# ENGINEERING CASE STUDY

## Project Type

Academic Group Project (Undergraduate Level) - 2020-II

## Project Description

This case study documents the mechanical design and validation of a power transmission system for a 5-kW horizontal-axis wind turbine.
The work focuses on shaft sizing, fatigue analysis, transmission layout, and motion studies, integrating classical mechanical design methods with CAD-based simulation.
The objective of this study was to ensure structural integrity, fatigue resistance, and reliable power transmission under realistic operating conditions.

## My Role and Contributions

### Lead Mechanical Design and Calculations

- Dimensioning of the main shaft and transmission shafts
- Fatigue analysis using: von Mises, Tresca, Goodman, Gerber criteria
- Load cases definition and stress evaluation
- Design and validation of the planetary gear transmission concept
- Motion study of blade system
- Interpretation of simulation results and engineering validation
- Development of automated calculation spreadsheets for iterative shaft sizing, load cases, and fatigue verification

Other team members contributed to component selection (bearings, keys) and complementary documentation under the same academic project.

## Tools and Methods

- Analytical calculations (Excel-based engineering spreadsheets)
- CAD modeling and static structural analysis
- Motion studies for blade behavior
- Classical mechanical design standards and fatigue criteria

## Academic Context

Developed as part of an undergraduate mechanical engineering course, under faculty supervision.

## Purpose of Inclusion

This case study is presented as evidence of engineering methodology, analytical rigor, and design validation skills, relevant to mechanical product development and system-level design.

## Use of AI-Assisted Documentation

This report was authored by the undersigned and documents previously developed engineering work. Artificial intelligence tools were used solely to assist in structuring, technical writing, and clarity improvement. All engineering calculations, assumptions, and design decisions were originally developed by the author using spreadsheet-based analytical models, which constitute the primary source of the results presented in this document. The use of AI did not replace engineering judgment, original analysis, or authorship of the work.

# Mechanical Design Case Study

## 5 kW Wind Turbine Power Transmission System

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## 1. Introduction

This document presents a mechanical engineering case study focused on the design and validation of a power transmission system for a 5-kW wind turbine. The objective is to demonstrate a structured engineering workflow, from definition of operating conditions to analytical sizing, component selection, and simulation-based validation.

The system includes: - A blade system subjected to aerodynamic loads - A blade shaft - A planetary gear transmission stage - A generator shaft - Rolling bearings and keys

The study prioritizes engineering reasoning, conservative assumptions, and execution-ready design decisions rather than exhaustive derivations.

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## 2. Case Study Definition

### 2.1 System Description

This case study focuses on the mechanical design of a power transmission system for a 5-kW wind turbine. The system includes a planetary gearbox coupling the blade shaft to the generator shaft, with each shaft supported by rolling bearings. Design parameters were defined based on operating conditions and environmental constraints.

### 2.2 Operating Conditions

*Table 1 System Operating Conditions*

| Parameter | Value |
|---|---|
| Rated power | 5 kW |
| Maximum input speed | 110 rpm |
| Minimum output speed | 550 rpm |
| Maximum operating temperature | 26 °C |

### 2.3 Blade System Parameters

*Table 2 Blade System Parameters*

| Parameter | Value |
|---|---|
| Blade frontal area | ~14.5 m$^2$ |
| Average wind speed | ~12.5 m/s |
| Air density | ~1.225 kg/m$^3$ |
| Blade mass | ~11 kg |

These parameters were used as input data for aerodynamic load estimation and subsequent mechanical component sizing.

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## 3. Blade System – Aerodynamic Load Estimation

Aerodynamic loads acting on the blades were estimated to determine the forces and torque transmitted to the first shaft of the drivetrain. A simplified drag-based approach was used to define conservative input loads for mechanical design.

*Table 3 Blade Aerodynamic Load Summary*

| Parameter | Value |
|---|---|
| Resulting drag force | ~14 N |

*Table 4 Loads Transmitted to Blade Shaft*

| Load type | Value |
|---|---|
| Blade weight | ~108 N |
| Axial load | ~41 N |
| Input torque | ~45 N·m |

These loads were used as input conditions for the blade shaft and transmission system design.

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## 4. Blade Shaft – Static and Fatigue Design

The blade shaft was designed considering static and fatigue loading, including the presence of stress concentrators such as keyways and geometric transitions. Classical failure criteria were applied to ensure structural integrity under operating conditions.

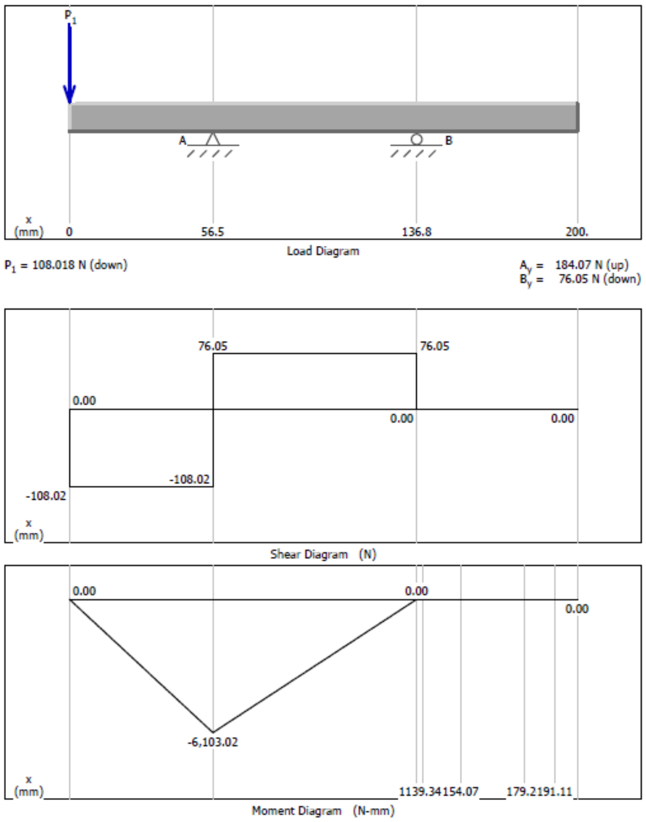


*Figure 1 Shear force and bending moment diagram for the blade shaft*

*Table 5 Blade Shaft Design Summary*

| Criterion | Required Diameter |
|---|---|
| Static (Von Mises / Tresca) | ~10 mm |
| Fatigue (Goodman) | ~10 mm |
| Fatigue (Gerber) | ~9.7 mm |
| **Final selected diameter** | **11 mm** |

The final diameter was selected based on the most conservative fatigue criterion, ensuring a minimum safety factor of 1.5.

## 5. Planetary Gear System – Kinematic and Load Analysis

A planetary gear system was designed to transmit power from the rotor to the generator, providing the required speed increase while maintaining compact geometry and load distribution. Kinematic relationships and transmitted forces were evaluated to ensure proper gear sizing.

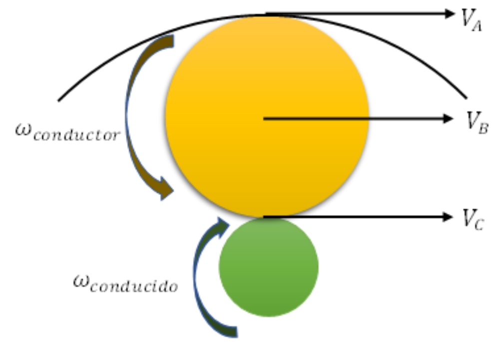


*Figure 2 Planetary gear layout showing sun, planets, ring gear, and direction of motion*

*Table 6 Planetary Gear Geometry*

| Element | Teeth (Z) | Module (m) | Pitch Diameter |
|---|---|---|---|
| Sun gear | 28 | 4 mm | 112 mm |
| Planet gear | 18 | 4 mm | 72 mm |
| Ring gear | 74 | 4 mm | 296 mm |

*Table 7 Transmitted Loads in Planetary Gear System*

| Parameter | Value |
|---|---|
| Transmitted torque | ~170 N·m |
| Tangential force | ~4.7 kN |
| Normal force | ~1.7 kN |

The resulting forces were used as input loads for the generator shaft and bearing design.

## 6. Generator Shaft – Static and Fatigue Design

The generator shaft was designed considering high torsional loads associated with continuous power transmission. A higher safety factor was selected due to the critical role of this component.

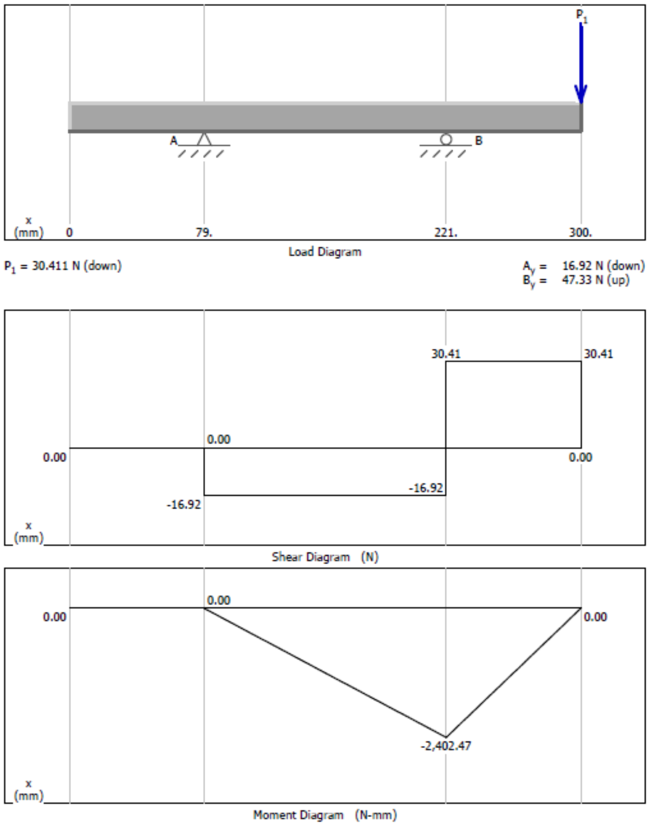


*Figure 3 Shear force and bending moment diagram for the generator shaft*

*Table 8 Generator Shaft Design Summary*

| Criterion | Required Diameter |
|---|---|
| Static (Von Mises / Tresca) | ~18 mm |
| Fatigue (Goodman) | ~25 mm |
| Fatigue (Gerber) | ~23 mm |
| **Final selected diameter** | **25 mm** |

The final diameter was selected based on the most conservative fatigue criterion, ensuring a safety factor of 2.5.

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## 7. Bearing Selection

Rolling bearings were selected based on shaft speed, applied loads, and required service life. Manufacturer data and standard bearing life criteria were used to determine a suitable commercial reference.

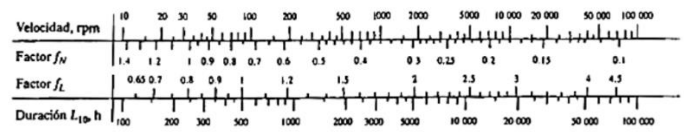


*Figure 4 Reference chart for speed factor ($f_N$) and life factor ($f_L$) used in bearing selection based on rotational speed and required service life.*

### 7.1 Blade Shaft Bearings

The calculated parameters used to validate the bearing selection are presented below, including load factors and service life considerations.

*Table 9 Bearing Selection Parameters – Blade Shaft*

| Parameter | Value |
|---|---|
| Operating speed | 110 rpm |
| Required service life | 30,000 h |
| Equivalent dynamic load | ~652 N |
| Selected bearing | Tapered roller bearings, single row 30203 |

*Figure 5 Tapered roller bearings, single row 30203*

### 7.2 Generator Shaft Bearings

The calculated parameters used to validate the bearing selection are presented below, including load factors and service life considerations.

*Table 10 Bearing Selection Parameters – Generator Shaft*

| Parameter | Value |
|---|---|
| Operating speed | ~560 rpm |
| Required service life | 30,000 h |
| Equivalent dynamic load | ~277 N |
| Selected bearing | Deep groove ball bearing 16005 |

*Figure 6 Deep groove ball bearing 16005*

## 8. Key Selection

Keys were selected according to DIN 6885, ensuring adequate safety against shear and bearing stresses under the defined load conditions.

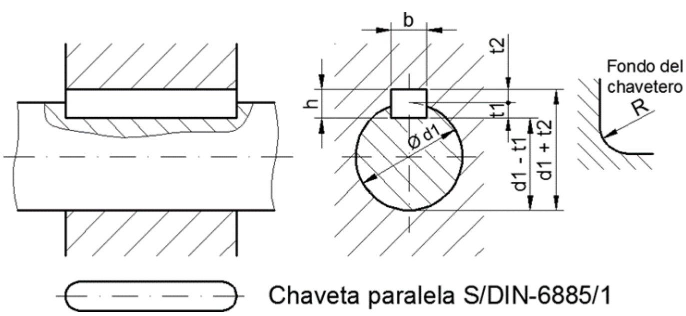


*Figure 7 Standard key geometry reference diagram*

### 8.1 Blade Shaft Key

Key stress verification was performed to ensure safe torque transmission under the applied loading conditions. The verification considers allowable design stress and minimum required key length according to DIN 6885.

*Table 11 Blade shaft Key Stress Verification*

| **Parameter** | **Symbol** | **Value** |
|---|---|---|
| Ultimate tensile strength | $\sigma_u$ | 600 MPa |
| Yield strength | $\sigma_y$ | 330 MPa |
| Shaft diameter | d | 17 mm |
| Key width | b | 6 mm |
| Key height | h | 6 mm |
| Number of keys | N | 3 |
| Allowable design stress | $\sigma_d$ | 110 MPa |
| Calculated minimum key length | $L_{calc}$ | 0.77 mm |
| Selected key length | $L_{sel}$ | 18 mm |

The calculated minimum key length required to safely transmit the applied torque is significantly lower than the selected standard key length. Therefore, the selected DIN 6885 key provides a high safety margin against shear and bearing stresses.

### 8.2 Generator Shaft Key

A stress verification was conducted for the generator shaft key to confirm compliance with allowable stress limits and dimensional requirements specified by DIN 6885.

*Table 12 Generator shaft key stress verification*

| **Parameter** | **Symbol** | **Value** |
|---|---|---|
| Ultimate tensile strength | $\sigma_u$ | 600 MPa |
| Yield strength | $\sigma_y$ | 330 MPa |
| Shaft diameter | d | 25 mm |
| Key width | b | 8 mm |
| Key height | h | 7 mm |
| Number of keys | N | 3 |
| Allowable design stress | $\sigma_d$ | 110 MPa |
| Calculated minimum key length | $L_{calc}$ | 30.88 mm |
| Selected key length | $L_{sel}$ | 32 mm |

The selected key length slightly exceeds the calculated minimum requirement, ensuring safe torque transmission while maintaining compact shaft geometry in accordance with DIN 6885.

## 9. Simulation-Based Validation

### 9.1 Static Structural Analysis

A static structural analysis was performed to verify stress distribution under applied loads and validate analytical sizing results.

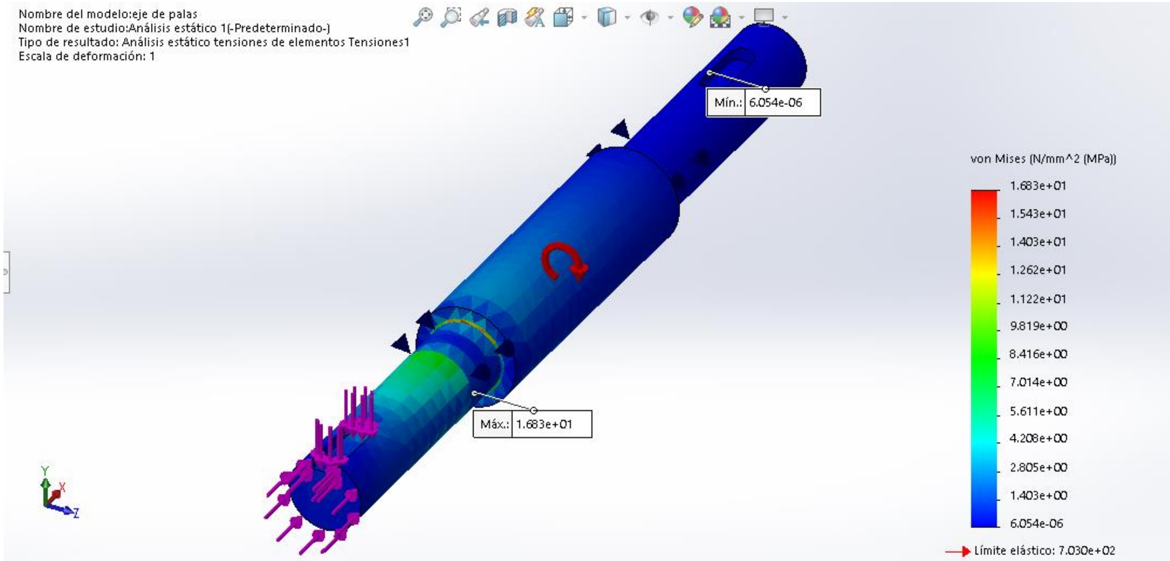


*Figure 8 FEA Von Mises stress plot for blade*

Maximum stresses remained well below the material yield limit, confirming the validity of the analytical design.

## 9.2 Fatigue Analysis (Comparative Study)

Fatigue analyses were performed as a comparative study to visualize damage distribution, relative life, and load influence. Simplified material curves were used; therefore, results are interpreted qualitatively rather than as absolute predictions.

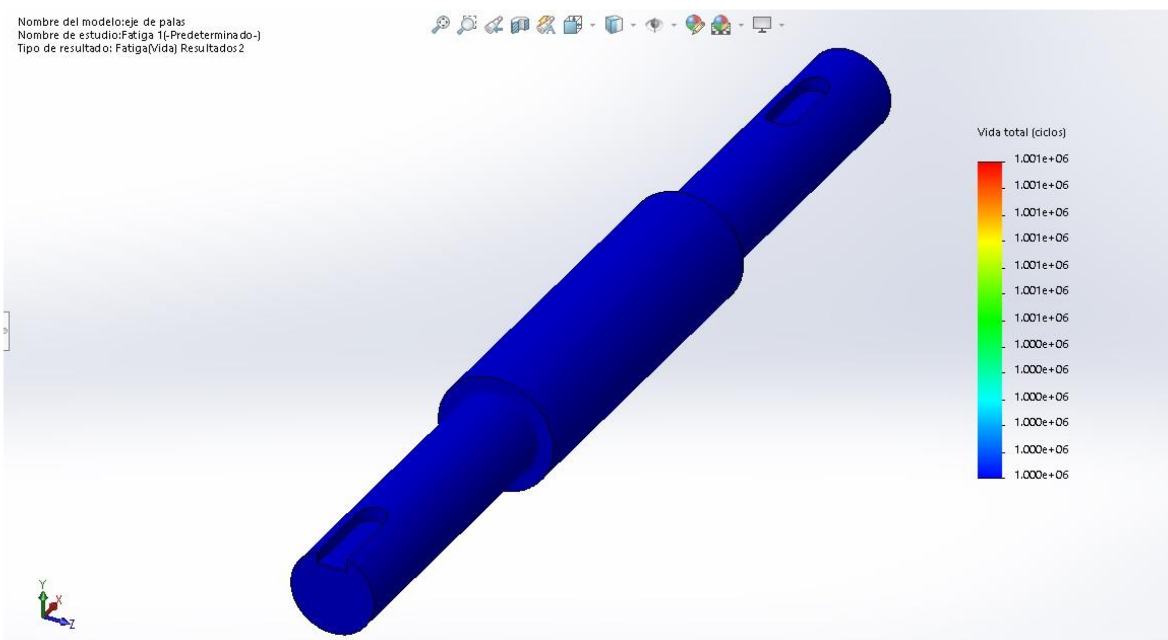


*Figure 9 Fatigue analysis result showing life distribution*

## 9.3 Kinematic Analysis and Validation

A motion study was performed to validate the kinematic behavior of the transmission system. Simulation results were compared against analytical calculations.

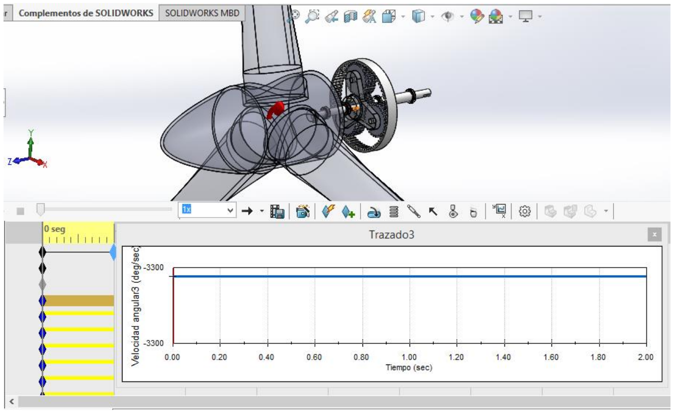


*Figure 10 Motion study of the planetary gear system*

The analytical output shaft speed was compared with the result obtained from the motion simulation. The relative deviation between both approaches remains within an acceptable range for preliminary mechanical design.

*Table 13 Kinematic Validation Results*

| **Parameter** | **Symbol** | **Value** |
| --- | --- | --- |
| Input shaft speed | $n_{in}$ | 110 RPM |
| Output shaft speed (analytical) | $n_{out,calc}$ | 562.22 RPM |
| Output shaft speed (simulation) | $n_{out,sim}$ | 550 RPM |
| Absolute deviation | $\Delta n$ | 12.22 RPM |
| Relative error | $\varepsilon$ | 2.17 % |

The observed deviation is attributed to modeling assumptions and simplifications inherent to the analytical formulation and simulation setup. Overall, the results confirm consistency between analytical and simulation-based approaches.

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## 10. Validation and Consistency Check

Analytical calculations, automated spreadsheet results, and numerical simulations show consistent trends, validating the engineering assumptions adopted in this case study.

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## 11. Conclusions and Observations

- The drivetrain system meets static and fatigue design requirements.
- Analytical and simulation results show good agreement.
- Simplifications were necessary and define the scope of validity of the study.
- The case study serves as a solid reference for preliminary wind turbine drivetrain design.