contr ols		None		None
Geometric Modification				
Cont act Geomet ry Correcti on	None			
Tar get Geome try	None			
Correcti on				

Interface Treatment		Add Offset, No Ramping		Add Offset, No Ramping
Offset		0. m		0. m

Table 3.10: Joints

Object Name	*Fixed - Tail shaft To Intermediate shaft*	*Fixed - Intermediate shaft To Thrust shaft*	*Fixed - Thrust shaft To drive*	*Revolute - Stern tube housing To Tail shaft*	*Revolute - Intermediate shaft bearing To Intermediate shaft*	*Revolute - Main thrust block To Thrust shaft*	*Revolute - Ground To drive*
State	Suppressed				Fully Defined		
Definition							
Connection Type	Body-Body						Body-Ground
Type	Fixed			Revolute			
Solver Element Type	Program Controlled						
Element APDL Name							
Suppress	Yes			No			

ed							
Torsional Stiffness				0. N·m/°			
Torsional Damping				0. N·m·s/°			
Reference							
Scoping Method	Geometry Selection						
Applied By	Remote Attachment						
Scope	1 Face			No Selection	2 Faces		
Body	Tail shaft	Intermediate shaft	Thrust shaft	Stern tube housing	Intermediate shaft bearing	Main thrust block	
Coordinate System	Reference Coordinate System						
Behavior	Rigid						
Formulation	MPC						
Relaxation Method	No						
Pinball	All						

Region							
Mobile							
Scoping Method	Geometry Selection						
Applied By	Remote Attachment						
Scope	1 Face			2 Faces			
Body	Intermediate shaft	Thrust shaft	drive	Tail shaft	Intermediate shaft	Thrust shaft	drive
Initial Position	Unchanged						
Behavior	Rigid						
Formulation	MPC						
Relaxation Method	No						
Pinball Region	All						
Stops							
RZ Min Type				None			
RZ Max				None			

Ty pe		

3.5.5 Meshing of the Model

To achieve accurate results, a fine mesh was generated, especially in critical areas such as near the bearings and at high-stress concentration points along the shaft. This mesh refinement enabled precise calculation of stress concentrations, which is essential for identifying potential failure zones.

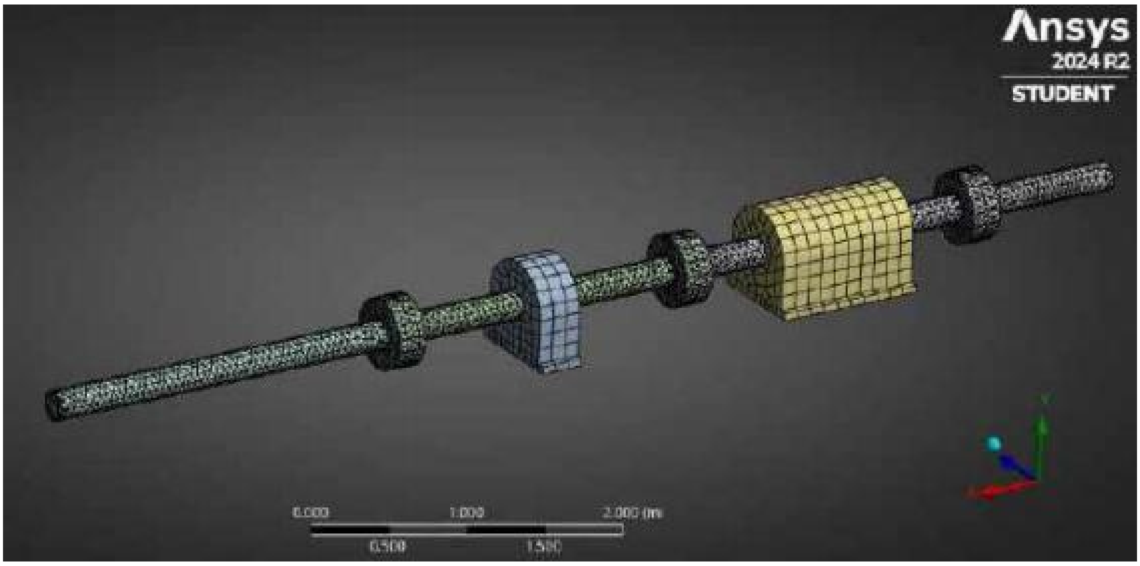

Figure 3.3: Meshed Model

Table 3.11 Mesh Details

Object Name	*Mesh*
State	Solved
Display	
Display Style	Use Geometry Setting
Defaults	
Physics Preference	Mechanical
Element Order	Program Controlled

Element Size	0.1 m
Sizing	
Use Adaptive Sizing	Yes
Resolution	Default (2)
Mesh Defeaturing	Yes
Defeature Size	Default
Transition	Fast
Span Angle Center	Coarse
Initial Size Seed	Assembly
Bounding Box Diagonal	8.1104 m
Average Surface Area	0.25115 m^2
Minimum Edge Length	9.7193e-003 m
Quality	
Check Mesh Quality	Yes, Errors
Error Limits	Aggressive Mechanical
Target Element Quality	Default (5.e-002)
Smoothing	Medium
Mesh Metric	None
Inflation	
Use Automatic Inflation	None
Inflation Option	Smooth Transition
Transition Ratio	0.272
Maximum Layers	5
Growth Rate	1.2
Inflation Algorithm	Pre
Inflation Element Type	Wedges
View Advanced Options	No

Advanced	
Number of CPUs for Parallel Part Meshing	Program Controlled
Straight Sided Elements	No
Rigid Body Behavior	Dimensionally Reduced
Triangle Surface Mesher	Program Controlled
Topology Checking	Yes
Pinch Tolerance	Please Define
Generate Pinch on Refresh	No
Statistics	
Nodes	46762
Elements	26536
Show Detailed Statistics	No

Table 3.12 Mesh Controls

Object Name	*Body Sizing*	*MultiZone*
State	Fully Defined	
Scope		
Scoping Method	Geometry Selection	
Geometry	4 Bodies	2 Bodies
Definition		
Suppressed	No	
Type	Element Size	
Element Size	5.e-002 m	
Method		MultiZone

Decomposition Type		Program Controlled
Mapped/Swept Type		Hexa
Surface Mesh Method		Program Controlled
Free Mesh Type		Not Allowed
Element Order		Use Global Setting
Src/Trg Selection		Automatic
Src/Trg Scoping Method		Program Controlled
Source(s) and Target(s)		Program Controlled
Sweep Size Behavior		Sweep Element Size
Sweep Element Size		Default
Element Option		Solid
Advanced		
Defeature Size	Default	
Behavior	Soft	
Preserve Boundaries		Protected
Mesh Based Defeaturing		Off
Minimum Edge Length		1.9439e-002 m
Write ICEM CFD Files		No
Use Split Angle		No

3.5.6 Static Structural Analysis for Bending and Torsional Failure

The **Static Structural Analysis** module in ANSYS was used to conduct the load failure analysis under two distinct scenarios:

1) **Scenario 1**: Power input of 10,000 kW at 300 RPM.

Table 3.13: Loads

Object Name	*Fixed Support @ Shaft Bearings*	***Torsional Moment***	***Propeller Weight***	***Propeller Thrust***	*Moment 2*	*Moment 3*
State	Fully Defined				Suppressed	
Scope						
Scoping Method	Geometry Selection					
Geometry	2 Faces	26 Faces	3 Faces	28 Faces		3 Faces
Definition						
Type	Fixed Support	Moment	Force		Moment	
Suppressed	No				Yes	

Define By		Components		Vector	Components
Coordinate System		Global Coordinate System			Global Coordinate System
X Component		**-3.1847e+005 N·m (ramped)**	0. N (ramped)		0. N·m (ramped)
Y Component		0. N·m (ramped)	**-10791 N (ramped)**		0. N·m (ramped)
Z Component		0. N·m (ramped)	0. N (ramped)		-26978 N·m (ramped)
Behavior		Deformable			Deformable

Applied By			Surface Effect			
Magnitude				**48069 N (ramped)**		
Direction				Defined		
Advanced						
Pinball Region		All			All	

2) **Scenario 2**: Power input of 20,000 kW at 300 RPM.

The analysis was executed with the calculated torque values, propeller weight, and axial thrust applied. Key outputs from this phase were evaluated to assess the potential for bending and torsional failure.

Table 3.14: >Loads

Object Name	*Fixed Support*	***Torsional Moment***	***Propeller Weight***	***Propeller Thrust***	*Moment*	*Moment*

	@				*2*	*3*
	Shaft Bearings					
State	Fully Defined				Suppressed	
Scope						
Scoping Method	Geometry Selection					
Geometry	2 Faces	26 Faces	3 Faces	28 Faces		3 Faces
Definition						
Type	Fixed Support	Moment	Force		Moment	
Suppressed	No				Yes	

Define By		Components		Vector	Components
Coordinate System		Global Coordinate System			Global Coordinate System
X Component		**- 6.3694e+005 N·m (ramped)**	0. N (ramped)		0. N·m (ramped)
Y Component		0. N·m (ramped)	**-10791 N (ramped)**		0. N·m (ramped)
Z Component		0. N·m (ramped)	0. N (ramped)		-26978 N·m (ramped)
Behavior		Deformable			Deformable
Applied By			Surface Effect		
Magnitude				**48069 N (ramp**	

			ed)	
Directi on			Defined	
Advanced				
Pinb all Regi on		All		All

3.5.7 Results for Bending and Torsional Failure

The following outputs were obtained to evaluate the shaft's response to bending and torsional loads:

1. **Equivalent Von Mises Stress**: This result provided a comprehensive view of the stress distribution along the shaft, with high-stress areas indicating zones susceptible to yielding.

2. **Total Deformation**: The total deformation data revealed the overall bending or twisting displacement of the shaft under load.

3. **Directional Deformations**: Specific directional deformations (axial, radial) were analyzed to provide a clear understanding of the shaft's behavior under both bending and torsional stresses.

4. **Factor of Safety (FOS)**: The FOS calculations for bending and torsion provided a measure of the shaft's load-carrying capacity and its risk of failure under the applied loads.

3.5.8 Fatigue Failure Analysis Setup

The ANSYS fatigue analysis tools were employed to assess the shaft's response to cyclic loading conditions. The following key fatigue metrics were derived:

1. **Life**: This metric provided an estimate of the fatigue life of the shaft, in cycles, under cyclic loading conditions.
2. **Damage**: Cumulative damage was calculated to predict the shaft's endurance limit and lifespan.
3. **Safety Factor for Fatigue**: This safety factor indicated the shaft's ability to withstand fatigue under repeated loading conditions.
4. **Biaxiality Indication**: The biaxiality index was used to assess complex stress states, which could influence fatigue life.
5. **Fatigue Sensitivity**: Fatigue sensitivity analysis was conducted to identify high-risk areas for fatigue failure, informing potential design adjustments or reinforcements.

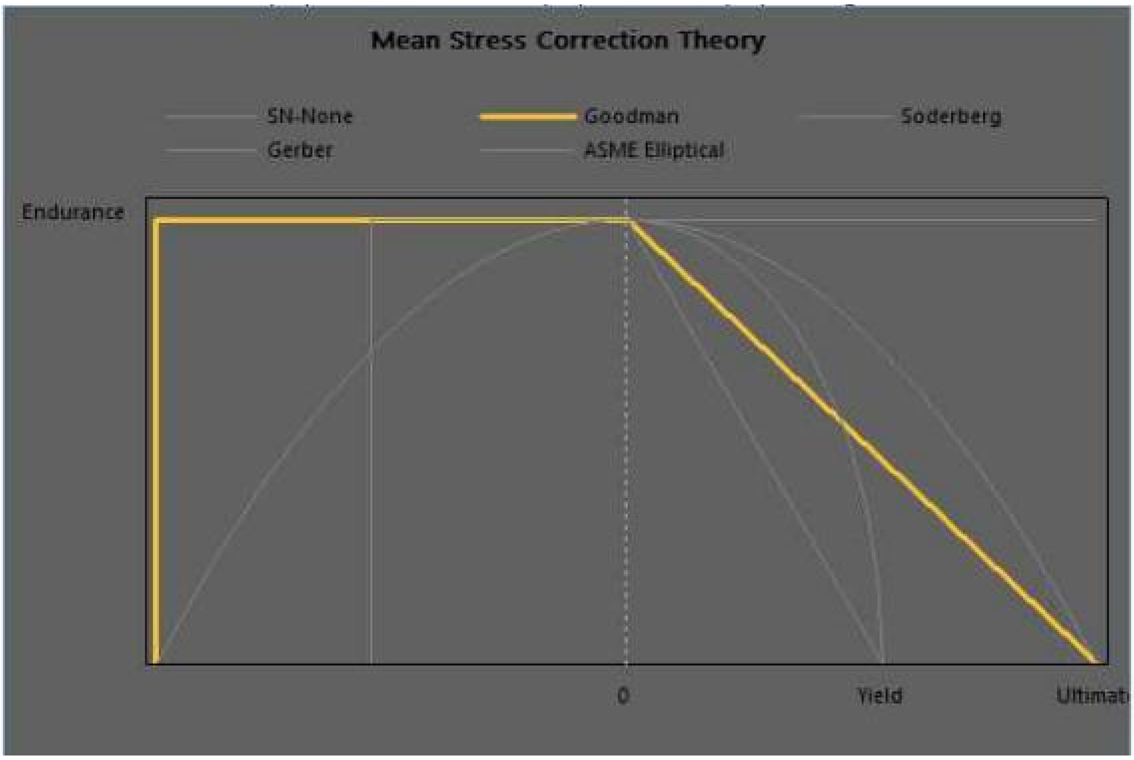

Figure 3.4: Mean Stress Correction Theory

3.6 Summary of Results and Recommendations

Results from the analysis provided valuable insights into the shaft's behavior under varying power conditions. The von Mises stress, deformation patterns, and fatigue life results showed distinct differences between the 10,000 kW and 20,000 kW scenarios, indicating the effects of increased power on the structural response of the shaft. Based on these findings, the propeller shaft's ability to withstand the specified loads was assessed. Recommendations for potential design modifications were suggested to improve durability and mitigate failure risks, particularly for scenarios involving high loading conditions

CHAPTER FOUR
RESULT AND DISCUSSION

4.1 Introduction

This chapter presents the results obtained from the simulation analyses conducted on the propeller shaft of the marine vessel, followed by a detailed discussion of the findings. The study focused on evaluating the shaft's structural integrity under different loading conditions and identifying potential failure modes, specifically bending/torsional failure and fatigue failure.

Using ANSYS, the shaft assembly was subjected to static structural analysis for two power scenarios: 10,000 kW and 20,000 kW, each at a rotational speed of 300 RPM. Applied forces included the calculated torque, axial thrust, and propeller weight. The results are reported in terms of equivalent (von Mises) stress, total deformation, directional deformation, and factor of safety for bending/torsional failure. For fatigue analysis, parameters such as fatigue life, damage, safety factor, biaxiality indication, and fatigue sensitivity were examined.

The discussions in this chapter analyze these results in relation to the theoretical expectations, examining how the applied loads influence the shaft's performance and highlighting any potential failure risks. By comparing the outcomes across different loading scenarios, valuable insights are provided into the shaft's operational limits, durability, and suitability for long-term use under varying conditions. This analysis is crucial for understanding the shaft's design

eliability and identifying areas that may require reinforcement to prevent mechanical failure.

4.2 Scenario 1: Power input of 10,000 kW at 300 RPM.

4.2.1 Torsional/Bending Failure

Table 4.1: Equivalent Stress

Time [s]	Minimum [Pa]	Maximum [Pa]	Average [Pa]
1.	1.8666e-002	1.2182e+008	5.9244e+006

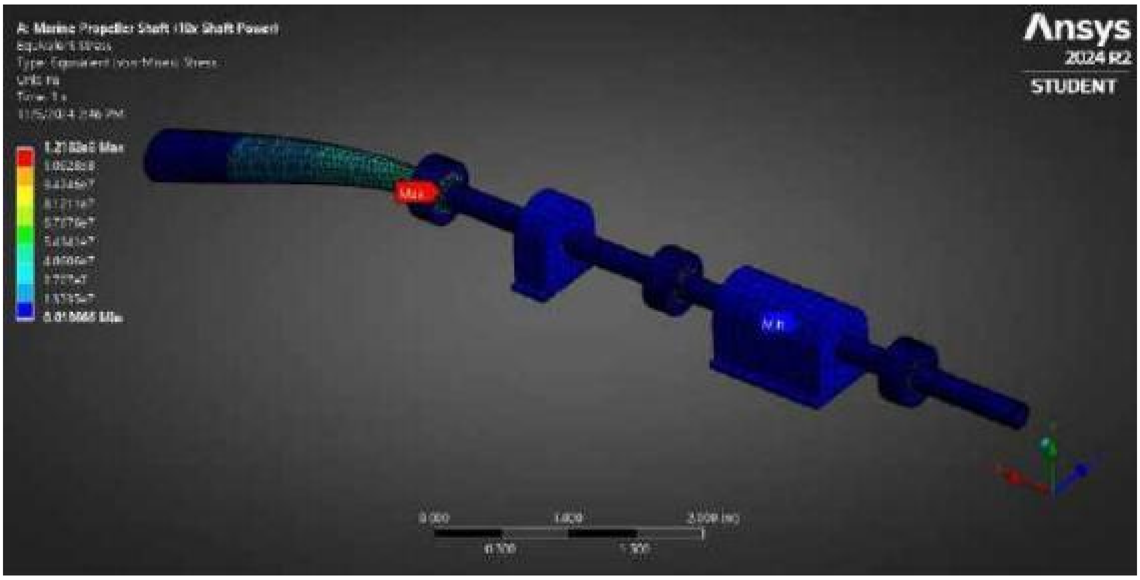

Figure 4.1: Contours of Equivalent Von Mises Stress

1. Equivalent Von Mises Stress Analysis

i. The maximum equivalent stress recorded is **1.2182 × 10^8 Pa (121.82 MPa)**.

ii. The high-stress concentration is observed near the shaft's critical transition points, with the highest occurring at the intermediate shaft- flange joint or neck.

iii. Of the three parts of the shaft the tail shaft experiences the highest average stress as it maintains direct contact with the propeller thereby receiving the brunt of the load developed.

iv. The maximum stress is below the yield strength of typical marine shaft materials (e.g., stainless steel, titanium alloys), indicating that the shaft can withstand the applied torque without immediate failure.

Table 4.2: Total Deformation

Time [s]	Minimum [m]	Maximum [m]	Average [m]
1.	0.	2.1243e-003	1.4778e-004

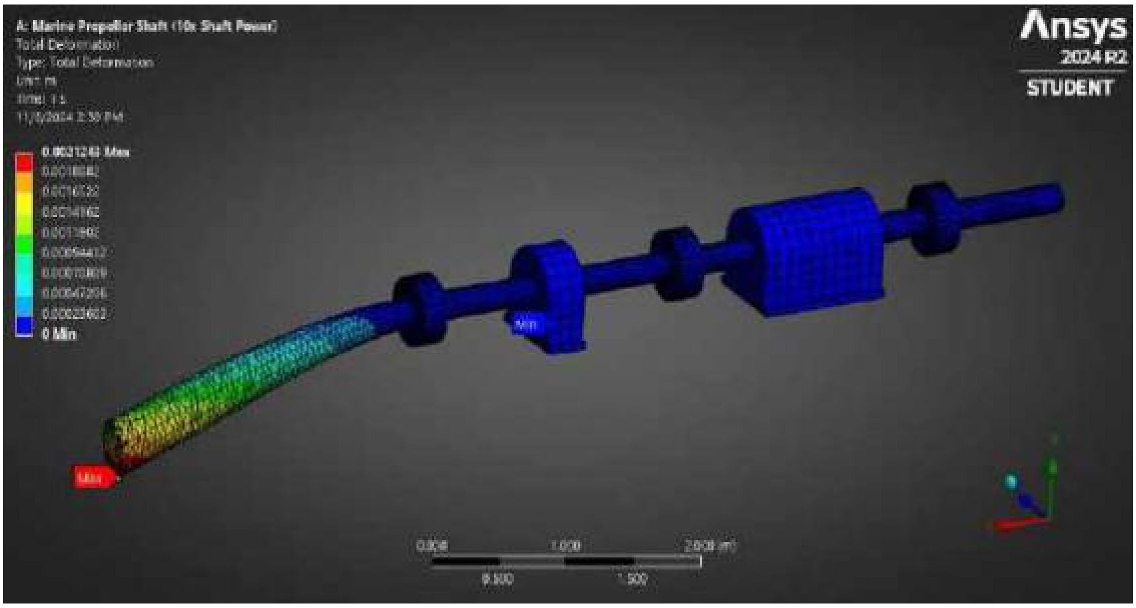

Figure 4.2: Contours of Total Deformation

4.2.2 Total Deformation Analysis

1) Maximum deformation observed: 2.1243×10^{-3} m (2.1243 mm).
2) The deformation is concentrated at the free end of the shaft, suggesting flexural bending effects due to torque transmission.
3) While deformation is within acceptable limits, excessive bending over time could lead to fatigue failure.

4) The tail/propeller shaft once again observes the highest average deformation due to its direct contact to the propeller and its direct reception of the load developed.

Table 4.3: X Axis - Directional Deformation

Time [s]	Minimum [m]	Maximum [m]	Average [m]
1.	-8.8093e-005	8.1935e-005	-7.8099e-007

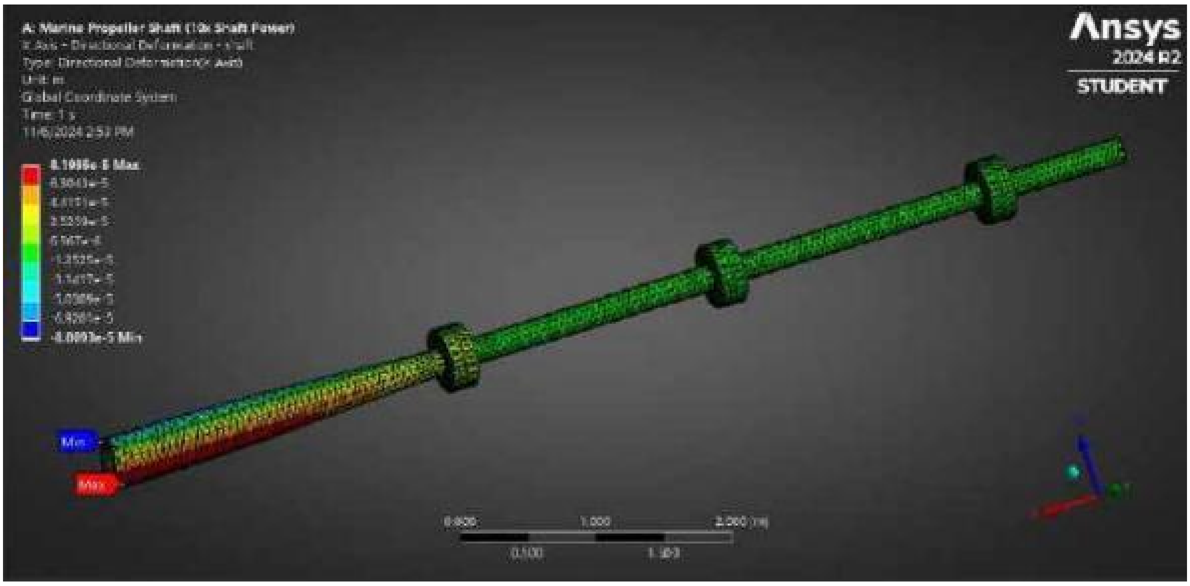

Figure 4.3: Contours of X-Axis Directional Deformation

Table 4.4: Y Axis - Directional Deformation

Time [s]	Minimum [m]	Maximum [m]	Average [m]
1.	-2.1241e-003	7.7745e-005	-1.4303e-004

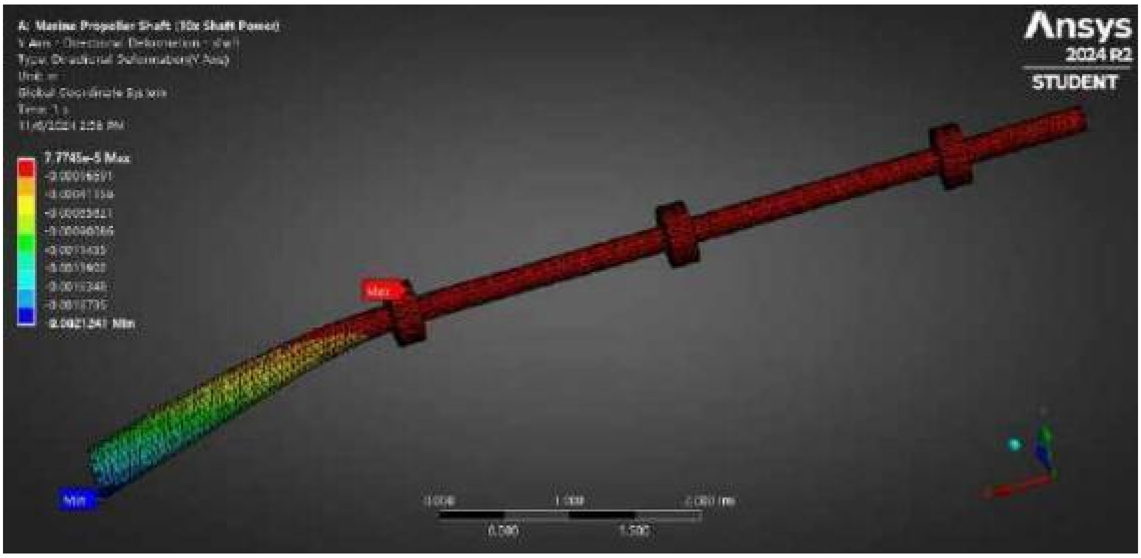

Figure 4.4:Contours of Y-Axis Directional Deformation

Table 4.5: Z Axis - Directional Deformation

Time [s]	Minimum [m]	Maximum [m]	Average [m]
1.	-4.7676e-004	4.2453e-004	-7.705e-006

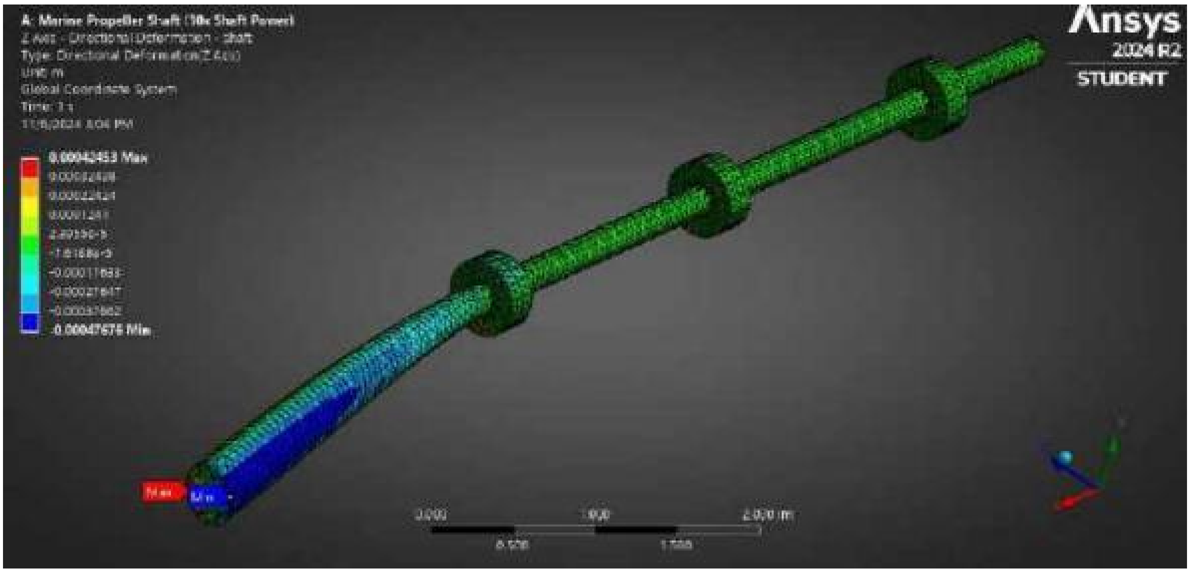

Figure 4.5: Contours of Z-Axis Directional Deformation

4.2.3 Directional Deformation Analysis

1) X-Axis: Maximum deviation: 8.1935×10^{-5} m (0.0819 mm). This is characterized by minimal lateral displacement, indicating minor axial misalignment.

2) Y-Axis: Maximum deviation: 7.7745×10^{-5} m (0.0777 mm). Characterized by light bending in the vertical plane, possibly due to gravitational effects.

3) Z-Axis: Maximum deviation: 4.2435×10^{-4} m (0.424 mm). Characterized by larger displacement, signifying torsional warping effects.

Table 4.6: Safety Factor

Time [s]	Minimum	Maximum	Average
1.	2.0523	15.	14.351

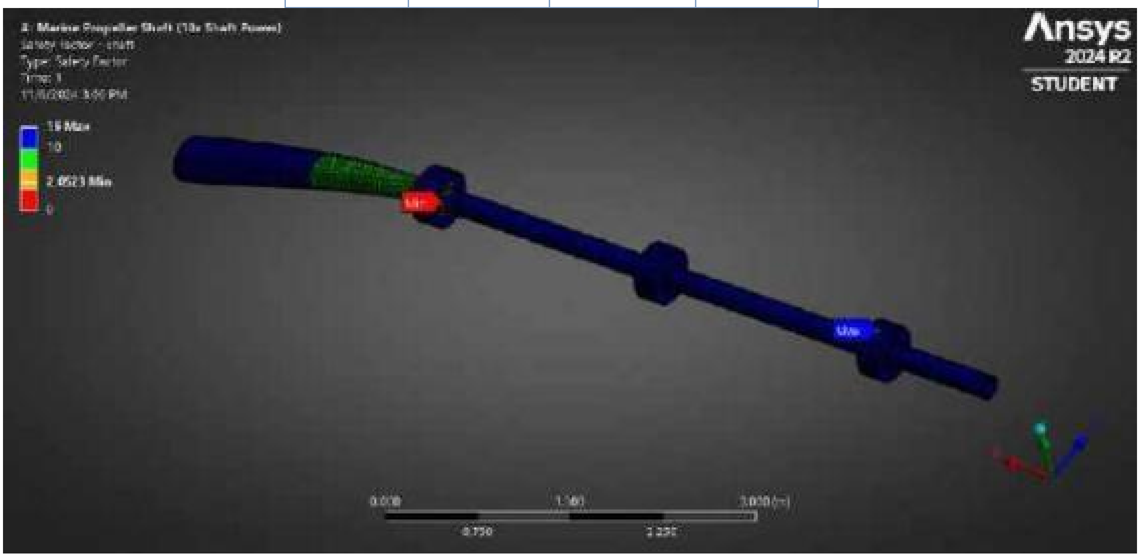

Figure 4.6: Contours of Factor of Safety

4.2.4 Factor of Safety (FOS) Analysis

i. The minimum safety factor is 2.0523, meaning the shaft has a structural margin of safety above failure limits.

ii. The maximum safety factor is 15, with an average of 14.351, indicating that most of the shaft remains in a low-stress state under operational conditions.

iii. A minimum FOS of 2.0523 suggests that the shaft can safely handle the given load, but optimization may be required to reduce stress concentration zones.

4.2.4.1 Fatigue Failure

Table 4.7: Life

Time [s]	Minimum	Maximum	Average
1.	1.847e+009	1.1574e+010	1.1569e+010

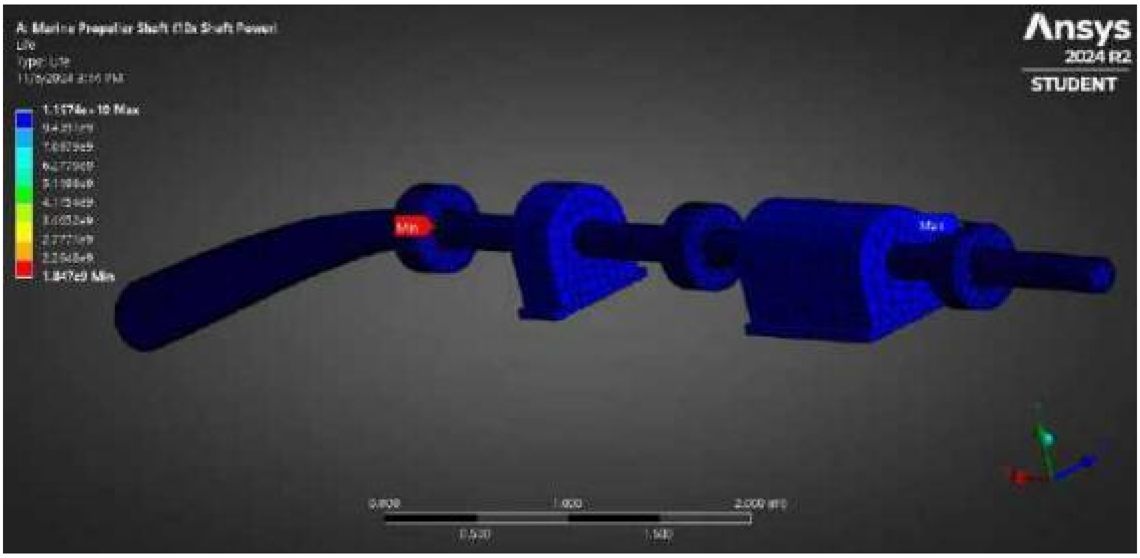

Figure 4.7: Fatigue Life Contour

4.2.5 Fatigue Life Evaluation

1) The fatigue life results indicate a minimum value of 1.847×10^{9} seconds and a maximum value of 1.1574×10^{10} seconds, with the minimum occurring at the intermediate shaft-flange joint or neck

2) The average fatigue life across the shaft is 1.1569×10^{10} seconds, suggesting that certain regions of the shaft experience significantly lower lifespans due to high stress concentrations.

3) The fatigue life contour plot shows that fatigue failure is most likely to occur at the intermediate shaft-flange joint or neck even though all other parts appear relatively safe.

Table 4.8: Damage

Time [s]	Minimum	Maximum	Average
1.	8.64e-002	0.54142	8.6499e-002

4.2.6 Damage Analysis

1) The fatigue damage assessment reveals a minimum value of 8.64×10^{-2} and a maximum value of 0.54142.
2) An average damage value of 8.6499×10^{-2} suggests that some regions of the shaft are approaching critical fatigue limits.
3) Higher damage values are concentrated in areas subjected to repeated cyclic loading and bending stresses.

Table 4.9: Safety Factor

Time [s]	Minimum	Maximum	Average
1.	1.1895	15.	13.62

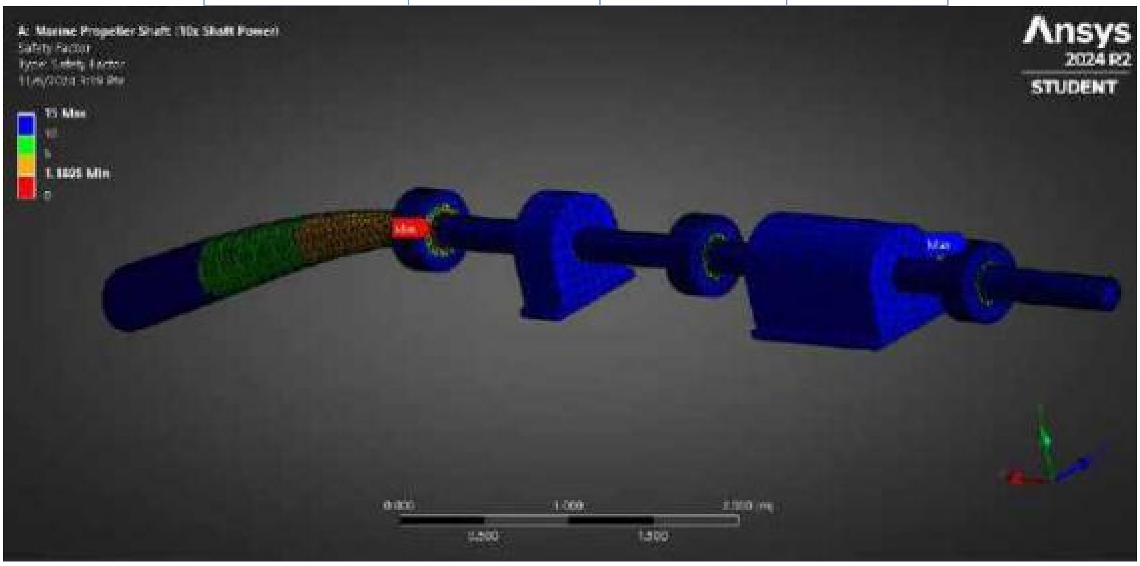

Figure 4.8: Fatigue safety factor contour

4.2.6 Safety Factor Evaluation

1. The minimum safety factor recorded is 1.1895, while the maximum is 15, with an average of 13.62.
2. From the fatigue safety factor contour it can be identified that areas around the tail shaft are approaching the failure threshold with the closest being at the intermediate shaft-flange joint or neck.
3. While the minimum factor of safety for torsional/bending stresses was at 2.0523, the minimum factor of safety for fatigue loads was significantly lower at almost 1.2. This indicates that while the shaft may be safe from torsional failure it has a greater risk of fatigue failure due to repeated loading cycles.

4.2.7 Biaxiality Indication

Table 4.10: Bixality Indication

Time [s]	Minimum	Maximum	Average
1.	-1.	0.91874	-0.47698

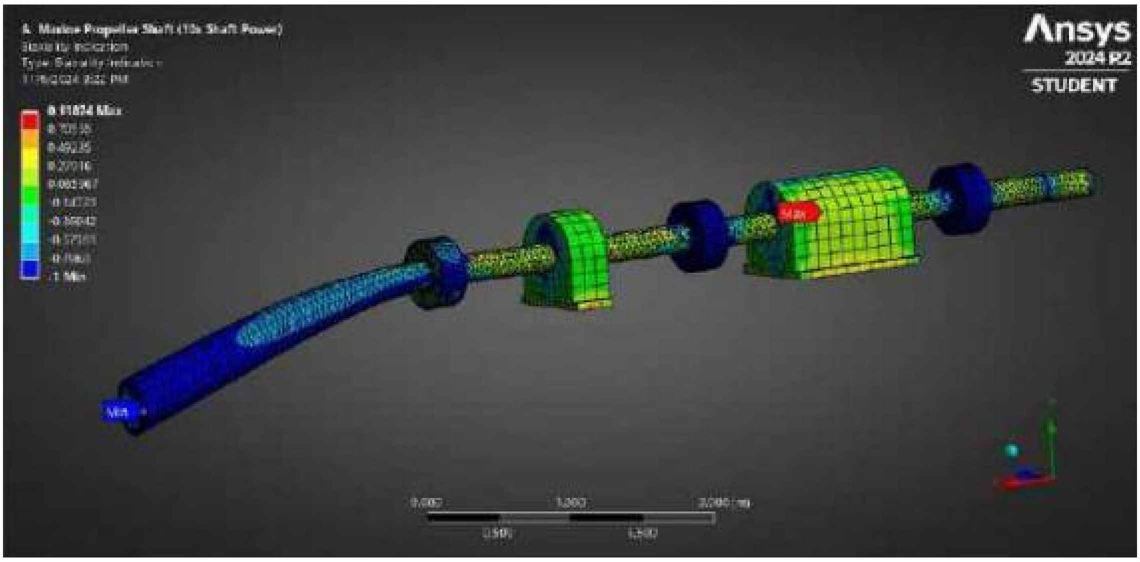

Figure 4.9: Biaxiality indication contour

1) The biaxiality factor ranges from -1 to 0.91874, with an average value of -0.47698.
2) This parameter helps determine the nature of stress distribution and the likelihood of multi-axial fatigue failure.
3) The biaxiality indication contour highlights areas where stress interactions could accelerate fatigue crack initiation.

4.2.8 Fatigue Sensitivity

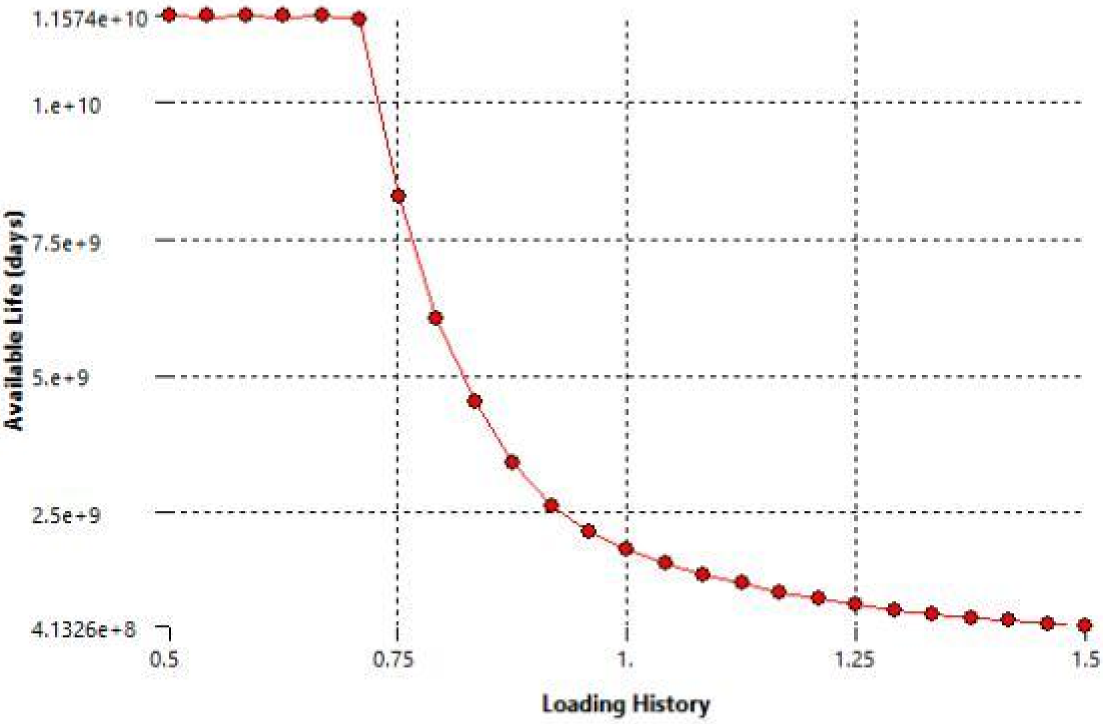

Figure 4.10: Loading History

4.2.8.1 Fatigue Sensitivity Analysis

1) The fatigue sensitivity graph illustrates how the available life of the shaft decreases as the loading cycles progress. In this case it calculated the change in fatigue life as loading is increased or decreased by 25% and 50% respectively.

2) At 10,000KW the minimum fatigue life stood at 1.85e+9 seconds, this is represented in the Loading History axis of the graph as 1. Increasing this load by 25% and 50% shows a steady but slow decrease in fatigue life, however decreasing by 25% shows a sharp increase in fatigue life up to a certain threshold where the fatigue life remains

faily constant even when loading is further decreased by 50%. The shaft is said to have reached infinite life at this point.

4.3 Scenario 2: Power input of 20,000 kW at 300 RPM.

4.3.1 Torsional/Bending Failure

Table 4.11: Equivalent Stress

Time [s]	Minimum [Pa]	Maximum [Pa]	Average [Pa]
1.	1.44e-002	2.4108e+008	1.1702e+007

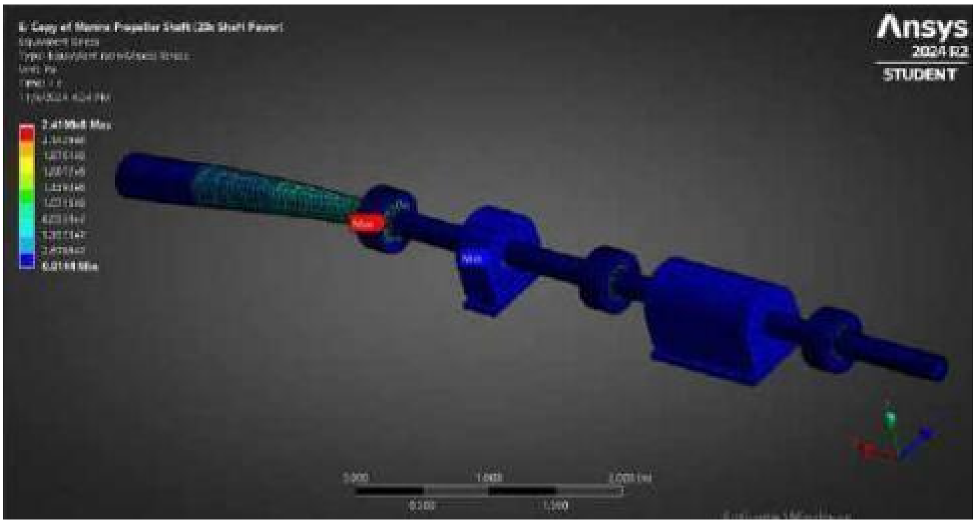

Figure 4.11: Contours of equivalent von-mises stress

1) A The maximum stress (241.08 MPa) has gotten significantly higher and is approaching the yield strength of the shaft material.
2) The average stress (11.70 MPa) is much lower than the peak value, indicating that while most of the shaft is experiencing relatively low stress, certain localized regions are under extreme load.
3) The tail shaft yet again experiences the highest average stress compared to the intermediate and thrust shaft
4) The highest equivalent stress occurs in the intermediate shaft-flange joint. This joint is the most susceptible to instantaneous bending/torsional failure.

Table 4.12: Total Deformation

Time [s]	Minimum [m]	Maximum [m]	Average [m]
1.	0.	2.5747e-003	1.7609e-004

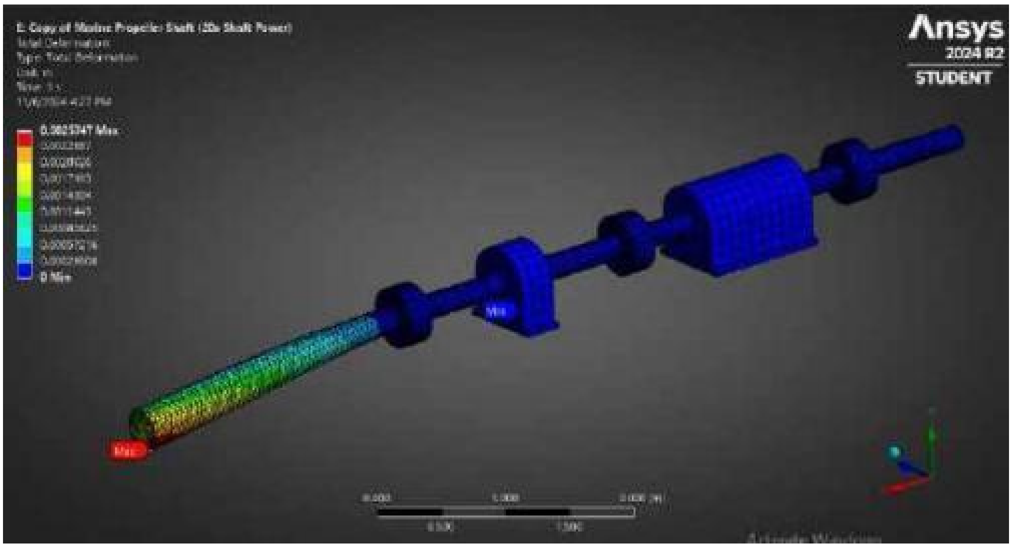

Figure 4.12: Contours of Total Deformation

4.3.2 Total Deformation Analysis

1) A maximum deformation of 2.57 mm suggests that the shaft experiences noticeable deflection under operational conditions. This is moderate but could lead to misalignment issues in mechanical components such as bearings and couplings.

2) The average deformation (0.176 mm) is relatively low, meaning that most of the shaft experiences minor displacement, but the critical regions (like the tail shaft) have much higher displacement, due to their close proximity to the loading points.

Table 4.13: X Axis - Directional Deformation

Time [s]	Minimum [m]	Maximum [m]	Average [m]
1.	-8.806e-005	8.1907e-005	-7.8085e-007

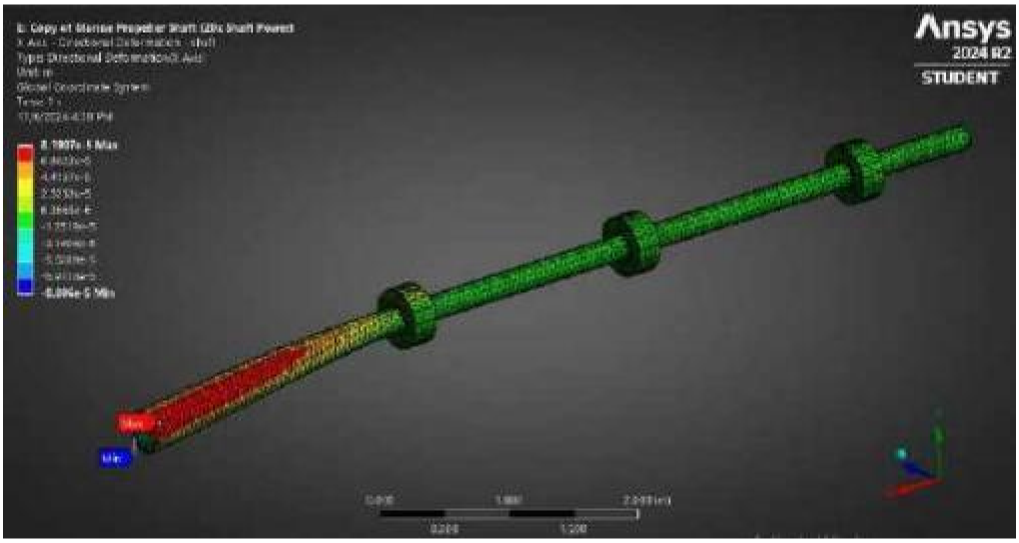

Figure 4.13: Contours of x-axis directional deformation

Table 4.14: Y Axis - Directional Deformation

Time [s]	Minimum [m]	Maximum [m]	Average [m]
1.	-2.5742e-003	2.2441e-004	-1.4328e-004

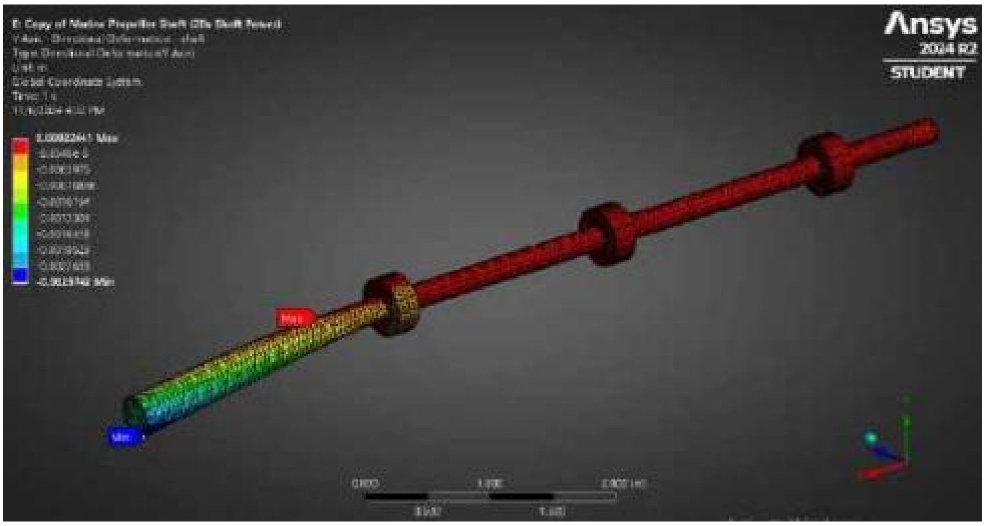

Figure 4.14: Contours of y-axis directional deformation

Table 4.15: Z Axis - Directional Deformation

Time [s]	Minimum [m]	Maximum [m]	Average [m]
1.	-9.5348e-004	8.4909e-004	-1.5409e-005

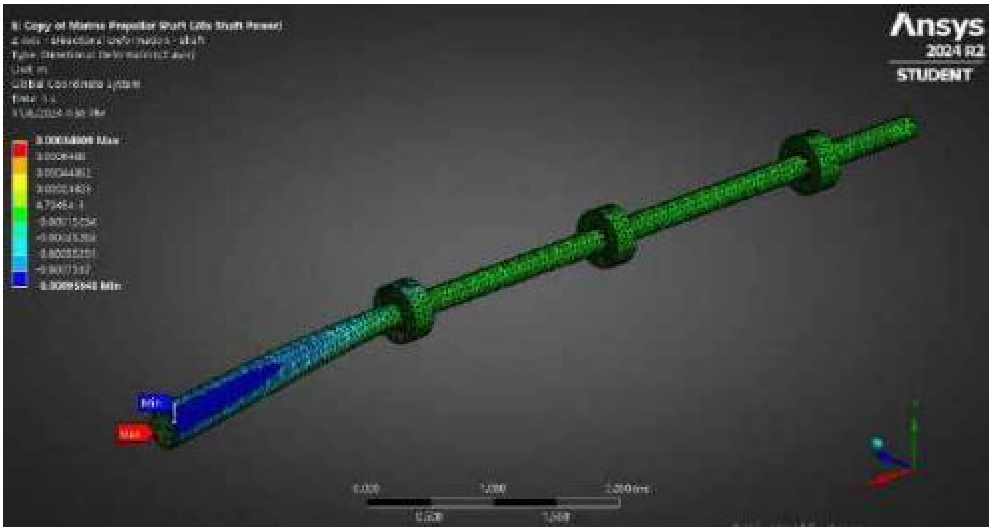

Figure 4.15: Contours of z-axis directional deformation

4.3.3 Directional Deformation Analysis

1) The X-axis deformation (0.0819 mm) is minimal, meaning that there is very little side-to-side displacement, which is good for maintaining radial stability.

2) The Y-axis deformation (0.2244 mm) indicates that there is some bending in the vertical plane, but it is within a reasonable range.

3) The Z-axis deformation (0.849 mm) is the highest, suggesting that the shaft experiences the most displacement along its length, likely due to torque- induced twisting.

4) The high Z-axis deformation indicates axial elongation, which may cause misalignment in coupling with other rotating components.
5) The bending effect along the Y-axis could cause imbalance, potentially leading to vibrations and reducing system efficiency.
6) X-axis stability is a positive outcome, meaning the shaft maintains good lateral alignment

Table 4 16: Safety Factor

Time	Minimum	Maximum	Average
1.	1.037	15.	13.412

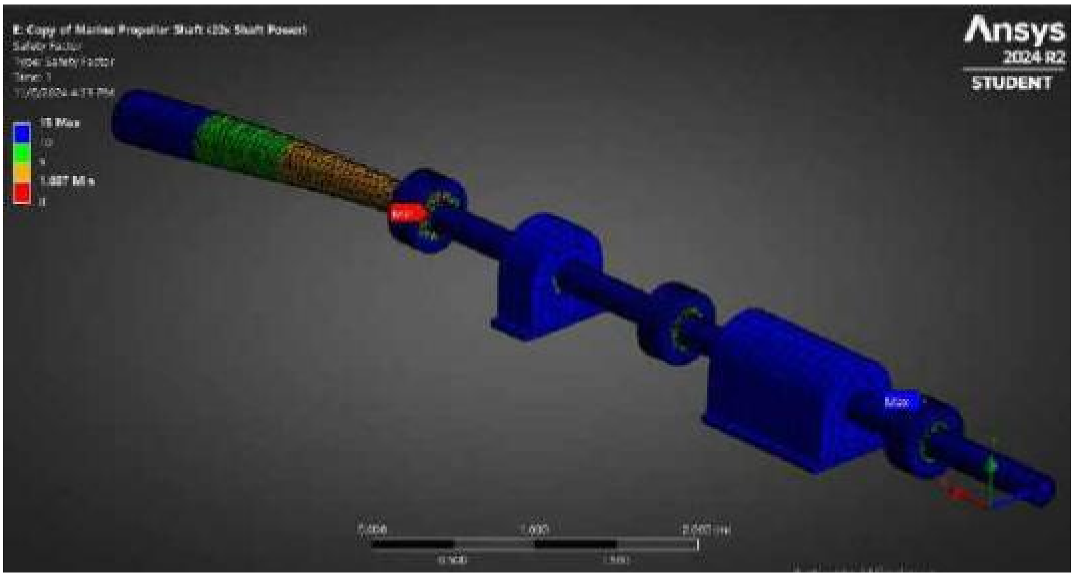

Figure 4.16: Contours of Safety Factor

4.3.4 Factor of Safety (FOS) Analysis

1) As expected, factor of safety dropped considerably with the increased load

2) The minimum safety factor of 1.037 is concerning, as it is very close to structural failure in certain areas.

a. Fatigue Failure

Table 4 17: Fatigue Tool > Life

Time [s]	Minimum	Maximum	Average
1.	1.5657e+008	1.1574e+010	1.1508e+010

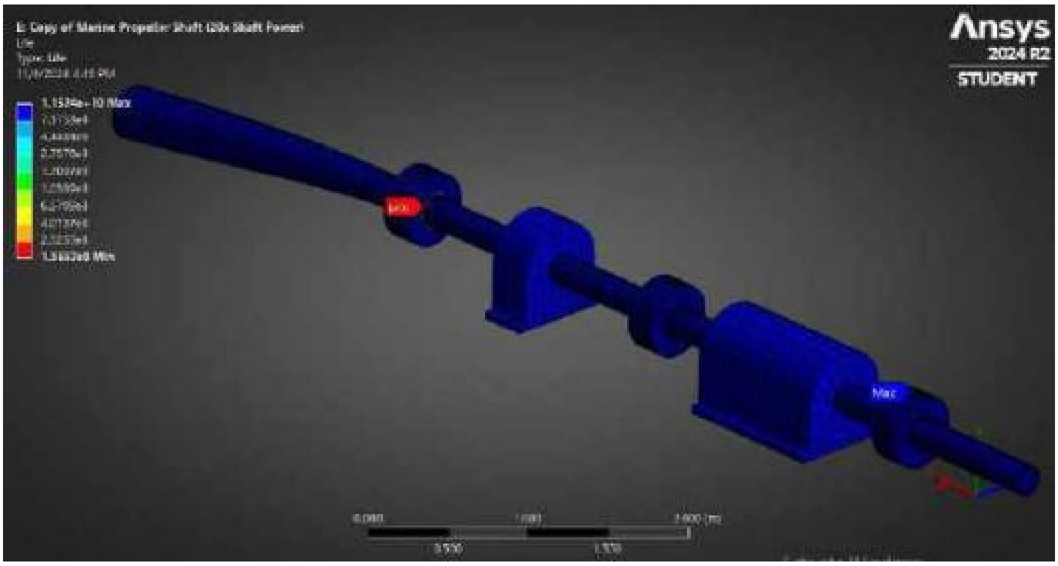

Figure 4.17: Contours of Fatigue Life

4.3.5 Fatigue Life Analysis

The minimum fatigue life is observed at critical stress concentration areas mostly in the intermediate shaft-flange joint. This suggests that localized stress raisers could lead to early fatigue failure in these regions. However, the average fatigue life is significantly higher, indicating that most of the shaft is structurally sound and capable of enduring cyclic loads over an extended period.

Table 4.18: Fatigue Tool > Damage

Time [s]	Minimum	Maximum	Average
1.	8.64e-002	6.3871	9.2083e-002

4.3.6 Fatigue Damage Analysis

The fatigue damage values indicate how much of the component's fatigue life has been consumed under the given loading conditions. The maximum damage value of 6.3871 suggests that certain regions of the shaft exceed their fatigue limit and may experience premature failure. The average damage remains relatively low, meaning most of the shaft structure remains within acceptable operational limits.

Table 4.19: Fatigue Tool > Safety Factor

Time [s]	Minimum	Maximum	Average
1.	0.60106	15.	12.486

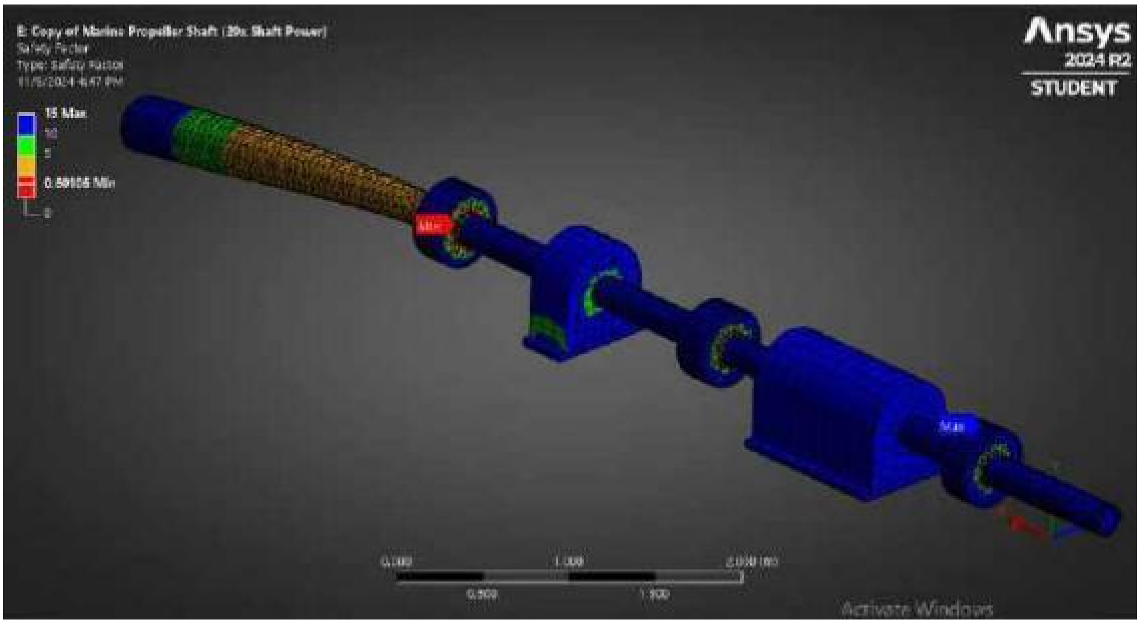

Figure 4.18: Contours of fatigue safety factor

4.3.7 Fatigue Safety Factor Analysis

A minimum safety factor of 0.60106 indicates that some areas of the shaft are experiencing high stress cycles, making them prone to fatigue failure if not reinforced. These areas consist mainly of the intermediate shaft-flange joint along with some parts of the tail shaft and the shaft-flange joints of the thrust shaft. Since the ideal fatigue safety factor should be greater than 1, regions with values below this threshold may need design optimization or material strengthening. The high average safety factor of 12.486 confirms that most of the shaft remains structurally stable under operational loads.

Table 4.20: Fatigue Tool > Biaxiality Indication

Time [s]	Minimum	Maximum	Average
1.	-0.99999	0.81964	-0.52777

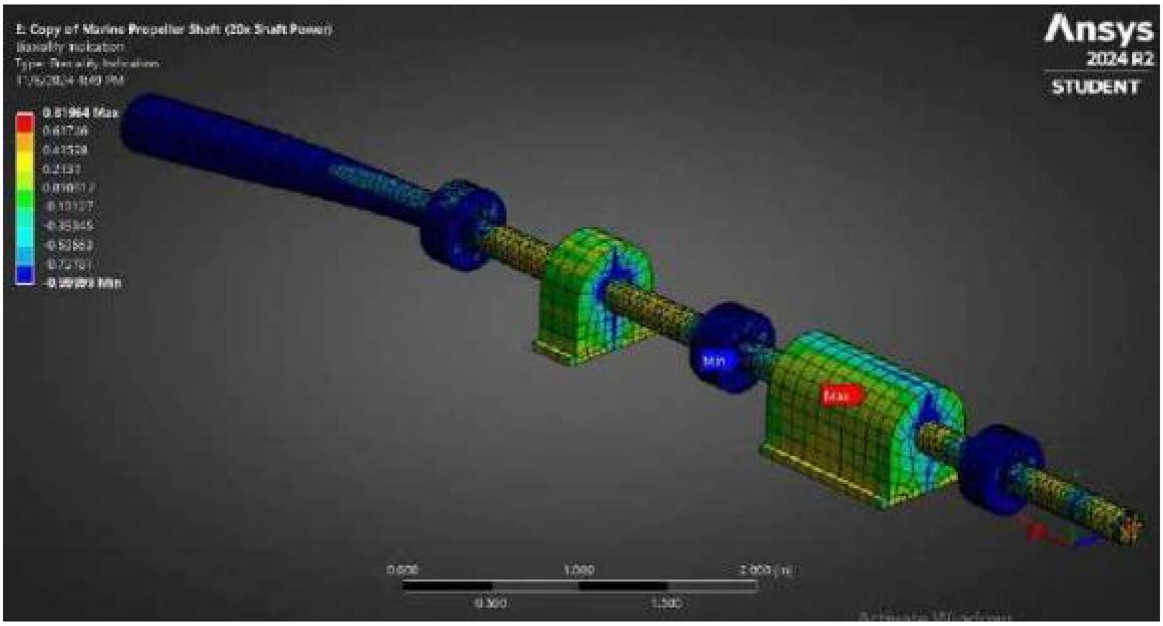

Figure 4.19: Contours of Biaxiality Indication

4.3.8 Biaxiality Indication Analysis

The biaxiality indication values describe the stress state at different points on the shaft. A value of -1 represents pure uniaxial stress, 0 represents pure shear stress, and +1 represents fully biaxial stress. The results indicate that most regions experience a dominant uniaxial stress state, which is typical for rotating shafts subjected to bending and torsional loads. However, some areas show mixed stress states, which may contribute to localized fatigue failure.

4.3.9 Fatigue Sensitivity Analysis

The fatigue sensitivity chart suggests a strong inverse relationship between the fatigue life and applied cyclic loading. Higher load amplitudes result in a rapid decrease in fatigue life, emphasizing the importance of controlling peak stress levels in high-power scenarios. The curve indicates that minor reductions in applied stress could significantly extend the shaft's operational life. For example, reducing the applied load by 50% leads to a 7.5 times increase in fatigue life.

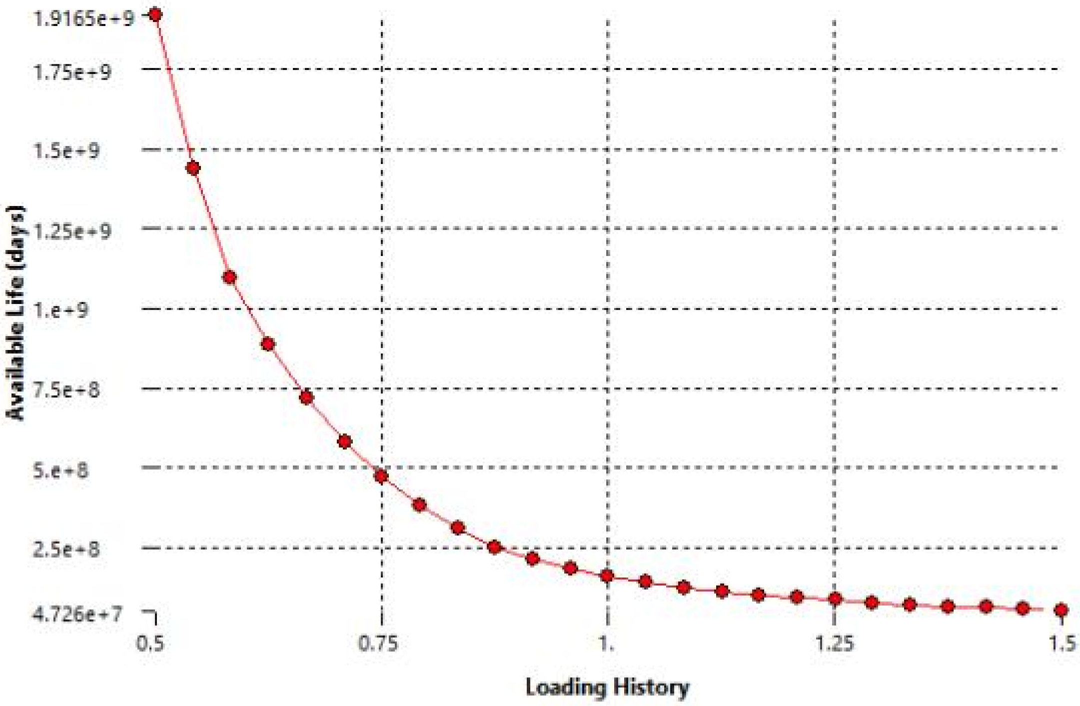

Figure 4.20: Loading History

CHAPTER FIVE

CONCLUSION AND RECOMMENDATIONS

5.1 CONCLUSION

This study examined the failure mechanisms of the propeller shaft in a twin- screw passenger ferry operating under cyclic loading conditions. The analysis focused on the effects of fatigue, bending, and torsional stresses at varying power levels. Using detailed simulations in Ansys Mechanical, the study identified critical stress points and potential failure zones, providing insights into the structural integrity of the shaft.

The results indicated that the intermediate shaft-flange joint experienced the highest equivalent von Mises stress, making it the most vulnerable location for fatigue and bending/torsional failure. Under a power input of 10,000 kW, the shaft remained within safe operational limits, demonstrating a minimum safety factor of 2.0523 for torsional and bending failure, while fatigue failure showed a lower safety factor of 1.1895. Despite remaining within acceptable safety margins, stress concentration zones were observed, suggesting the need for design optimization to enhance long-term reliability.

When the power was increased to 20,000 kW, the maximum stress rose significantly to 241.08 MPa, approaching the yield strength of commonly used marine shaft materials. This substantial increase in stress levels heightened the risk of structural failure, indicating that the shaft design may not be sufficiently robust to handle such high power levels without modifications. Further analysis of fatigue life revealed that areas exposed to repeated cyclic loading, particularly the tail shaft and flange joint, were more susceptible to long-term degradation. The study also highlighted how multi-axial stress interactions contributed to fatigue crack initiation. Biaxiality indication and fatigue sensitivity analysis demonstrated that these interactions accelerated the onset of cracks, ultimately compromising the durability of the shaft over time.

Overall, the findings suggest that while the shaft is structurally capable of operating at 10,000 kW with minor risks, operating at 20,000 kW considerably increases the likelihood of failure. To ensure safe and reliable performance at higher power levels, design improvements such as stress redistribution, material enhancements, or structural reinforcements would be necessary.

5.2 RECOMMENDATIONS

Based on the study's findings, the following recommendations are proposed to enhance the robustness and operational safety of marine propeller shafts:

1. Material Optimization

The shaft can be made of high-strength, fatigue-resistant alloys such as duplex stainless steel or titanium alloys to improve fatigue life and resistance to cyclic loading.

(i) Marine propeller shafts are subjected to continuous cyclic loading, which can lead to fatigue failure over time. High- strength alloys like duplex stainless steel offer superior mechanical properties, including higher tensile strength and enhanced corrosion resistance, making them ideal for withstanding harsh marine environments. Titanium alloys, although more expensive, provide excellent strength-to-weight ratios, superior fatigue resistance, and exceptional corrosion resistance, particularly in seawater applications.

(ii) Selecting the right alloy composition can significantly enhance the durability of the shaft, reducing maintenance costs and extending service life.

(iii) The surface can be treated with shot peening or nitriding to reduce surface stresses and delay fatigue crack initiation.

(iv) Surface treatments play a crucial role in improving the fatigue resistance of the propeller shaft. Shot peening involves bombarding the surface with small spherical media to introduce beneficial compressive residual stresses, which help to counteract the tensile stresses that cause fatigue cracks. This technique improves surface hardness and significantly extends the fatigue life of the shaft.

(v) Nitriding is another effective surface treatment where nitrogen is diffused into the surface of the metal to form a hardened outer layer. This process enhances wear resistance and reduces susceptibility to surface-initiated fatigue cracks, particularly in high-load regions.

Implementing these surface treatments can mitigate early fatigue failure, ensuring long-term reliability.

2. Geometrical and Structural Modifications

The intermediate shaft-flange joint can be reinforced with fillets or a tapered transition to reduce stress concentration.

The shaft-flange joint is a critical region where high stress concentrations often lead to fatigue failure. Sharp transitions between the shaft and the flange create localized stress peaks, which accelerate crack formation.

By introducing fillets (curved transitions) at the joint, the stress concentration can be reduced, allowing for a more uniform distribution of loads. A tapered transition further improves load transfer, reducing sudden changes in stress and minimizing the risk of failure.

The shaft diameter can be increased at critical transition points to distribute stress more evenly and mitigate localized overloading. Stress distribution across the shaft is influenced by variations in diameter, particularly at transition points. Increasing the shaft diameter at areas where stress concentration is highest (e.g., near flanges, bearings, and coupling interfaces) can lower the stress intensity by spreading loads over a larger cross-sectional area.

This modification reduces localized overloading, improving the shaft's structural integrity and resistance to fatigue-induced fractures.

3. Load and Power Optimization

The operational power can be limited to below 20,000 kW or controlled adaptively to minimize excessive loading during peak operations.

Excessive power transmission through the propeller shaft increases torsional stress and accelerates wear and tear. Implementing an operational limit of 20,000 kW or lower helps prevent excessive loading, reducing mechanical fatigue and thermal buildup.

Adaptive power control systems, such as variable frequency drives (VFDs) or automated load monitoring, can be integrated to dynamically adjust power output based on real-time operational conditions. This optimization reduces unnecessary stress and prolongs the shaft's lifespan.

The loading conditions can be monitored regularly to implement predictive maintenance based on operational data.

A predictive maintenance strategy leverages data from onboard sensors to assess the condition of the shaft in real time. Parameters such as torque, vibration levels, and temperature can be continuously monitored to detect early signs of mechanical wear or fatigue.

Advanced machine learning algorithms and AI-driven diagnostics can analyze these data trends to predict potential failures before they occur, allowing for timely maintenance and reducing the risk of unexpected breakdowns.

4. **Fatigue Life Enhancement Strategies**

The maintenance intervals can be made stricter, particularly for high-stress regions such as the intermediate shaft-flange joint and tail shaft, to extend fatigue life.

Regular maintenance is critical for ensuring the longevity of marine propeller shafts. High-stress regions, especially the intermediate shaft-flange joint and tail shaft, experience the most significant fatigue loads and require frequent inspections.

Shortening maintenance intervals (e.g., from every 10,000 hours to every 7,000 hours of operation) can help detect and address potential issues before they escalate into major failures.

Implementing non-destructive testing (NDT) methods, such as ultrasonic inspection and dye penetrant testing, allows for early crack detection without dismantling the entire system.

The shaft can be equipped with real-time condition monitoring systems to track stress accumulation and fatigue damage, allowing for proactive interventions.

Real-time condition monitoring involves using strain gauges, vibration sensors, and thermal cameras to continuously assess the structural integrity of the shaft.

These systems provide immediate feedback on stress accumulation, enabling ship operators to take corrective action before significant damage occurs.

Proactive interventions, such as redistributing loads, adjusting propulsion speed, or scheduling preventive maintenance, can be implemented based on live data, ensuring optimal performance and reliability. By implementing these recommendations, the robustness and operational safety of marine propeller shafts can be significantly improved. Material advancements, structural modifications, optimized power control, enhanced fatigue management, and rigorous experimental validation will collectively contribute to increased reliability, reduced maintenance costs, and extended service life of propulsion systems in marine vessels.

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