WEC Development Project

Winter Semester 2024/2025

**Final Design Report**

A logo with a windmill

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Project Name**:**  **Optimus Shakti 5.0**

Sub-Project: **Rotor Hub and Pitch System**

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# List of abbreviations

*[Placeholder for abbreviations]*

*[Example:****OptSha***  *Optimus Shakti5.0]*

# List of symbols

To be continued…

*[Placeholder for abbreviations]*

*[Example:* *density]*

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# Introduction

India ranks fourth globally in wind energy, with an installed capacity of 45 GW, supported by a strong manufacturing base of 15,000 MW annually and a thriving ecosystem. The government fosters growth through incentives like Accelerated Depreciation and concessional customs duty exemptions. With 695 GW of untapped potential, India aims to achieve 140 GW by 2030, having added 2.8 GW in 2023, highlighting its dedication to renewable energy and sustainability. [1]

In line with these national goals, the OPTIMUS SHAKTI project aims to design a 5 MW onshore wind turbine tailored to the environmental conditions of the Gadag district in northern Karnataka, India. The project focuses on developing a state-of-the-art Wind Energy Converter (WEC) optimized for high energy yields in low wind speed regimes. It incorporates an integrated energy storage system to enhance capacity factors while striving for a low economic footprint, reduced CO₂ emissions, and high recyclability. The design targets a power density of 200 W/m² within a market-safe supply chain and adheres to Wind Type Class IEC III(b), suitable for mean wind speeds of 7.5 m/s. The ultimate objective is to achieve the Lowest Cost of Energy (LCOE) while maintaining efficiency and environmental responsibility.

To accomplish this, the project is divided into 10 specialized sub-teams: Project Management, Loads and Dynamics, Dynamic Control, Control Strategy, Rotor Blades, Electrical Drivetrain, Rotor Hub & Pitch System, Rotor Bearing System, Gearbox Coupling Brake, Machine Bed & Yaw System, Energy Storage, and Tower and Foundation. Each team is tasked with designing and developing their respective components to ensure the successful integration and functionality of the wind turbine system.

As part of this effort, our team is responsible for designing the rotor hub and pitch system, including critical components such as the pitch bearing, pitch motor, pinion, lubrication system, and spinner. Furthermore, this team works closely with the Rotor Blades and Main Bearing teams to ensure seamless integration and coordination, a key factor for the project's overall success.

Mostafa Mozafary, Islam Mohammed and Rahul Patil are the members of rotor hub and pitch system team, which was guided by Mr. Christian Bulligk.

# Relevant standards and guidelines

The design of wind turbines must achieve certain objectives such as ensuring regular operational conditions, safeguarding personnel and equipment, and minimizing risks to human life. The structures and components should meet their expected lifespan, and the overall system must demonstrate a sufficient level of reliability.

To meet these goals, wind turbines must be designed according to standards that provide principles, technical requirements, and guidelines for the design and manufacturing of machinery components and structures for wind turbines. These standards ensure safety and performance in both ultimate and serviceability limit states.

For our project the Optimus Shakti, we are following DNV standards. ‘Machinery for wind turbines – DNV-ST-0361’ was consulted to establish standards for designing the Rotor hub and Pitch system components. The design requirements for components are outlined below. [2].

## 2.1 Pitch System

For pitch systems with a pitch gearbox, the strength analysis of the pitch system shall include the pitch gearbox teeth, the blade bearing teeth, the load capacity considering the fatigue loads, static strength against tooth breakage and pitting (Sec.7.2.2.1- [2]).

For the calculation of the loading of the blade pitching system, the design loads as per section 4 of DNV-ST-0437 shall be applied.

In case of pitch systems with a pitch gearbox, the gear load capacity calculation of the pitch gearbox and pitch teeth shall be based on the ISO 6336 series. Additionally, a static strength analysis should be conducted to evaluate the resistance to tooth breakage and pitting, ensuring adherence to the specified safety factors. The safety factors for pitting are defined as 1.0 for the gearbox teeth and 1.0 for the pitch system teeth. For tooth root breakage, the safety factors are specified as 1.1 for the gearbox teeth and 1.2 for the pitch system teeth. (Sec.7.4.2.1- [2])

For the bearings in the actuators of pitch and yaw systems, the static safety factor S0 according to ISO 76:2006 shall be at least 1.1.(Sec.6.5.1.4- [2])

The locking devices shall be so designed that even with a brake removed they can reliably prevent any rotation of the rotor, nacelle or the rotor blade. The locked (engaged) and the unlocked (disengaged) position of the locking devices shall be secured against unintentional locking and unlocking as per Sec.-7.5- [2].

## 2.2 Electric motor

Motors shall be designed according to the operating times and temperatures to be expected. The designed duty types shall be given as specified in IEC-60034 Part-1 ‘Rating and Performance’. (Sec.7.2.4.5- [2])

The nominal torque (Mn) and the equivalent torque (reference torque) of auxiliary motors (e.g. pitch motor) shall be in compliance with the corresponding load calculations [2].

## 2.3 Blade bearing

For blade and yaw bearings, which experience primarily small back-and-forth motions, the static rating is directly calculated from the maximum contact stress between rolling elements and the raceway. The maximum allowable Hertzian contact stress must be specified by the bearing manufacturer, considering factors such as material, surface hardness, and hardening depth. This value should be documented in the design calculations.

The static safety factor for blade and yaw bearings is the ratio between the maximum permissible Hertzian contact stress and the maximum contact stress, and it should be at least 1.1 according to Sec.-6.5.1.3. [2].

The maximum permissible Hertzian contact stress of 4000 MPa for all the roller bearings defined by the ISO 76:2006.

2.4 Lubrication

A consistent supply of lubricant must be maintained on the blade bearing teeth and between rolling elements and the track surface for all wind turbine operations.  
   
A lubrication system is required for the teeth, and its functionality must be documented. If needed, this can be verified through a test run, typically performed once every 24 hours. During this test, the rotor blades and blade bearings should be rotated to ensure adequate relubrication.  
   
Proper collection reservoirs should be provided to handle excess lubricant from the blade bearing components. (Sec.-7.4.2.12- [2])

## 2.5 Bolted Connection

Analytical calculations of axially loaded bolted joints should be performed on the basis of VDI 2230 or other widely recognized design codes and analytical methodologies [2].

## 2.6 Hub and Spinner

For the Hub, cast iron with spheroidal graphite (EN-GJS) according to EN 1563:2012-03 may be used, depending on the mechanical properties required.

Without additional verification, cast iron with a fracture elongation A < 12.5% or an impact energy < 10 J (mean value of three tests) shall not be used for components that play a significant role in the transmission of force and are under high dynamic loading, e.g. rotor hub.

For requirements for manufacturers such as used material, bonding process, building-up the laminates, curing time, resin application and finishing process of nacelle covers and spinners made of fibre-reinforced plastics (FRP), refer to Sec.5 of DNVGL-ST-0376.

Areas of spinner that exhibit the hazard of falling from height or slip-trip shall be equipped with suitable safety attachment points as per Sec.11.2.5- [2].

# Concepts Design and Calculation

## 3.1 Rotor Hub

The rotor hub serves as the initial component of the mechanical drive train. While it is technically part of the rotor, it is functionally and structurally closely linked to the mechanical drive train. Hub serves as the fixture for connecting the rotor blades to the main shaft and transmitting the rotational energy generated by the blades to the mechanical drive train. In pitch-controlled wind turbines, the hub houses the components of the blade pitch mechanism enabling the adjustment of blade angles to optimize energy capture and manage turbine performance under varying wind conditions. [3]

The rotor hub is among the most heavily stressed components of a wind turbine, as it bears the concentrated rotor forces and moments at a single point. Consequently, its material must be chosen with utmost care to ensure durability and fatigue resistance. This necessitates extensive and meticulous work in strength calculations and dimensioning to prevent localized stress concentrations. [3]

There are essentially three possible solutions concerning the selection of materials and the associated design and construction:

1. Welded sheet steel
2. Cast steel
3. A person working on a machine

   Description automatically generatedA person standing next to a large white object

   Description automatically generatedForged steel.

Figure:-3.1. 1 Welded Steel and Casted Steel hub,2012, Source [3]

In the past, all three variations could be found in wind turbines. In the course of time, the cast steel hub has become generally accepted. The technology of cast materials has, however, made considerable progress since then and a much more suitable material for components having dynamic load spectrum has been found to be a spheroidal graphite cast-iron.

The cast hub can be shaped with smooth contours following the load paths. Local stress peaks, resulting from corners and from discontinuities in the wall thickness profile, are avoided.

### 3.1.1 Shapes of the Hub

A drawing of a round object

Description automatically generatedFor casted hubs in modern wind turbines, three popular designs are commonly used: the spherical shape, b) the star shape, and c) the topologically optimized hub, as illustrated in Figure 03:- Shapes of Hub.

b)

c)

a)

Figure:-3.1. 2 Shapes of Hub ,2012. Source [25]

Comparision between spherical and star shape hub as these two sphapes are the basis to start the hub design. And then, the computational approach used to minimize material usage while maintaining structural integrity and resulting design called Topologically Optimized design.

|  |  |  |
| --- | --- | --- |
| **Spherical shape hub** | **Criteria** | **Star shape hub** |
| Follow curvature | **Load direction** | Straight |
| Distributes stress more evenly due to its symmetrical geometry. | **Stress concentration** | Stress is concentrated along the arms and connection points, requiring careful design. |
| Offers lower aerodynamic drag in some cases due to smooth surfaces. | **Aerodynamic Performance** | May generate slightly higher drag due to sharp edges or protrusions. |
| Heavier and less costly | **Weight and Cost** | Lighter and Higher production cost due to complexity |
| Easier to manufacture due to its simple geometry. | **Manufacturing** | Requires more complex manufacturing techniques to shape the arms accurately. |
| High durability due to even stress distribution. | **Durability** | Requires precise engineering to avoid fatigue or failure at stress points. |

Table:-3.1. 1 Comparision between Spehrical and star shape Hub ,2012. Source [25]

To conclude, the spherical-shaped hub ensures uniform stress distribution, enhancing durability and reliability. Its simple design is easier and more cost-effective to manufacture, and the smooth surface minimizes aerodynamic drag. This design is ideal for robust performance in varied conditions and medium-sized wind turbines. These factors make the spherical shape the preferred choice for the Optimus Shakti wind turbine.

### 3.1.2 Material Of the Hub

Spheroidal cast iron (EN-GJS-400-18) is a highly suitable material for wind turbine hubs, offering an exceptional combination of strength and ductility that allows it to withstand both high and cyclic loads efficiently. Its strong fatigue resistance ensures durability over extended periods, while the economical casting process facilitates the creation of intricate hub designs. Additionally, the material’s excellent machinability streamlines precision manufacturing, and its moderate corrosion resistance makes it well-suited for challenging environmental conditions. These benefits, coupled with its established track record in wind turbine applications, make it a reliable and cost-effective option for hub construction. [4]

|  |  |
| --- | --- |
| **Spheroidal cast Iron (EN-GJS-400-18)** | **Value** |
| Tensile strength | 360 MPa |
| Compressive Strength | 275 MPa |
| Yield strength | 220 MPa |
| Young’s Modulus | 169 GPa |
| Density | 7100 Kg/m3 |
| Poisson’s ratio | 0.28 |

Table:-3.1. 2 Mechanical Properties of Spheroidal cast Iron (EN-GJS-400-18) ,2014. Source [26]

### 3.1.3 CAD Design and Dimensions

A white round object with holes

Description automatically generated

**Inner blade bearing**

**Lifting eye**

**3 Manholes**

**Area for mounting of electrical components**

**Rotor shaft bore**

**Access cutout**

Figure:-3.1. 3 CAD model of HUB (Optimus Shakti) ,2025. Source:- Own creation

A white object with a blue line

Description automatically generated with medium confidenceA white object with a red button and blue lines

Description automatically generated with medium confidenceFigure:-3.1. 4 :-CAD model of HUB (Optimus Shakti) ,2025. Source:- Own creation

|  |  |
| --- | --- |
| Rotor Hub diameter (m) | 4.9 m |
| Effective radius (m) | 1.81 m |
| Distance Between Hub Center to Hub Flange (m) | 4.6 m |

Table:-3.1. 3 Dimentions of HUB (Optimus Shakti),2025. Source:- Own Creation

## 3.2 Pitch System

In existing wind turbine technology, all wind turbines are equipped with rotors that utilize blade pitch control called the ‘Pitch System’. This mechanism serves several purposes. The first is to regulate the blade pitch angle, which is essential for controlling the rotor's power output and speed. Typically, a pitching range of 20 to 25 degrees is sufficient to achieve this. This operation functions in case of power optimization, power limitation, and sound operation modes. The mechanism has a secondary function that significantly impacts its design. To enable the aerodynamic braking of the rotor, the blades must be able to pitch to a feathered position. This requirement expands the pitching range to approximately 90 degrees to stop the wind turbine as well as to protect it from the over speeding in extreme weather conditions. Additionally, it aids in reducing the load on the turbine by adjusting the angles of individual blades. [3]

### 3.2.1 Classification of Pitch system of Wind turbine

A diagram of a pitch system

Description automatically generated

### 3.2.2 Comparison between Different concepts of Pitch System

Modern wind turbines commonly use either hydraulic or electric pitch systems, that will be discussed in following sections.

### 3.2.2.1 Hydraulic Pitch System

The hydraulic pitch system functions by maintaining a continuous supply of hydraulic pressure from a power unit, with the fluid delivered through a rotary lead-through and a hollow shaft to the hub. Within the hub, the pitch controller manages valves to direct hydraulic fluid to the pistons, which adjust the rotor blade pitch by extending and retracting. For safety, hydraulic fluid is stored in pressure tanks located in the hub, allowing the blades to move to a feathered position during emergencies. In such situations, a valve releases the stored fluid, driving the piston to position the blades safely as shown in figure. [3]

**A close-up of a machine

Description automatically generated**

Figure :- 3.2. 1 Hydraulic Pitch system ,2025. Source:- [5]

The hydraulic system has advantages such as delivering high power output with compact components, high reliability, fewer parts (eliminating gears), no backlash, and no need for tooth lubrication. However, it also has notable drawbacks, including the risk of fluid leakage, higher maintenance requirements, lower efficiency, and increased energy consumption since the pump operates continuously. Additionally, operational costs are elevated due to regular maintenance, including replacing cylinder seals, tubes every 5–10 years, and periodic oil and filtration replacements [5].

### 3.2.2.2 Electric Pitch System

Modern MW-class wind turbines use electric pitch systems with a planetary gear connected directly to the motor. The motor's torque is transmitted through the gear to drive the pinion wheel, which rotates the blade’s gear ring. The planetary gear reduces the high torque required to move the blade and adjusts the motor’s speed to match the low rotational speed of the pitch system. For safety, each pitch motor has its own power storage unit, like lithium-ion batteries or capacitors, to ensure operation during power outages. [3]

A close-up of a machine

Description automatically generated

Figure :- 3.2. 2 Electrical Pitch System,2024.Source [30]

The electrical system offers significant advantages, including high efficiency with quick responses, low maintenance requirements, and quiet operation, making it ideal for environments sensitive to noise. However, its complexity, higher chance of failure, and the need for gear lubrication are some drawbacks. [5]

### 3.2.2.3 Market Share of Pitch system concepts

A graph of a market share

Description automatically generated

Figure :- 3.2. 3 Global market share of pitch concepts,2024.Source [5]

As demonstrated by the data, the electric pitch system is increasingly favored by manufacturers in comparison to the hydraulic pitch system. Approximately 60% of manufacturers opt for the electric pitch system, a trend that is similarly reflected among Wind Turbine Generators (WTGs) OEMs in India, where over 77% of manufacturers rely on the electric pitch system.

|  |  |  |  |
| --- | --- | --- | --- |
| **Sr.No** | **OEM** | **WTG MODEL** | **Type of Pitch System** |
| 1 | Adani New Industries Limited | MWL-160-5.2MW | Electrical |
| 2 | Envision Wind Power Technologies India | EN-156/3.3 MW | Electrical |
| 3 | Suzlon Energy Limited | S120 DFIG 2.1 MW (50 Hz) | Electrical |
| 4 | Vestas Wind Technology India Private Limited | Vestas V100-2MW | Hydraulic |
| 5 | Siemens Gamesa Renewable Power Private Limited | SG 2.2-122 | Hydraulic |
| 6 | Sany Wind Energy India Private Limited | SI-16840 (4 MW) | Electrical |
| 7 | GE India Industrial Private Limited | GE 2.7 - 132 | Electrical |
| 8 | Inox Wind Limited | INOX DF/3000/145 | Electrical |
| 9 | Senvion Wind Technology Private Limited | 2.3M130/2.7MW | Electrical |

Table:- 3.2. 1 Market share of pitch concepts in indian Market ,2024.Source [32]

Despite certain challenges, the electric pitch system's cost-effectiveness and operational efficiency make it a preferred choice for manufacturers globally. These attributes have been key factors in its selection for the Optimus Shakti.

### 3.2.3 Electric Pitch system

The electric pitch system in a wind turbine consists of several key components that work together to adjust the blade pitch angle for optimal performance and safety. At its core is the electric pitch motor, pitch drive and back-up system.

### 3.2.3.1 Electric Motor

Fundamentally two types of electric motor are available for the pitch actuator of wind turbines, AC(Alternating Current) and DC(Direct Current), which both of them have some pros and cons. Generally, AC motors are cheaper and less service are needed for them that is why they are more common in the current design. On the other hand, DC motors are more expensive but directly can feed them with emergency power supply without any extra converter.

However, underneath we investigate briefly all available types then based on the project priorities we chose that one has a better match.

1. **Asynchronous induction motors**

The conventional three-phase AC induction motor is a constant-speed motor that can easily be used in variable-speed applications, albeit usually with reduced efficiency. [6] The squirrel cage rotor induction motor (IM) has low starting torque and high starting current.

The torque of an IM is directly proportional to the product of the flux per pole, the rotor current, and the rotor's power factor. The maximum torque, also known as the breakdown torque, depends on the rotor reactance per phase at standstill and is independent of the rotor resistance per phase. However, the speed or slip at which it occurs is determined by the rotor resistance per phase. [7]

A graph with lines and numbers

Description automatically generated

Figure :- 3.2. 4 Typical torque/speed curve of Asynchronous induction motors. Source [31]

1. **Permanent magnet motors**

The permanent-magnet synchronous motor offers numerous advantages over other types of machines traditionally used for servo drives. It has a higher torque-to-inertia ratio and greater power density compared to an induction motor or a wound rotor synchronous motor. This makes it suitable for applications such as robotics and aerospace actuators. However, it is challenging to control due to its nonlinear dynamic behavior and time-varying parameters. [8]

Unlike induction motors, the torque-speed characteristic of a does not involve slip, as the rotor is synchronized with the stator's magnetic field. Instead, torque is controlled directly by regulating the component of the stator current.

A diagram of voltage limit curves

Description automatically generated

Figure :- 3.2. 5 Torque-Speed characteristic curve of PMSM AC. Source [31]

1. **Synchronous AC reluctance motor**

The rotor design of a distinguishes it from its and counterparts. Compared to these conventional motors, achieves higher reliability and easier maintenance (due to very low winding and bearing temperatures, as well as the absence of a cage or in the rotor structure), lower cost (due to the lack of compared to ), faster dynamic response (due to its smaller size within the same power range and lower moment of inertia), a higher speed range (due to wide constant-power operation compared to ), and higher efficiency within the same power range and frame size (due to cold rotor operation compared to , and higher power density and torque per ampere compared to ). In this sense, offers the high performance of while remaining as inexpensive, simple, and service-friendly as . [9]

A diagram of a speed and short time overload

Description automatically generated

Figure :- 3.2. 6 Sample torque-speed characteristic curve of SynRM AC. Source [9]

1. **Direct current DC motors:**

The motor consists of a stator and a rotating part, the armature, separated by an air gap. The armature is wound with coils that are connected to a commutator. During rotation, the polarity of the armature winding is changed through the sliding contacts (carbon brushes) running on the commutator, thereby generating a constant torque.

If motors are used in applications requiring relatively high positioning accuracy, servo technology can be integrated into the machines. The additional module required for this is an encoder system, which can take the form of a resolver, an incremental encoder, or, more commonly, an absolute encoder. This enables the exact speed of the motor to be measured and the precise position of the armature to be determined. [10]

### 3.2.3.2 Pitch drive selection

In this project, the design priorities are cost, reliability, and low weight. In terms of cost, asynchronous induction motors and synchronous reluctance motors are more favorable. However, as mentioned earlier, offers higher performance compared to . Therefore, has been chosen for this project.

**Pitch actuator static calculation:**

In the pitch actuator calculation procedure, torque and rotational speed are two important parameters that should be investigated. Additionally, regarding torque, we first calculate the required torque on the blade side. Then, we examine whether the selected pitch actuator from the market can fulfill the required torque as well as the rotational speed.

**Required torque on the blade-side:**

However, the required torque on the blade side consists of three parts, the acting wind torque , the friction torque of the pitch bearing , and the torque obtain from blade inertia The relevant equation is shown in 1.1. and is pitch angular acceleration [11].

The inertia of blade about the pitch axis is and the pitch angular acceleration is . While the desired pitch angular speed is assumed 2 to convert it to we have:

The friction of the pitch bearing comes with highest uncertainty; it depends on the type of bearing, the number of rolling elements, and the lubrication system. Which in Optimus Shakti pitch bearing is a three-row roller bearing [11].

However, to estimate the friction torque of the pitch bearing, we have used equations provided by bearing manufacturer. The overall friction torque is made up of rolling friction, dynamic friction and lubricant friction. [12]

: Speed of the large diameter bearing

: Raceway factor = 1.73 (constant)

: Friction coefficient

: Axial load

: Radial load

: Bolt connection factor

: Specific friction force

: Raceway diameter

: K-factor

: Adjacent construction factor

So, we have:

The acting wind torque varies with different pitch angles and wind speeds, so an aeroelastic simulation is required. On the other hand, the acting wind torque can either be supportive or not, depending on direction of rotation.

The total required torque to turn blade in the ring of the pitch bearing with a safety factor is:

\* added as the safe factor.

**Provided torque on motor-side:**   
initial data:

* Bolt Circle diameter of the blade = BCD = 3600
* Gear Module =
* Gear Pitch diameter =
* Number of teeth for the ring =
* Number of teeth for the Pinion =

The transmission ratio of the bearing and the pinion:

The required torque in the pinion of the gearbox:

Regarding the gearbox of pitch motor, while we know three stages gearbox is common therefore, we assumed 170 as its ratio:

Total pitch system ratio is:

(3.2.9)

Obtaining the motor torque depends on the transmission ratio between the pinion and the blade bearing ring and the electric motor gearbox ratio . Moreover, losses in teethes, gearbox, motor and converter increase the required power as well.

(3.2.10)

On one hand, 217 Nm represents the maximum torque pitch motor should provide. On the other hand, in Optimus Shakti dynamic load is not available so we do not know for how long we need this maximum torque therefore we assume it needs in short time.

A white sheet with black text

Description automatically generatedNext, we investigated the available electric motors in the market to find a suitable one. We chose IE5 Synchronous AC reluctance motor from with a frame size is a nominal power and a nominal rotational speed . The nominal torque is while the maximum rotational speed can reach and the ratio between the overload and nominal torque is . [13]

Figure :- 3.2. 7 ABB Synchronous reluctance motors. Source [13]

Based on the selected electric motor, we examine the torque provided by the motor against the required maximum blade-side torque to determine if it is sufficient to counteract it. So:

(3.2.11)

Since the system's efficiency is not an additional is incorporated to account for losses.

Regarding the selecting of the proper gearbox, the motor with frame size and mounting configuration is fully compatible with the Bonfiglioli gearbox due to its adherence to standardized flange dimensions.

**Brake:**  
The highest brake torque for the pitch actuator in the Shakti Optimus wind turbine is most likely to occur during where the wind turbine brought the blades to a safe position (often feathered) quickly under extreme wind conditions. Then required brake torque would be so:

While **dynamic loads** were not available for Shakti Optimus project, we assumed that selected drive is feasible for all probable conditions.

## 3.3 Blade bearing

For blade pitching to be implemented, it is essential to enable the rotor blades to rotate around their longitudinal axis. This rotational capability is essential for adjusting the blade's pitch angle, which optimizes energy capture and protects the turbine under various wind conditions. Although the required rotation angles and speeds are relatively modest, the blades are predominantly supported by bearings at their roots. [3]

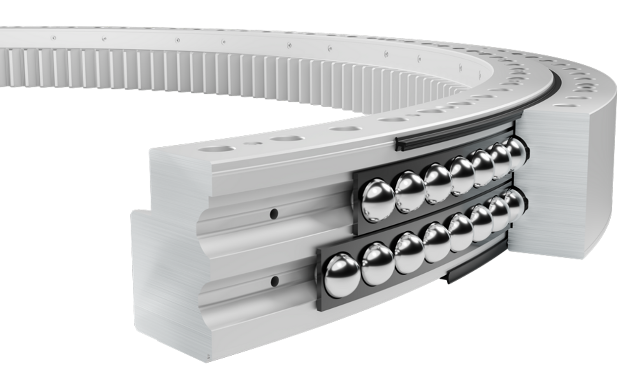
Blade bearings connect to the rotor blade and the rotor hub of a wind turbine by means of bolted connections. Their rings tend to be made of steel, with 42CrMo4 type steel is the most common choice of material, whereas rolling elements are made from 100Cr6. [14]

### 3.3.1 Types of Bearings

The following basic design types are in commercial use as pitch and yaw rolling bearings of wind turbines:

1. **Four-point contact ball bearings**

Four-point contact ball bearings are the dominant type for pitch bearings bearings. For smaller turbines, they have one row; for larger turbines, usually two. Four-point contact ball bearings are generally capable of carrying loads in any degree of freedom and allow for high deformations in the interfaces.

A curved metal object with a yellow stripe

Description automatically generatedDouble row 4-point contact ball bearings utilize a **point contact design** that leads to **high contact pressure**. They have an **initial contact angle of 45 degrees**, which enables them to accommodate combined axial and radial loads. However, their **load-bearing capacity is comparatively lower** than that of triple row roller bearings, as the contact area is reduced. These bearings are prone to **high bearing ring deformation** and ring separation under heavy loads, which can impact their longevity. They also experience **high edge loading**, as the balls tend to move toward the edges of the raceway under load, increasing wear and stress Additionally, they handle **high radial loads**, but this contributes to periodic ring stresses, which can lead to fatigue over time.

Figure :- 3.3. 1 Double row four-point contact ball bearing external and internal toothing,2024. Source [27]

Apart from some disadvantages this type of bearing offers advantages such as **both raceways carry the load simultaneously**, enabling better load distribution and enhanced stability. Their **manufacturing complexity** is **moderate**, as their design is simpler compared to triple row roller bearings. Consequently, the **manufacturing costs** are **comparatively cheaper**, making them more economical for applications where cost is a significant factor. These bearings experience **less wear** due to their **point contact mechanism**, which reduces friction and surface stress. However, their **service life** is **shorter** when subjected to very high load applications, limiting their suitability for extreme operating conditions. [15]

1. **Three-row roller bearings**

A curved metal object with a yellow stripe

Description automatically generatedThe three-row roller bearing has two rows of axial roller elements that help it endure axial loads and one row of radial roller elements that help it survive radial loads. These two-row roller components aid in absorbing heavy radial loads.

Figure :- 3.3. 2 Three raw rollers bearing with internal toothing and External toothing,2024. Source [27]

Triple row roller bearings are characterized by a **line contact design** that results in **low contact pressure** compared to other bearing types. They have a specific contact angle configuration, with the axial row at **90 degrees** and the radial row at **0 degrees**, enabling better support for axial and radial loads. Due to their **large contact area**, they exhibit a **high load-bearing capacity**, making them ideal for applications requiring robust performance. Additionally, they experience **low bearing ring deformation** and minimal ring separation under load, enhancing their durability. Edge loading, which occurs due to roller tilt during deformation, is generally **low** in these bearings. Moreover, they handle radial loads effectively, with very **low stress levels**, due to their dedicated radial row of rollers. [16] [11]

The manufacturing complexity of these bearings is very high due to their intricate design. This complexity translates to high manufacturing costs. However, their **robust design** provides a **high life expectancy**, making them ideal for applications demanding **long-term durability** and **reliability**. This type of bearing is **more compact** compared to double-row four-point contact ball bearings for the same load capacity, making it well-suited **for Optimus Shakti**.

1. **Three-row ball and roller bearings**

This type of bearing is an improvement of double row four-point contact ball bearing. Three row ball and roller bearings tend to have lower radial deformations under load than four-point contact ball bearings, but they do not provide significantly higher load ratings. This is why this type of bearing is not used in large-capacity wind turbine. [16]

A diagram of a machine

Description automatically generated

Figure :- 3.3. 3 Three raw ball and roller bearing,2024. Source [16]

1. **Two-row angular contact bearings**

Angular contact ball bearings have inner and outer ring raceways that are displaced relative to each other in the direction of the bearing axis. This means that these bearings are designed to accommodate combined loads, i.e. simultaneously acting radial and axial loads.

The axial load carrying capacity of angular contact ball bearings increases as the contact angle increases. The contact angle is defined as the angle between the line joining the points of contact of the ball and the raceways in the radial plane, along which the combined load is transmitted from one raceway to another, and a line perpendicular to the bearing axis.

A diagram of a ball bearing

Description automatically generatedA ball bearing with several balls

Description automatically generated

Figure :- 3.3. 4 Double row angular contact ball bearing,2024.Source [28]

While they perform well under combined loads, their load capacity is lower than that of larger, more robust designs like triple-row roller bearings. This limits their use in very large-capacity wind turbines. The **point contact design** leads to higher localized stress, which can result in increased wear and reduced service life under extreme loading conditions. [16]

### 3.3.2 Blade attachment

The connection of Wind turbine blades with the hub can be possible via two different concepts, inner rotating and outer ring rotating. Each choice has its own advantages and disadvantages. Many aspects influence the decision, such as the cost of the surrounding parts, assembly processes, Maintenance accessibility and technical feasibility, this is a very important step, which will again influence the further design of the Hub.

A close-up of a machine

Description automatically generated**3.3.2.1 Outer ring Rotating**

In this configuration, the blade is attached to the outer ring of the pitch bearing, while the inner ring is fixed to the hub. This design offers several advantages, such as enabling a smaller and lighter hub, simplifying the connection of the lubrication system to the blade bearing, and creating additional space within the hub.

However, a significant drawback is the increased risk of blade detachment if the bearing cracks, posing serious safety concerns.

Figure :- 3.3. 5 Outer Ring mounted blade,2024.Source [29]

Additionally, since pitch system are located outside the hub, extra protective measures, such as a spinner and additional support structures (teeth cover), are necessary to shield these components from environmental exposure. [14]

**3.3.2.2 Inner ring Rotating**

As the name suggests, this configuration involves the blade being mounted on the inner ring, while the outer ring is fixed to the hub. This setup allows for a stiffener plate to be placed between the inner ring and the blade without requiring additional components, enhancing the structural stiffness of the bearing. Furthermore, in the event of a ring crack, the likelihood of blade detachment is significantly reduced.

A close-up of a machine

Description automatically generatedDuring turbine assembly, blade bolts can be tightened from within the hub, making the process more convenient compared to external configurations. Additionally, the pitch drives are housed within the hub, eliminating the need for extra housing and simplifying maintenance tasks.

Figure :- 3.3. 6 Inner Ring mounted blade,2024.Source [30]

The inner ring configuration also benefits from a larger diameter, allowing for the inclusion of more rolling elements. This design ensures better load distribution and reduces the forces on individual rolling elements if the moment remains constant. [14]

In summary, the choice between inner and outer ring configurations depends on multiple factors. While the outer ring design offers a more compact structure, it is less favorable for maintenance and the safety of components and technicians. Conversely, the inner ring design prioritizes durability, ease of maintenance, and safety, making it the preferred choice for the Optimus Shakti turbine.

## 3.3.3 Three Rows roller bearing Dimensioning of the Raceway system

The statical calculations aim to ensure that the blade bearing can withstand the maximum load situations occurring for all operating conditions of the turbine.

The calculation starts with the analysis of the load time series for the DLCs in order to identify the maximum resulting bending moment.

The dominant load of a pitch bearing is the resulting moment (Mxy) Which is the combination of flap-wise and edge-wise bending moment.

Mxy = (Source [14] Eq. 3.1, p. 11) (3.3.3.1)

Radial (Fz) and Axial (Fxy) forces influence the rolling body loads as well

The following data are provided by the load team

* **Bending Moment Mxy = 40000 KNm**
* **Radial force (Fz) = 800 KN**
* **Axial force (Fxy) = 900 KN**

**Assumption for the Calculations**

* The upper and lower axial raceways have the same raceway diameter Dpw [mm]
* The upper and lower axial raceways have the same roller diameter Dw [mm]
* Roller length L [mm] = roller diameter D [mm]
* The radial raceway is not considered

**Calculation of Hertzian pressure**

**Step 1: Define raceway system parameters (Dpw, Dw, L and Z)**

* Set raceway diameter Dpw [mm]

For calculating Dpw (bearing raceway diameter), bolt circle diameter DBcd  need to considered. Which was 3600 mm, given by the Blade team.

Dpw  = DBcd +(2\*W3)+(2\*S\_u)+(WA-L)+L = 3801 mm (3.3.3.2)

* A drawing of a mechanical part

  Description automatically generatedSet roller diameter Dw [mm]

Roller diameters up to 80 mm would theoretically be possible (from raceway surface hardening point of view). However, the length of the rollers can cause large bearing ring deformations. Therefore, roller diameters greater than 65 mm should be avoided.

Use roller diameters 50, 55, 60, 65 mm.

Dw  = 65 mm

* Roller length L [mm] = roller diameter Dw [mm] = 65 mm

Figure :- 3.3. 7 Three roes roller bearing,2024.Source [27]

* Estimate number of rollers per axial raceway Z for the given Dpw and Dw

Z = = 160 (3.3.3.3)

**Step 2: Calculate maximum roller force (Qmax)**

Maximum roller force for ideal conditions (rigid companion structure, same stiffness over the entire bearing circumference):

Qmax\_rigid = + = 274.665 KN (3.3.3.4)

The influence of the companion structure can be considered by an additional factor Kq: and For good rotor blade and rotor hub designs: Kq ≤ 1.15.

Qmax\_flexible = Kq\* Qmax\_rigid  = 315.865 KN(3.3.3.5)

The highest contact force Qmax obtained which is used to calculate the highest resulting Hertzian pressure Pmax .

**Step 3: Calculate maximum contact stress (p0)**

The Hertzian calculations cannot be solved analytically. Hence, equations for an approximation were well established. For example, Houpert published a method. The maximum pressure (Pmax ) follows according to 3.3, the Hertzian contact ellips radii are required inputs for the equation. Here, another iterative process comes into play.

b= 0.00335\* = 1.3743 (Source [16] Eq. 6.14, p. 37) (3.3.3.6)

= L - (2\*r\_axial) = 61 mm (Source [16] Eq. 6.14, p. 37) (3.3.3.7)

P0max  = = 2398.72 N/mm2 (Source [16] Eq. 6.5, p. 35) (3.3.3.8)

Due to bearing ring deformation, the rollers tilt leading to an uneven pressure distribution and thus higher contact stresses. This can be considered by an additional factor Kp: For stiff bearing design, Kp ≤ 1.20.

Pmax = P0max \* Kp (1.20) = 2878.47 N/mm2 (3.3.3.9)

According to ISO 76, Permissible contact stresses shall be less than 4.2 GPa for ball bearings and 4.0 GPa for roller bearings. However, since these pressures are related to extreme wind conditions, typical contact pressures during normal operation of the turbine are lower. However, when using the criterion 3300 N/mm², it is likely that the bearing is also feasible for the fatigue loads.

### 3.3**.4 Pitch bearing teeth gear: - Safety factors against pitting and tooth root breakage**

The load limit for gear wheels is determined by the load capacity. According to DIN 3979, there are three main types of damage to gears that determine the load limit:

* Tooth breakage due to excessive bending stress in the tooth root,
* Tooth flank fatigue due to material fatigue (pitting, chipping),
* Scuffing due to the combined effect of pressure and sliding speed.

Tooth breakage typically refers to the complete or partial fracture of a tooth when its load-bearing capacity is surpassed. To prevent this, the safety and integrity of the tooth root must be evaluated.

Fatigue on tooth flanks appears as pitting when the contact pressure exceeds the tolerable limit. Repeated loading and unloading cause material fatigue, leading to pits after numerous rollovers. This is critical only if pitting worsens with service life or pits grow larger, indicating diminished load capacity due to exceeded permissible flank pressure, so the load capacity of the tooth flank needs to be checked.

Scuffing occurs when the lubricating film between tooth flanks breaks due to high pressure and sliding speed (hot scuffing) or high pressure and low speed (cold scuffing). This causes direct metal contact, leading to roughened bands and potential localized welding. Contributing factors include surface roughness, insufficient backlash, or unsuitable lubricants. In high-speed gearboxes, scuffing increases heat, forces, and noise, potentially causing severe flank damage and tooth breakage. The scuffing load-bearing capacity ensures adequate safety against such damage.

Scuffing can be effectively prevented by choosing appropriate materials, ensuring proper maintenance and lubrication, or following a correct running-in process. Consequently, the focus is typically on evaluating the tooth root and flank load-bearing capacities through recalculations, requiring comprehensive knowledge of all gearing parameters. [17]

**Forces on gear**

**Maximum forces**

By given Maximum torque of 500 KNm, the tangential force on the gear teeth can be calculated, as follows:

Ft1,2 = = 290 KN, (Source [17] Eq. 21.66, p. 821) (3.3.4.1)

A diagram of a curved object

Description automatically generated with medium confidenceWhere, = effective diameter of pitch bearing gear teeth,

= mn\*Z2 = 3440 ,

This Tangential force is divided into two components as shown in fig, the normal tooth force Fb (perpendicular to the flank and mating flank at the point of contact) and the radial force (always directed towards the respective wheel center) Fr.

Since 𝛼 is the same as (20°) these forces can be calculated with the following equations (Roloff-mateck, Pg 821):

Figure :- 3.3. 8 Forces on Gear ,2005.Source [17]

Normal tooth force

Fbn12= = 309.35 KN (Source [17] Eq. 21.68, p. 821) (3.3.4.2)

Radial force

Frn12 = = 105.80 KN (Source [17] Eq. 21.69, p. 821) (3.3.4.3)

**Nominal forces:**

With the given nominal torque of 250 kNm per pitch system the tangential force on the gear can be calculated by the following equation.

Ft1,2\_nomi = = 145.34 KN (Source [17] Eq. 21.66, p. 821) (3.3.4.4)

And Fb and Fr can be calculated same as above equations.

Fb12\_nomi= = 154.67 KN (Source [17] Eq. 21.68, p. 821) (3.3.4.5)

Fr12\_nomi = = 52.90 KN (Source [17] Eq. 21.69, p. 821) (3.3.4.6)

**Stress Influencing Factors**

According to DIN 3990-1, methods A to E are used to determine these factors:

* **Method A**: Precise measurements and detailed analysis, requiring complete gear and load data.
* **Method B**: Simplifies each gear pair as a mass-spring system, ignoring other stages.
* **Method C**: A further simplification of B, assuming subcritical speeds and basic material properties.
* **Methods D & E**: Introduce additional simplifications, such as constant line load.

For practical design, simplified methods (C & D) are often used as most industrial gearboxes operate in the subcritical range, ensuring faster calculations with sufficient accuracy.

The application factor (operating factor) Ka​, accounts for external additional forces specific to the input and output machines connected to the gearbox. These forces include shocks, torque fluctuations, and load peaks.

Ka = 1 (Source [17] TB 3-5, p. 1069)

Dynamic factor Kv, it records the internal dynamic additional forces that arise under load due to deformation of the teeth wheel centre and all other force-transmitting elements of the gearbox.

Kv = 1+ \*K3 = 1.003 (Source [17] Eq. 21.73, p. 824) (3.3.4.7)

Where,

K1 = 24.5; Gear quality 8 considered (Source [17] TB 21-14, p. 1069)

K2 = 0.0193; (Source [17] TB 21-14, p. 1069)

K3 = 0.01\*Z1\*Vt\* = 0.0062 (Source [17] Eq. 21.74, p. 824) (3.3.4.8)

**Face width factors**

Face width factors "KHϐ" and "KFϐ" They take into account the effects of uneven force distribution over the tooth width on the flank stress (‘KHϐ’) or on the tooth root stress (‘KFϐ’).This is caused by the flank line deviations that occur in the loaded state as a result of mounting and elastic deformations (fsh) and manufacturing deviations (fma).

KHϐ = 1 + = 1.2587 (Source [17] Eq. 21.75, p. 825) (3.3.4.9)

Taking into account the following influencing variables:

Fsh Flank line deviation due to deformation can be determined as a first approximation from experience with TB 21-16 (Source [17] Eq. 21.75, p. 825) (3.3.4.10)

Fsh = 0.023\*\* =5.3895µm

Where, d1 = pitch circle diameter of pinion = 256

b = tooth width = 162.5 mm

= Kv\*(Ka\*Ft/b)= 894 N/

Now, we have to find fma is production related flank line deviation as below,

Fma = c\*4.16\*\* = 21.97 µm (Source [17] Eq. 21.76, p. 825) (3.3.4.11)

Where , c = 1 for wheel pairs without adjustment measures

= 2.59, (Source [17] TB 21-14, p. 1069)

Effective flank line deviation before running-in

Fϐx= Fma + 1.33\* Fsh = 29.14 µm (Source [17] Eq. 21.77, p. 825) (3.3.4.12)

This amount is reduced by the run-in amount ‘y’ according to TB 21-17, so that after the run-in the effective flank line deviation is,

Fϐy = Fϐx-yb = 23.14 µm (Source [17] Eq. 21.78, p. 826) (3.3.4.13)

Where, yb  = 6 (Source [17] TB 21-17, p. 1072)

The exponent for determining the width factor for the tooth root results from,

Nf = = 0.8121 (Source [17] Eq. 21.79, p. 826) (3.3.4.14)

When this is done, the tooth base width factor is calculated,

Kfϐ = = 1.2054 (Source [17] Eq. 21.74, p. 825) (3.3.4.15)

Next values for the face load factor 𝐾𝐻𝛼 and 𝐾𝐹𝛼 are assumed according to (Source [17] TB 21-19, p. 1073), with the toothing quality of 8 and regular straight gears.

𝐾𝐻𝛼 = 𝐾𝐹𝛼 = 1.1

With all these values the total influence factors 𝐾𝐹𝑔𝑒𝑠 (tooth base strength factor) and 𝐾𝐻𝑔𝑒𝑠(tooth flank/pitting factor) are determined,

𝐾𝐹𝑔𝑒𝑠 = Ka\*Kv\* 𝐾𝐹𝛼\* Kfϐ = 1.3263 (Source [17] Eq. 21.81, p. 827) (3.3.4.16)

𝐾𝐻𝑔𝑒𝑠 = = 1.1768 (Source [17] Eq. 21.81, p. 827) (3.3.4.17)

**The maximum local tooth root stress** occurring when a fault-free gear is loaded by the static nominal torque can be determined from,

Ϭf01 = \*Yfa\*Ysa\*Yε = 164.14 N/mm2  (Source [17] Eq. 21.82, p. 827) (3.3.4.18)

Where, Yfa = 2.42 (Form factor ) (Source [17] TB 21-20 a, p. 1317)

Ysa= 1.82 (Stress correction factor) (Source [17] TB 21-20 b, p. 1317)

Yε= 0.25 + = 0.6667 (Overlap Factor) (Source [17] Eq. 21.82, p. 828)

Total tooth root stress can be calculated as follows,

Ϭf12= Ϭf01\* 𝐾𝐹𝑔𝑒𝑠 = 217.71 N/mm2  (Source [17] Eq. 21.83, p. 829) (3.3.4.19)

**Tooth root limit strength** can be calculated by,

Ϭfg1  = 2\* Ϭflim\*YNT\*YX = 642.06 N/mm2 (Source [17] Eq. 21.84, p. 829) (3.3.4.20)

Where, Ϭflim = 369 N/mm2 for Bearing and 500 N/mm2 for the Pinion(Roloff-mateck, Eq 21.84, Pg 829)

YNT= 1 (Service life factor) (Source [17] TB 21-20 a, p. 1317)

YX = 0.87 (Size Factor) (Source [17] TB 21-20 d, p. 1317)

**Safety of the tooth root load capacity**

Sf1 = = 2.94 (Source [17] Eq. 21.85, p. 830) (3.3.4.21)

**Verification of pit load-bearing capacity**

The calculation of the pitting load-bearing capacity is based on the flank pressure ϬH at the rolling point , and The flank pressure that occurs at the rolling point is, (Source [17] Eq. 21.85, p. 830) (3.3.4.22)

ϬH0 = \*\*\*\* = 755.17 N/mm2 (3.3.4.22)

Where,

* is zone factor, takes into account the flank curvature at the rolling point, which is 2.5 for straight gears. (Source [17] TB 21-21 a, p. 1319)
* is Elasticity factor, which is 189.8 N/mm2 (Source [17] TB 21-21 b, p. 1319)
* is Coverage factor, takes into account the influence of the load distribution on several pairs of flanks involved in the engagement on the calculated Hertzian pressure, which is 0.825 (Source [17] TB 21-21 c, p. 1320)

is Screw factor, it records the improvement in load-bearing capacity under flank pressure with increasing helix angle, which is 1.

flank pressure occurring at the rolling circuit is,

ϬH= ϬH0\* 𝐾𝐻𝑔𝑒𝑠 = 888.71 N/mm2 (Source [17] Eq. 21.89, p. 832) (3.3.4.23)

**flank limit strength**

Flank limit strength can be determined by the following equation, (Source [17] Eq. 21.90, p. 832)

ϬHG = ϬHlim\*\*\*\*\*\* = 1356.87 N/mm2  (3.3.4.24)

Where,

* ϬHlim is tooth flank fatigue strength, 1204.44 N/mm2  (Source [17] TB 21-21, p. 1320)
* is Service life factor, takes into account a higher permissible pressure if a limited service life is required in timing gears, which is 1. (Source [17] TB 21-22, p. 1321)
* is Lubricant factor, for mineral oils depending on the nominal viscosity at 50° or 40° , which is 1.19 (Source [17] TB 21-22 d, p. 1320)
* is Velocity factor, takes into account the influence of the circumferential speed on the flank load-bearing capacity, which is 0.89 (Source [17] TB 21-22 b, p. 1319)
* is Roughness factor, determines the influence of surface roughness, which is 0.87 (Source [17] TB 21-22 c, p. 1319)
* is Material pairing factor, takes into account the increase in flank strength of a gear made of structural steel, heat-treated steel, which is 1.125 (Source [17] TB 21-22 e, p. 1321)
* is Size factor, calculates the influence of the tooth dimensions, which is 0.87 (Source [17] TB 21-20 d, p. 1318)

**Safety of the flank load capacity**

= = 1.5268 (Source [17] Eq. 21.90 a, p. 833) (3.3.4.25)

### 3.3.5 CAD model of Bearing and dimensions

## 3.4 Bolted Connection between Hub and Shaft

The hub and shat are connected by a bolted joint, these bolts experience axial, lateral, bending and torsional loads from static, periodic, and random forces. The goal is to calculate the loads on each screw to identify the most stressed one, ensuring safety and reliability. The diagram illustrates these forces and moments on the rotor flange and its connection.

Calculating loads on screws is essential to prevent overloading, optimize their design, and identify potential failures early for effective maintenance. [5]

Diagram of a machine with a diagram

Description automatically generated

Figure :- 3.4. 1 Loads on bolts on Rotor flange connection,2024.Source [5]

A diagram of a curved metal object

Description automatically generated with medium confidenceNow, there are different concepts for bolted connection as shown in fig.

The design A provides increased capacity to handle torque and bending moments, allows for easy installation from the nacelle, requires machining of only one flange, and ensures an optimal flow of forces for improved structural efficiency.

Both designs are similar; however, the primary drawback of design B is that the internal flange requires machining, and installation is only possible from the hub side, increasing handling effort.

Figure :- 3.4. 2 Rotor flange connection,2024.Source [5]

Therefore, we have chosen concept A for Optimus Shakti. [5]

### 3.4.1 Calculation of Bolted Joint

Assuming specific conditions for calculation such as, the flange must be rigid to minimize deformations and prevent connection skidding, with screws designed to function without pre-stressing.

The following data are provided by the load team

* **Total tensile force (Axial force) of Flange (F\_a\_f\_ges) = 700 KN**
* **Lateral force on Flange (F\_q\_f\_ges) = 2000 KN**
* **Bending moment on Flange (M\_b) = 25000 KNm**
* **Torsional Moment on flange (Mt) = 12000 KNm**

**Step 1: Total Shear force on bolt by Torsion and Lateral forces ( Fq\_ges)**

Shear force on Bolt by External Lateral Force () = = 17.24 KN (3.4.1.1)

Where, = Number of bolts in inner ring = 56

= Number of bolts in outer ring = 60

Shear Force on screw by torsional moment () = = 96.50 KN (3.4.1.2)

Where, = Outmost screw circle diameter, 1.170 m

= Number of bolts per bolt circle diameter, (60 and 56)

= Number of bolt circle diameter (2)

**Total Shear force on bolt by Torsion and Lateral forces ( Fq\_ges)**

**Fq\_ges =** += 113.84 KN (3.4.1.3)

**Step 2: Total axial force on bolt by bending moment and Tensile force (F\_a\_ges)**

A diagram of a circle with circles and arrows

Description automatically generatedAxial force on bolt by Bending moment

() = = 400.00 KN (3.4.1.4)

Where, =

= Distance of the perpendicular screw to the bending axis [m]

= Total number of Screws

Figure :- 3.4. 3 Distance of bolts to bending axis,2024.Source [5]

Axial force on bolt by tensile force () = = 6.0345 KN (3.4.1.5)

**Total axial force on bolt by bending moment and Tensile force (F\_a\_ges) =**

**F\_a\_ges =**  + = 406.03 KN (3.4.1.6)

**Step 3: Required area of bolts for carrying total load A\_req**

First, we have to calculate, = = 379.46 KN (3.4.1.7)

Total force on bolts is summation of total axial force and clamping force,

=  **+**  = 785.506 KN (3.4.1.8)

Now, the bolt diameter required for the design of the screw connection can be calculated by,

= = 835.64 (3.4.1.9)

Where, = proof stress of the screw material (Source [17] TB 08-4, p. 1130)

So, A\_req(835.64 ) is < As\_M39 (976 from (Source [17] TB 08-1, p. 1127) So, the design is safe.

## 3.5 Bolted connection between Bearing and Hub

The blade is attached to the blade root and hub flange using a bolted connection. This connection must provide a secure and permanent bond capable of withstanding the dynamic and ultimate loads throughout the turbine's service life. Additionally, the bolted connection facilitates the preloading of the rolling elements in the three-row roller bearing.

The preliminary design for the outer ring follows an iterative approach to balance various requirements. The initial step in defining the bolted connection design involves determining the number of bolts. This number should be based on the available space and the ultimate loads expected to be carried by the bearing flange. [14]

N = = 147 Number of bolts (3.5.1)

Where, D = bolt circle diameter of Bearing outer ring = 4006 mm

d = bolt diameter = 39 mm (M39 bolts)

The number of bolts is determined by the space required for tightening tools, such as hydraulic torque wrenches or bolt tensioners. This design uses 145 bolts, resulting in an approximate 87 mm spacing on the outer ring, which appears adequate.

Following are the Loads are given by the loads team,

* Axial load () = 800 KN
* Radial load () = 900 KN
* Resulting bending moment based on blade bottom coordinate () = 40000 KNm
* Bolt circle diameter of the outer race of pitch bearing () = 4006 mm

The ultimate loads acting on the flange have to be transferred into an axial force carried by the bolts. [5]

= = 280.96 KN (3.5.2)

The clamping force needed to provide enough resistance against the opening of the contact can be estimated as follows

= = 337.16 KN (3.5.3)

The factor s is set to 1.2, which is typical for dynamic load situations. In general, the clamping force F K should be higher than the working load F to ensure a positive residual clamping load. Both loads are used to estimate the stress cross section needed for the bolted connection.

This estimation needs values such as , deter mines a factor for the tightening method, *K* is a reduction factor based on the type of bolt, ϐ a factor for the elastic resilience of the used bolt, *E* the Young’s modulus for steel, a factor for embedding, the clamping length and the proof stress of the used bolt. For the preliminary design in this section, the following values are chosen according to reference. [5]

*=* 900 MPa according to bolt type 10.9 (Source [17] TB 8-4, p. 1130)

= 1.2 Hydraulic tensioning (Source [17] TB 8-11, p. 1139)

*K* = 1.15 for the shank screws(Source [17] TB 8-12 b, p. 1140)

= 0.011 , Axial load related factor

*ϐ =* 1.1 for shank screws (DIN EN ISO 4014 and DIN EN ISO 4762)

= 450 mm, length of clamped part (initial estimate)

*E* = 210 MPa, Young's modulus for steel [N/mm^2]

Based on the values estimated for this preliminary design, the needed stress cross section according to (3.5.4)is, [5]

|  |  |
| --- | --- |
| = | (3.5.4) |
| |  |  | | --- | --- | | = 956.07 |  | |  |

The stress cross section of the bolt used shall be equal or larger than the one estimated with (3.2.4). The calculated area is 956.07 is smaller than the cross-sectional area of M39 bolts (976 (Source [17] TB 8-1, p. 1130)so they seem to be sufficient. Therefore, the preliminary design of the bolted connection between the outer ring of the bearing and the hub shall consist of **145** bolts of the size **M39**.

## 3.6 Lubrication System

Most wind turbines typically require substantial repairs or full replacements of tribological components within 5 to 7 years. Among these, gearboxes, being costly and prone to high failure rates, significantly contribute to increased operating costs and the overall expense of wind energy production. To mitigate wear and enhance performance, various bearings within the turbine are lubricated using automatic lubrication systems, which are equipped with oil filters to maintain the required cleanliness levels. [18]

In wind turbines, pitch systems rely on specialized bearings to ensure precise motion and durability. Pitch systems use four-point contact ball-bearing, which are selected based on the rotor blade size. These bearings experience fluctuating loads and irregular operation, making proper lubrication essential to minimize wear, prevent overheating, and ensure smooth operation. Effective lubrication maintains bearing performance, improves reliability, and reduces maintenance costs, which are critical to turbine efficiency and longevity. [18]

Applying the right amount of lubricant at the right intervals is critical because both inadequate and excessive lubrication can lead to premature bearing failure and costly machine downtime. In fact, inadequate lubrication is responsible for approximately 36% of all bearing failures, underscoring the importance of accurate lubrication practices. [19] This chapter focuses on calculating the grease quantity and relubrication interval for bearings based on the guidelines provided in the SKF Maintenance Handbook. The calculations will ensure optimum lubrication to maximize bearing performance and life while minimizing downtime and maintenance costs.

|  |  |  |
| --- | --- | --- |
| **Aspect** | **Manual Lubrication** | **Automatic Lubrication** |
| |  | | --- | | **Mechanism** |  |  | | --- | |  |  |  | | --- | |  | | Grease applied manually; bearings must be stationary for safety. | Grease supplied via pump and distributor while turbine operates. |
| **Advantages** | Sufficient for raceways. | Allow continuous operation.  Ensure even grease distribution.  Improves safety by minimizing manual intervention. |
| **Disadvantages** | Gear lubrication may be inadequate (grease pushed out of contact areas).  Time-consuming and labor-intensive.  Uneven grease distribution. |  |

### 3.6.1 Manual vs. Automatic Lubrication

*Table 3.6.1 :- Manual Vs. Automatic Lubrication, 2025. Source:-*

### 3.6.2 Type of automatic lubrication systems

|  |  |  |
| --- | --- | --- |
| **System Type** | **Advantages** | **Disadvantages** |
| **Progressive Distributor** | - Simple, robust, economical. - Good monitoring. - Flexible sizes and configurations. - Supports various lubrication types. | -Residual pressure during downtime causes oil-thickener separation, leading to leaks and inconsistent lubrication. |
| **Injector Lubrication** | - Simultaneous dispensing. - Complete pressure release when idle. | - Expensive. - Requires more components. |

*Table 3.6.2 :- type of Lubrication system, 2025. Source:-*

### 3.6.3 Relubrication

Grease in bearings gradually loses its lubricating ability over time due to factors such as temperature, mechanical stress, aging, and contamination. Relubrication, the process of adding fresh grease, is essential to maintain optimum bearing performance. Proper relubrication depends on three critical factors [19]:

1. Selecting the correct type of grease.
2. Determining the correct quantity
3. Establishing an appropriate relubrication interval.

This section will not discuss the point of selecting the grease type as it is not within the scope of the report, and it depends mainly on the manufacturer's recommendations.

### 3.6.3.1 Correct quantity

The quantity of grease required for bearings is a crucial aspect of proper lubrication, as it ensures optimal performance and longevity. Excessive grease can lead to excessive heat and seal damage, while inadequate grease may result in increased wear. The appropriate grease quantity depends on the bearing size and the method of application, whether directly to the bearing or through the housing.

Replenishment:

* **When lubricating directly through the bearing center**:  
  Gp​=0.002DB
* **When lubricating from the side**:  
  Gp​=0.005DB

Where:

Gp​ = grease quantity for replenishment (g)

D = bearing outside diameter (mm)

B = bearing width (or height H for thrust bearings) (mm).

### 3.6.3.2 Relubrication Interval

The interval for relubrication represents a critical element in the maintenance of adequate lubrication within bearings. It establishes the periodicity with which new grease should be introduced into the bearing configuration, with the objective of preventing the degradation of the grease and guaranteeing optimal bearing functionality.

The relubrication interval (tf) can be calculated based on the speed factor (A), bearing factor (bf), and load ratio (C/P).

1. **Gather Required Information:**

* **Rotational speed (n)**: Speed of the bearing in revolutions per minute (RPM).
* **Bearing diameter (d and D)**: Inner and outer diameters of the bearing.
* **Load ratio (C/P)**: The ratio of the dynamic load rating C to the equivalent load P.
* **Bearing factor (bf)**: Depending on the bearing type and load conditions.
* **Operating temperature**: Generally, 70°C (160°F) for the standard calculation

1. **Speed Factor (A):**

The speed factor A is calculated as:

A =

Where:

* A = Speed factor in mm/min
* = Rotational speed in RPM
* ​ = Mean diameter of the bearing, calculated as:

1. **Bearing Factor (bf):**

The bearing factor depends on the type of bearing and its load conditions. For example:

* Deep groove ball bearings: bf = 1.0
* Cylindrical roller bearings: bf = 1.5 (for non-locating bearings)

1. **Load Ratio (C/P):**

The load ratio is calculated as: C/P

Where,

* C = Dynamic load rating of the bearing.
* P = Equivalent load on the bearing (in N).

A graph with lines drawn on it

Description automatically generated

1. **Use a Relubrication Interval Chart:**

Once all of the previous parameters are obtained, use a relubrication interval chart shown in Fig.21 to calculate the interval. These charts typically give relubrication intervals based on operating speed and load conditions.

Figure:-3.6. 1 Lubrication intervals Chart,2024.Source

Several buckets of grease

Description automatically generatedCalculating grease requirements for wind turbines ensures optimal performance and longevity of critical components. Using SKF LGEP 2/18 grease, the total annual requirement includes 15.63 kg for bearing raceway lubrication and 5 kg for pitch bearing gear lubrication, summing up to 20.63 kg/year. For a 7-month operational period, the monthly grease consumption is 1.72 kg, resulting in a requirement of 12.04 kg per bearing. With three pitch bearings, the total grease volume amounts to 36.12 kg. Considering the grease density of 0.9 kg/L, the total volume needed is approximately 40 liters, ensuring efficient lubrication over the specified period.

Figure:-3.6. 2 Lubricant for Pitch bearing and gear,2024.Source

## 3.6.4 Lubrication Pinions & Pump

Lubrication pinions are essential tools designed to apply grease uniformly across the entire tooth flank of gears. Made from durable materials such as aluminum or rubber-based composites, they ensure consistent and efficient lubrication. These pinions are commonly used in critical applications like yaw and pitch bearing teeth, where one or more lubrication pinions work together to maintain smooth operation and reduce wear. Lubrication Pinions Selection: LP2 Lubrication pinion from SKF.

A group of men working on a machine

Description automatically generated

Figure:-3.6. 3 Lubricant pinion and Pitch pinion assembly,2024.Source

A blue and white sheet of paper with text

Description automatically generated

Figure:-3.6. 4 Technical data,2024.Source

A diagram of a diagram of a diagram

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Figure:-3.6. 5 Lubricant pinion,2024.Source

The **Compact P 653 M** is a powerful and robust automatic lubrication pump designed for high-demand applications. It features a reservoir size of up to **100 liters (26.4 gallons)**, providing ample capacity for extended lubrication cycles. This pump is capable of driving up to **three pump elements**, ensuring efficient and reliable lubrication across multiple points. Its durability and performance make it ideal for industrial systems requiring consistent and automated grease distribution.

A blue and white technical data

Description automatically generated with medium confidence

Figure:-3.6. Technical data for Lubrication Pump,2024.Source

## 3.7 Spinner

A spinner is a structural element mounted on a wind turbine's hub, covering both the hub and the inner cylindrical sections of the rotor blades. Its design, often spherical or paraboloidal, is streamlined to reduce aerodynamic drag and improve efficiency.

Spinners are typically made from lightweight, durable materials like carbon fiber or fiberglass. These materials provide strength, resistance to environmental factors such as corrosion, and ensure the spinner maintains its structural integrity while keeping the turbine lightweight.

**Requirements of Spinner**

1. **Improved aerodynamics and performance**

A spinner, shaped either spherically or paraboloidal, promotes smooth airflow around the hub and directs wind towards the profiled sections of the rotor blades. This design minimizes turbulence and maximizes the efficiency of wind energy capture.

1. **Improved lifespan of Components**

It protects the hub, internal components, and blade attachments from environmental factors such as debris, rain, and corrosion, enhancing their longevity and reliability.

1. **Enhance Safety of Technician**

The spinner improves technician safety by enclosing the hub and internal components, limiting exposure to moving parts and harsh weather conditions. Its streamlined design reduces risks by preventing debris and ice buildup. Furthermore, it facilitates easier and safer access for maintenance and inspections.

1. **Aesthetics**

The spinner improves the aesthetics of wind turbines by providing a sleek, streamlined design that integrates smoothly with the turbine's overall look. Its clean and modern appearance reduces visual disruption, making turbines more visually appealing in both urban and scenic settings.



Figure:-3.7. Spinner of Optimus Shakti[Own Creation]

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|  |  |
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# Appendix