Fatigue analysis of a generic 7.5 MW reference rotor hub

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Master Thesis

Fatigue Analysis of a Generic 7.5 MW Reference Rotor Hub

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ABSTRACT

Abstract

In recent decades, there has been a tremendous increase in the utilization of wind energy to generate electricity. In order to cater to the rising power demand, the capacities of wind turbines are continuously increasing. As the capacity of the wind turbines increase, so does the problems associated with them. To provide solutions to such problems, it is essential to have well documented literature on all components of the wind turbine. There has been a lot of research carried out on some parts of the wind turbine. However, it has been observed that there is inadequate information available on rotor hubs. In this thesis, an effort is made to add literature to the field of fatigue analysis of rotor hub. To accomplish this goal, a generic 7.5MW reference rotor hub is modelled and subjected to extreme load cases followed by the fatigue life cycle analysis by applying the Damage Equivalent Loads (DELs). The design of the model is then optimized using the initial results in order to obtain a rotor hub with minimum size and weight, which is able to withstand the extreme loads and avoid failure due to fatigue for 1×10^7 cycles.

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LIST OF SYMBOLS

DELs	Damage Equivalent Loads	
HAWTs	Horizontal Axis Wind Turbines	
VAWTs	Vertical Axis Wind Turbines	
P_{wind}	Power present in the wind	W
ρ	Density	Kg/m ³
A	Cross sectional area	m ²
v	Velocity of the wind	m/s
E_{ex}	Extracted kinetic energy	J
$\dot{m}$	Mass flow rate	Kg/s
$c_{p,Betz}$	Power co-efficient	
σ_m	Mean stress	N/m ²
σ_{max}	Maximum stress	N/m ²
σ_{min}	Minimum stress	N/m ²
σ_r	Range of stress	N/m ²
S	Stress amplitude	N/m ²
R	Stress ratio	
σ_b	Tensile strength	N/m ²
σ_w	Fatigue strength	N/m ²
σ_{wk}	Fatigue strength after considering surface roughness and stress gradient	N/m ²
F_{ok}	Total influencing factor	
F_o	Surface roughness effect	
β_k	Notch factor	
R_z	Surface roughness	μm
X	Stress gradient	1/mm
α_k	Stress concentration factor	
σ_A	Stress amplitude at the knee of the SN curve	N/m ²
F_m	A factor which takes mean stress effect into consideration	
M	Mean stress sensitivity	
m	Slope of the S-N curve	

N_D	Number of cycles to failure	cycles
σ_1	Upper limit of the fatigue life line	N/m ²
σ_a	Alternating stress	N/m ²
σ_m	Mean stress	N/m ²
S_e	Fatigue strength	N/m ²
S_u	Ultimate strength	N/m ²
S_y	Yield strength	N/m ²
F_x	Component of force in X direction	N
F_y	Component of force in Y direction	N
F_z	Component of force in Z direction	N
M_x	Component of Moment in X direction	Nm
M_y	Component of Moment in Y direction	Nm
M_z	Component of Moment in Z direction	Nm
F_B	Force in the blade co-ordinate system	N
M_B	Moment in the blade co-ordinate system	Nm

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OVERVIEW

Chapter 1 introduces the reader to the world of wind energy, briefly explains the procedure involved and results expected from the work. It also tells the reader about the motivation behind carrying out this particular work.

Chapter 2 gives detailed information about the different components used in the wind turbines and explains their working.

Chapter 3 explains the basic principles of fatigue and the various approaches that are used in order to determine the fatigue life of components. Along with this, it also deals with the factors affecting the fatigue life of components and the ways to improve it.

In chapter 4, an effort is made to compile the available literature in the field of rotor hub design.

Chapter 5 deals with the procedure involved in the detailed design of the generic 7.5MW reference rotor hub in parametric form.

The prerequisites for setting up the analysis, mesh sensitivity tests and the process to manufacture/ synthesize an S-N curve for the material EN-GJS-400-18LT is explained in chapter 6

Chapter 7 throws light on the results which are obtained from the initial analyses and the modifications done to the model during the optimization process. The detailed design and analyses of the optimized models are explained in this chapter.

Chapter 8 concludes this report by briefly summarizing the entire work and gives an insight into the work that can be carried out further.

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Chapter 1

INTRODUCTION

We live in a world wherein electricity has become a basic need for every human being. There are many ways of generating electrical power varying from using coal to biogas. But there are certain industries which have been receiving a lot of attention due to their non-polluting nature and being renewable. One such industry is the wind power industry.

In early days, wind energy was harnessed by making use of wind mills which converted power in the wind directly to mechanical energy, which in turn was used to grind grains and pump water. At the end of 19th century, people started to generate electricity from the wind which laid the foundation for the modern day wind turbines. The first recorded use of wind turbine to generate electricity was at Cleveland, Ohio in 1888. [1]

In recent decades, there has been an increase in the use of wind turbines to generate electricity. The capacities of the wind turbines are continuously increasing, from a modest 12kW wind turbine which was installed at Cleveland, Ohio in 1888 to an 8MW Vestas V164 prototype wind turbine which is installed offshore in 2015.

As the capacities of the wind turbines increase, so does the problems associated with them. In order to deal with such problems, it is necessary to have well-documented literature on all components of wind turbine. However, it was observed that although there is abundant literature available on the design and analyses of rotor blades and other components, the same cannot be said about rotor hubs. Rotor hubs are components which are used to connect the rotor blades to the generator (via a drive train in case of geared wind turbines).

The current project focuses on developing a generic 7.5 MW reference rotor hub which is capable of withstanding the Damage Equivalent Loads (DELs) and working in extreme situations without undergoing failure for 1×10^7 cycles. Along with this, an attempt is made to compile the literature available on rotor hubs and its fatigue properties.

The tasks include modelling a 7.5MW reference rotor hub using NX as the modelling software. Once, the model is created, it is then imported to ANSYS workbench where the model is subjected to extreme load situations. If the model withstands the extreme loads, it will be further subjected to Damage Equivalent Loads (DELs), if not; the model has to be modified in such a way that it withstands the extreme loads.

The models subjected to DELs are analyzed for fatigue failure and is expected to withstand the loads for 1×10^7 cycles. The model thus obtained is further optimized, in order to get a rotor hub with minimum weight which is able to sustain the extreme conditions and avoid failure due to fatigue for 1×10^7 cycles.

Chapter 2

WIND TURBINE COMPONENTS

2.1 Brief Description of Wind Turbines

Wind turbines generate electricity by harnessing the power of the wind. The energy in the wind turns the blades around a rotor and the rotor is connected to a generator (via a gearbox in case of geared wind turbines) generating electricity in a cleaner way compared to the conventional thermal power plants [2] .

Modern day wind turbines can be classified into 2 basic groups

1. Horizontal Axis Wind Turbines (HAWTs).
2. Vertical Axis Wind Turbines (VAWTs).

VAWTs have their shaft mounted on a vertical axis perpendicular to the ground and do not have to align themselves according to the wind direction but, as their efficiency is less when compared to that of HAWTs they are not widely used [3].

In HAWTs, the rotor is rotating around a horizontal axis. They need to align themselves constantly in the direction of the wind. The technology used in HAWTs have grown by leaps and bounds in the past few decades as their advantages (higher efficiency compared to VAWTs) outweigh their disadvantages, making them the most widely used wind turbines both onshore as well as offshore [3].

The HAWTs can be classified into upwind and downwind turbines based on the position of the rotor with respect to the tower.

Most of the wind turbines make use of upwind configuration. In such a configuration, the rotor is located in front of the tower in the direction of the wind and makes use of an active yaw system to align the rotor in the direction of the wind [3].

Downwind turbines have their rotor behind the tower, when looked in the direction of the wind and they allow the turbine to have free yaw. This configuration has its rotor blades

coned downwards, resulting in the centrifugal moments counteracting the moments produced due to thrust, reducing the root flap bending moments. However, the flow reaching the rotor is strongly disturbed by the tower and causes turbulence which increases the fatigue loads acting on the blades and hence are not generally preferred. Figure 1 shows an upwind and downwind turbine.

Figure 1 Horizontal axis upwind and downwind turbine. [3]

The main parts of a wind turbine are:

1. Rotor

- Rotor blade.
- Rotor hub.
- Pitch system (aerodynamic brake).

2. Drive train

- Rotor shaft(s).
- Bearings.
- Brake.
- Gearbox (not present in direct drive wind turbines).
- Generator.

3. Yaw system

- Yaw bearing.
- Yaw drive(s).

4. Support Structure

- Foundation.
- Tower.
- Nacelle.

Figure 2 shows the schematic layout of a horizontal axis upwind wind turbine.

Figure 2 Schematic layout of a horizontal axis wind turbine. [2]

2.2 Rotor

2.2.1 Rotor Blades

The conversion of kinetic energy of wind to the mechanical rotary motion occurs due to rotor blades. A blade has an airfoil shape which changes the airflow streamlines and causes a pressure difference. This difference of pressure over the blade creates a lift force which in turn creates torque in the wind turbine rotor. A symmetrical airfoil creates no lift force when the angle of attack is zero. However, if the angle of attack is more than zero, lift occurs as a consequence of the pressure difference between the two surfaces. The angle of attack depends on the wind velocity, the rotational speed, and the distance from the blade root. The flow of airstream over an airfoil is shown in figure 3.

Figure 3 Flow of air over a wind turbine blade (aerofoil shape). [16]

2.2.2 Rotor hub

A rotor hub is used to connect the rotor blades to the drive train. It is discussed in detail in chapter 4.

2.2.3 Blade pitch system

One of the ways to influence the input power is by varying the aerodynamic angle of attack. It can be done by adjusting the rotor blade pitch angle and is termed as pitch regulation. Pitch regulation system consists of an active control system which senses

the blade position, measures the rotor speed, power output and uses this measured data to make appropriate changes to pitch the blade. [4] Depending on the incoming wind velocity, there is an optimal angle of attack which allows the rotor to deliver maximum power to the main-shaft [2]. Hence, the blade pitch system is used to pitch the blades to the optimum angle for the corresponding wind speed in order to increase the generated power. Currently, wind turbines make use of collective pitch systems which feathers all the blades with a same angle. Individual blade pitching can be applied in future turbines to improve the power production. [2] Figure 4 shows the hub of an electrical pitch control system of a 1.5MW wind turbine.

Figure 4 Hub of a pitch-controlled 1.5MW wind turbine with electrical gear motors (REpower) [3]

2.3 Drive Train

The drive trains are used in geared wind turbines to transfer power from the rotor to the generator and consist of mechanical components such as bearings, shafts and gears. The drive train consists of a main shaft which is connected to the rotor (hub and blades). The main shaft supports the rotor. The rear of the main shaft is connected to the slow-rotating side of the gearbox. The gearbox increases the rotational speed and is connected to a high speed shaft which in turn is connected to the electrical generator. The generator has a disc brake which can be used to keep the turbine in stop position. [2] In case of direct drive wind turbines, moment bearings are used to connect the rotor to the generator. The schematic diagram of a drive train used in geared wind turbines is shown in figure 5.

Figure 5 Figure showing the components of the drive train [15]

2.4 Yaw System

The yaw drive systems are used to orient the wind turbine in the direction of the wind. There are two types of yaw systems [5].

1. Passive yaw systems (autonomous yawing).
2. Active yaw systems.

The Downwind turbines make use of autonomous / passive yawing systems. If the wind direction is not parallel to the rotor axis, the force of the wind acting on the rotor causes a yaw moment around the tower axis and orients the rotor in the direction of the wind. [3]

The active yaw system used in upwind turbine consists of a yaw motor which is controlled by the wind turbine control system. When the wind turbines are producing electricity, these systems continuously keep the nacelle headed in the direction of the incoming wind. Even a small deviation from the correct heading will reduce the power production and increase mechanical wear on all moving parts. The control system is thus connected to a set of sensors which monitors the changing speed and direction of the wind at all times, making it possible to activate the yaw system quickly. [5] Figure 6 shows an electrical yaw drive system.

Figure 6a) Section of a yaw drive with multi-stage planetary gear; 6b) Electrical yaw drive system, yaw brakes and yaw angle [3]

2.5 Support Structure

2.5.1 Foundation

The foundation of a wind turbine must be sufficient to keep the turbine upright and stable under the most extreme design conditions. Wind turbines generally make use of a foundation made from concrete blocks. The weight of the concrete is chosen such that it provides resistance to overturning in all conditions. Figure 7 shows an image of the flat foundation used in a wind turbine. [5]

Figure 7 Flat foundation of a wind turbine. [3]

2.5.2 Tower

Towers are support structures which are used to raise the wind turbine up in the air so that its blades can clear the ground. Taller towers enable the turbine to capture more energy from the wind and generate more electricity, as the wind speed increases with height. Generally, the tower height should not be less than 24m as the wind is turbulent and has less speed closer to the ground [5]. However, an increase in the tower height will increase the weight of the tower and thus its cost [2]. Therefore, it is necessary to optimize the wind turbine tower based on the requirement.

There are three kinds of towers which are used in horizontal axis wind turbines:

1. Cantilevered pipe (tubular tower)
2. Free standing lattice (truss)
3. Guyed lattice or pole.

The cantilevered pipe or tubular tower is the most commonly used tower for HAWTs as it does not rely on bolted connections which have to be torqued periodically. Also, tubular towers provide a safe climbing area to access the machines. Figure 8 shows a tubular tower.

Figure 8 Flat foundation of a wind turbine. [3]

2.5.3 Nacelle

The Nacelle is located at the top of the tower. It is connected to the rotor and houses several components such as generator and the drive train. [2]

2.6 Extraction of Power from Wind

The power present in the wind is converted into mechanical power of the rotor by decelerating the flowing air mass. It is not possible to completely convert the energy present in wind into mechanical power of the rotor, as it requires decelerating the mass flow to zero. Such deceleration would result in blockage of the cross sectional area for the air masses following it.

The power present in the wind that flows at a velocity v through an area A is given by

$$P_{wind} = \frac{1}{2} \rho A v^3 \quad (2.1)$$

In order to determine the maximum power that can be extracted from the wind, Betz assumed a homogenous air flow with velocity v_1 which is reduced as it passes through the turbine to a velocity v_3 far downstream. He assumed a stream tube with divergent streamlines for reasons of continuity; it is shown in figure 9

$$\rho v_1 v A_1 = \rho v_2 v A_2 = \rho v_3 v A_3 \quad (2.2)$$

Figure 9 Air flow through the rotor of a wind turbine, divergence of the stream tube resulting from the flow deceleration [3]

The air density is assumed to be constant. The extracted kinetic energy is the difference between the upstream and downstream kinetic energy.

$$E_{ex} = \frac{1}{2} m (v_1^2 - v_3^2) \quad (2.3)$$

The power extracted from the wind turbine is,

$$\dot{E}_{ex} = \frac{1}{2} \dot{m} (v_1^2 - v_3^2) \quad (2.4)$$

If the wind speed is not reduced then, $v_3 = v_1$, hence no power would be extracted, whereas, if $v_3 = 0$, then the mass flow rate would be zero leading to the congestion of the tube, hence once again no power will be extracted.

Therefore, we need to choose a value for v_3 between $v_3 = 0$ and $v_3 = v_1$. Let this value be v_2 . The mass flow rate will then be,

$$\dot{m} = \rho A v_2 \quad (2.5)$$

The only plausible assumption for the value of v_2 is

$$v_2 = \frac{v_1 + v_3}{2} \quad (2.6)$$

This assumption is validated by Froude- Rankine theorem.

If the mass flow rate equation (i.e. equation 2.5) and the rotor plane velocity equation (i.e. equation 2.6) are inserted into the equation (2.4) we get,

$$\dot{E}_{ex} = \frac{1}{2} \rho A v_1^3 \left[\frac{1}{2} * \left(1 + \frac{v_3}{v_1} \right) * \left(1 - \left(\frac{v_3}{v_1} \right)^2 \right) \right] \quad (2.7)$$

The term $\left[\frac{1}{2} * \left(1 + \frac{v_3}{v_1} \right) * \left(1 - \left(\frac{v_3}{v_1} \right)^2 \right) \right]$ is the power co-efficient represented by c_p thus, power in the wind is multiplied by a factor c_p which depends on the ration of $\frac{v_3}{v_1}$

$$c_{p,Betz} = \frac{16}{27} = 0.59 \quad (2.8)$$

Thus, the maximum power that can be extracted by an ideal turbine is nearly 60% of the total power present in the wind.

In this chapter an effort was made to give a brief description of the different components used in the wind turbine. All these components are subjected to fatigue loading due to the stochastic nature of the wind. Hence, it becomes necessary to gain some background information about fatigue and factors which affect the fatigue life in order to proceed with the fatigue analysis.

Chapter 3

FATIGUE ANALYSIS

3.1 Introduction to Fatigue loads

A characteristic feature of wind turbine is that, it experiences operational stress due to vibrations. A wind turbine in its service life of 20 years, experiences load cycles of the order 10^8 cycles. Hence, fatigue analysis of wind turbines becomes important. [3] This chapter familiarizes the readers with the concept of fatigue, failure mechanisms and the factors affecting the fatigue life of components in order to have basic understanding about them.

“The term metal fatigue refers to the gradual degradation and eventual failure that occurs due to loads which vary with time, and which are lower than the static strength of the metallic specimen, component or structure concerned” [6]. Fatigue is considered to be the single largest cause of failure in metals as it is estimated to comprise about 90% of all metallic failures [7].

The process of fatigue failure occurs by the initiation and propagation of cracks, and normally the fracture surface is perpendicular to the direction of the applied stress [6]. The applied stress can be axial, flexural or torsional in nature. Generally there are three different fluctuating stress–time modes.

1. Reversed stress cycle:

It is represented by a regular and sinusoidal time dependent curve, wherein the amplitude is symmetrical about a mean zero stress level, alternating from a maximum tensile stress ($\sigma_{\max}$) to a minimum compressive stress ($\sigma_{\min}$) of equal magnitude [7].

2. Repeated stress cycle:

In this case, the maxima and minima are asymmetrical relative to the zero stress level. [7]

3. Random Stress cycle:

The stress level varies in both amplitude and frequency. [7]

In reality, all the stress cycles are random in nature; hence we make use of mathematical models to convert these random stress cycles into reversed stress cycles in order to determine the fatigue life of components. Figure 10 shows the graphical representation of the different stress-time modes

Figure 10 Graphical representations of the different stress-time modes. [7]

The parameters which are used to characterize fluctuating stress cycle are as follows.

Mean stress (σ_m)

$$\sigma_m = \frac{\sigma_{max} + \sigma_{min}}{2} \quad (3.1)$$

Range of stress (σ_r)

$$\sigma_r = \sigma_{max} - \sigma_{min} \quad (3.2)$$

Stress amplitude (S)

$$S = \frac{\sigma_r}{2} = \frac{\sigma_{max} - \sigma_{min}}{2} \quad (3.3)$$

Stress ratio (R)

$$R = \frac{\sigma_{min}}{\sigma_{max}} \quad (3.4)$$

Fatigue cycles can be classified into low cycle fatigue (less than 10^4 cycles) and high cycle fatigue greater than 10^4 or 10^5 cycles [7]. Based on the number of cycles to failure, either strain based approach or stress based approach is chosen for the analysis.

3.2 Strain Based Approach

The strain-life method is based on the observation that in any component the response of the material in critical locations is strain or deformation dependent. In the strain life approach, the plastic strain or deformation is directly measured and quantified. At high cycles, where plastic strain is negligible, stress and strain are easily related. Then, the strain-life and stress-life approaches are the same [8]. The strain life approach is not discussed in detail as it is not used in the current work.

3.3 Stress Life Approach

In high cycle fatigue situations, the performance of a material is normally characterized by an S-N curve, also called as Wöhler curve. It is a plot of the stress amplitude (S) versus the number of cycles to failure (N) for each of the specimens and is usually represented on a log-log scale.

For some ferrous and titanium alloys as shown in figure 11, the S-N curve becomes horizontal at higher N values; or there is a limiting stress level called the fatigue limit, below which fatigue failure will not occur. This fatigue limit represents the largest value of fluctuating stress that will not cause failure for essentially an infinite number of cycles. [7]

Most nonferrous alloys do not have a fatigue limit, which means that the S-N curve continues its downward trend at increasingly greater N values. Thus, fatigue will ultimately

occur regardless of the magnitude of the stress. For these materials, the fatigue response is specified as fatigue strength, which is defined as the stress level at which failure will occur for some specified number of cycles. [7]

Another important parameter that characterizes a material’s fatigue behavior is fatigue life. It is the number of cycles to cause failure at a specified stress level, as taken from the S-N plot. [7].Figure 12 shows a plot of S-N curve for non-ferrous metals.

Figure 11 S-N plot for ferrous & titanium alloys [7].

Figure 12 S-N plot for non-ferrous metals [7].

One of the major disadvantages of S-N approach is that, it treats all strains as elastic and does not take into account the true stress-strain behavior. This simplifying assumption is valid only if the plastic strains are small [8]. The S-N approach should not be used to estimate lives below 1000 cycles as the plastic strains are high at such low cycles due to the high load levels and it violates the assumption made. Hence, when carrying out low cycle analysis, strain based approach is more appropriate [7].

3.3.1 Synthetic stress-life (S-N) curve

Majority of wind turbine rotor hubs are made up of the material EN-GJS-400-18-LT but there is very little information available on the S-N curve and fatigue properties of it. Hence we need to manufacture or synthesize an S-N curve in order to carry out fatigue analysis on the rotor hub. H. GUDEHUS et al. in their book [9] has suggested a procedure to synthesize S-N curve; this procedure is explained below.

The first step in synthesizing the S-N curve is to obtain the tensile strength (σ_b) of the material; it can be obtained from the tensile test R_m .

$$\sigma_b = R_m \quad (\text{N/m}^2) \quad (3.5)$$

If the tensile test data is not available then we can obtain the tensile strength of the material from the handbook EN 1563:2011 [10] i.e. $R_{m,min}$. the value obtained from the handbook must be multiplied by 1.06 (nominative value as prescribed by GL).

$$\sigma_b = 1.06 * R_m \quad (\text{N/m}^2) \quad (3.6)$$

The fatigue strength of the material σ_w in N/m^2 for different types of cast iron is given by the following formulae,

$$\sigma_w = 0.27 * R_m + 85 \quad - \text{ for GS(Cast Iron)} \quad (3.7)$$

$$\sigma_w = 0.27 * R_m + 100 \quad - \text{ for GGG(Nodular cast Iron)} \quad (3.8)$$

$$\sigma_w = 0.27 * R_m + 110 \quad - \text{ for GTS (Malleable cast Iron)} \quad (3.9)$$

The fatigue strength which is obtained from the above mentioned formulae 3.7, 3.8 & 3.9 is for a smooth fatigue specimen without any stress gradient. The fatigue

strength of the material after considering the surface roughness effect F_o and stress gradient effect β_k is given by:

$$\sigma_{wk} = \frac{\sigma_w}{F_{ok}} \quad (\text{N/m}^2) \quad (3.10)$$

Where F_{ok} is the total influencing factor and is given by

$$F_{ok} = \sqrt{\beta_k^2 - 1 + \frac{1}{F_o^2}} \quad (3.11)$$

The value of F_o and β_k is given by the following formulae

$$F_o = 1 - 0.22 * (\log R_z)^{0.64} * (\log \sigma_b) + 0.45 * (\log R_z)^{0.53} \quad (3.12)$$

$$n = 1 + 0.32 * X^{0.77} \quad (3.13)$$

$$\beta_k = \frac{\alpha_k}{n} \quad (3.14)$$

$$X = \frac{1}{\text{thickness}} \left(\frac{1}{\text{mm}} \right) \quad (3.15)$$

Where,

R_z is the surface roughness (μm).

X is the stress gradient (1/mm).

β_k is the notch factor.

α_k is the stress concentration factor.

The stress amplitude σ_A at the knee of the SN curve is given by

$$\sigma_A = \sigma_{wk} * F_m \quad (\text{N/m}^2) \quad (3.16)$$

Where, F_m is a factor which takes the mean stress effect into consideration. F_m for Stress ratio, $R=-1$ is 1 and for $R=0$ it is given by

$$F_{m,R=0} = \frac{1}{1 + M} \quad (3.17)$$

Where, M is the mean stress sensitivity and can be calculated using

$$M = 0.00035 * \sigma_b + 0.08 \quad (3.18)$$

The slopes of the synthesized SN curve are given by,

$$m_1 = \frac{5.5}{F_{ok}^2} + 6 \quad (3.19)$$

$$m_2 = 2 * m_1 - 1 \quad (3.20)$$

Number of load cycles at the knee of the synthetic S-N curve N_D is given by

$$N_D = 10^{6.8 - \frac{3.6}{m_1}} \text{ cycles} \quad (3.21)$$

The upper limit of the fatigue life line σ_1 is given by

$$\sigma_1 = R_m * \left(\frac{1-R}{2} \right) \text{ (N/m}^2\text{)} \quad (3.22)$$

The S-N curve synthesized by using the above mentioned formulae is shown in figure 13.

Figure 13 Representative graphical presentation of a synthetic S-N curve [11]

3.4 Failure Criteria

Components that are loaded under pure oscillation fail (undergo fracture) when their stresses reach their materials fatigue limit while the ones loaded under pure static loads fail at the material yield limit. For components that are loaded by a combination of oscillatory and static loads, the following criteria provide a way to calculate the failure limit [8].

Soderberg Relation [7]

$$\frac{\sigma_a}{S_e} + \frac{\sigma_m}{S_y} = 1 \quad (3.23)$$

Goodman relation [7]

$$\frac{\sigma_a}{S_e} + \frac{\sigma_m}{S_u} = 1 \quad (3.24)$$

Gerber relation [7]

$$\frac{\sigma_a}{S_e} + \left(\frac{\sigma_m}{S_u}\right)^2 = 1 \quad (3.25)$$

Where,

σ_a = Alternating stress in N/m².

σ_m = Mean stress in N/m².

S_e = Fatigue strength in N/m².

S_y = Yield strength in N/m².

S_u = Ultimate strength in N/m².

Figure 14 shows the plot comparing the three failure criteria.

Figure 14 Plot comparing the three failure criteria [6]

In the present work, Goodman criterion is used as the failure criteria as it is suitable for materials which do not have definite yield strength.

3.4.1 Palmgren-Miner linear damage rule (Miner’s rule)

Initially, fatigue tests were carried out at constant amplitude fatigue loading and designers faced the problem of predicting the life of components subjected to a wide range of variable amplitude load histories by using this fatigue data [12]. This problem was addressed by Palmgren (1924) who conducted an experiment to predict the life of ball bearings. He assumed the damage caused to the ball bearings by the loading to be accumulated linearly with the number of revolutions. Later Langer (1937) and Miner (1945) suggested that the fatigue damage that occurs at a particular stress level is a result of damage that is accumulated linearly with the number of stress cycles. This is called as Palmgren-Miner linear damage rule [6]. This rule requires three main classes of input data: the load spectra, material properties, and inflow characterization.

The damage law may be written in the following form:

$$\sum_i^M \frac{n_i}{N_i} = \frac{n_1}{N_1} + \frac{n_2}{N_2} + \frac{n_3}{N_3} \dots \dots \dots + \frac{n_M}{N_M} = < 1 \tag{3.26}$$

Where n_1 is the number of stress cycles at σ_1 and N_1 is the number of cycles that it takes to fail at σ_1 and the summation has to be less than 1.

3.5 Cycle Counting

The objective of all kinds of cycle counting methods is to compare the effect of variable amplitude load histories to fatigue data and fatigue curves obtained with constant amplitude load cycles. A good cycle counting method should count a cycle with the range from the highest peak to the lowest valley and seek to count other cycles in a manner that maximizes the ranges that are counted. In other words, intermediate fluctuations are less important than the overall differences between highest and lowest points [13]. The different kinds of cycle counting method which are in use are:

1. Range counting.
2. Rain flow counting.
3. Level cross counting.
4. Peak counting.
5. Narrow band approximation.

The most commonly used cycle counting method/algorithm is the rain flow counting algorithm which is discussed in detail in this section. [14]

Rain flow counting method helps us to apply miners rule to access fatigue life of a structure subjected to complex loading by converting a spectrum of varying stresses into a set of simple stress reversals. In this method, the water is imagined to flow down the stress-time history and it was first proposed by Endo. The procedure for rain flow counting algorithm is as follows [14].

The flow starts at the beginning of the time series, then at the inside of the peaks in the order of the applied peaks. The flow stops when it encounters a flow from a higher level or a point of opposite peak which is arithmetically greater or equal to the point from which it started or when it reached the end of the time series. Each separate flow is counted as a half cycle and there will always be a complimentary half cycle of opposite sign except for the flow which starts at the beginning of the time series or at the end of it. [14]

The stress ranges are identified and the corresponding number of cycles is determined. The rain flow counting diagram is shown in figure 15.

Figure 15 Rain flow counting diagram [13].

3.6 Mechanisms of Metal Fatigue Failure

The process of fatigue failure occurs in three steps [7]:

1. Crack initiation.
2. Crack propagation.
3. Failure which occurs rapidly once the crack reaches its critical size.

3.6.1 Fatigue crack initiation

The cracks which are associated with fatigue failure most of the times initiate on the surface of a component, at the point of stress concentration. The most common crack initiation sites are keyways, sharp fillets, threads, surface scratches and dents. In ductile materials, microscopic surface discontinuities are produced due to cyclic loading and these discontinuities acts as stress raisers and therefore, as crack initiation sites. [7]

Fatigue Crack initiation in the case of ductile materials occurs as a result of reversed plasticity that takes place within a grain on a scale of 10^{-3} mm. As surface grains are the weakest, they deform plastically and produce a micro crack within a grain. These

micro cracks can be produced at stresses which are much lower than the tensile yield strength. Resistance to crack initiation strongly depends on the surface roughness, residual stresses and the environment. [6]

3.6.2 Fatigue crack propagation

Once the crack is initiated, it advances incrementally with each stress cycle. The fracture surface formed during the process of crack propagation is identified by two kinds of markings; striations and beachmarks. These markings indicate the crack tip position at some point and they appear as concentric ridges that expand away from crack initiation site. [7]

The Beachmarks are macroscopic in nature and can be seen with naked eye. They are usually found on components that experience interruptions during the crack propagation stage for example, a component that operated only during specific hours in a day. Each beachmark represents the crack growth over a period of time. Figure 16 shows the beachmark ridges that were formed in a rotating steel shaft which was subjected to fatigue failure.

Figure 16 Beachmark ridges that were formed in a rotating steel shaft which was subjected to fatigue failure. [7]

Striations on the other hand are microscopic in nature and can be seen only with the help of an electron microscopic (such as SEM, TEM). Each striation represents the distance advanced by the crack front during a single load cycle. [7] Figure 17 shows the striations as seen from a transmission electron fractograph in aluminum.

Figure 17 Striations as seen from a transmission electron fractograph in aluminum. [7]

3.7 Factors that Affect Fatigue Life

Some of the factors which affect the fatigue life of the components are discussed in detail below.

3.7.1 Size effect

Fatigue failure in materials occurs as a result of initiation of the crack, propagation of crack until it reaches a critical size followed by fracture. The process of crack initiation in a material occurs at the weakest link and the possibility of finding weak spots increases with the increase in the material volume. Hence, there is a greater possibility of initiating a fatigue crack in larger components. [8]

3.7.2 Surface finish

During machining operations, grooves and small scratches are introduced into the surface of the work piece by cutting tool action. These surface markings can act as the crack initiation sites and limit the fatigue life. In order to avoid this, the surface finish of the component has to be improved. [7, 8]

3.7.3 Surface treatments

One of the ways to improve the fatigue life of the material is to introduce residual compressive stresses in the outer surface layer. When an external tensile stress is applied on the surface, it will be partially nullified by the residual compressive stresses induced and the probability of crack initiation and thus, fatigue failure is reduced. Residual compressive stresses are normally introduced into the ductile materials by the process of shot peening. Shot peening is a process in which the surface to be treated is bombarded with small hard particles (shot) with diameters in the range of 0.1 to 1mm at high velocity. The deformation caused by the shots will induce compressive stresses to a depth of one quarter to one half of the shot diameter. [7]

The other surface treatment technique which can be used to improve both the surface hardness and fatigue life of metals is case hardening. It is a process in which the surface to be treated is exposed to a carbonaceous (carburizing) or nitrogenous (nitriding) atmosphere at elevated temperature resulting in the formation of a carbon rich or nitrogen rich layer on the outer surface. This layer normally has a depth of about 1mm and is harder when compared to the core material and thus reduces the risk of crack initiation on the surface. [7]

3.7.4 Environment

There are two types of environment assisted fatigue failure i.e. thermal fatigue and corrosion fatigue.

Thermal fatigue is a result of the fluctuating thermal stresses that occur at elevated temperatures. A structural member experiences thermal stresses only when it is constrained from free expansion / contraction. The magnitude of thermal stresses σ is determined by the relation given below

$$\sigma = E\alpha_t\Delta T \text{ N/m}^2 \quad (3.27)$$

Where,

α_t = Thermal expansion co-efficient in $\text{m/m}^{\circ}\text{C}$

E = Modulus of elasticity in N/m^2

ΔT = Change in temperature in $^{\circ}\text{C}$

Thermal fatigue can be avoided by reducing or eliminating the thermal stresses which can be done by removing the restraint or at least reducing the restraint source thus allowing for the free expansion of the structural member.

Failure that occurs by the combined action of chemical attack and cyclic stresses is termed as corrosion fatigue. Small pits may be formed on the surface of the material due to the chemical reaction that takes place between the environment and the material. These pits act as a stress concentration points and therefore, crack initiation sites. Also, the rate at which the crack propagates is enhanced by the corrosive environment. There are certain measures which can be applied to prevent the fatigue failure of component due to corrosive environment such as coating the material with a corrosion protective layer and selecting a better corrosive resistant material. [7]

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Chapter 4

STATE OF THE ART

In this section an effort is made to compile the literature available in the field of wind turbine rotor hub design.

The Rotor hubs are considered to be the first component of the mechanical drive train system although they are a part of the rotor, due to their structure and function. They are one of the highly stressed components of the wind turbine with all the rotor forces and moments acting on them. [3]

4.1 Types of Rotor Hubs

The rotor hubs which are used in modern day wind turbines are of three types.

1. Rigid hubs.
2. Teetering hubs.
3. Hubs from hinged blades.

The schematic diagram of the different kinds of rotor hubs are shown in the figure 18

Figure 18 Schematic diagrams of different kinds of rotor hub. [3]

4.1.1 Rigid hubs

The Rigid hubs are the most commonly used for wind turbines with three or more blades. The design of a rigid hub is such that all the major parts are in a fixed position relative to the main shaft. Rigid hubs allow only pitching motion of the blades and all other motions of the blades are restricted. [5]

The main body of a rigid hub is a casting or weldment to which the blades are attached and this in turn is connected to the main shaft. A rigid hub must be strong enough to withstand the loads that arise due to the aerodynamic loads acting on the blades and the dynamically induced loads, such as those due to rotation and yawing. Figure 19 shows a rigid hub.

Figure 19 Rigid hub. [5]

4.1.2 Teetering hubs

Teetering hubs are commonly used in two bladed wind turbines. A teetering hub helps to reduce the loads which are caused due to the aerodynamic imbalances and loads due to dynamic effect caused by the rotation of the rotor or turbine yawing. Teetering hubs are more complex in nature when compared to rigid hubs. They consist of two main parts; the main hub body and a pair of trunnion pins along with the bearings and dampers.

The hub body is usually made up of steel or cast iron. At either ends of the hub body there are points for blade attachment. The blades are pre-coned downwind from the plane of rotation hence; the planes of attachment are not perpendicular to the long axis of the hub. The hub body consists of teeter bearings which are held in position by removable bearing blocks. The teeter bearings carry all the loads passing between the hub body and the trunnion pin. The trunnion pin is connected rigidly to the main shaft. [5]

A teetering hub is allowed to move only a few degrees forward and backward during its normal operation but during high winds, starts and stops or high yaw rates, greater teeter excursions can occur. In order to prevent impact damage under these conditions, teeter dampers and compliant stops are used. A teetering hub is shown in Figure 20.

Figure 20 Teetering hub [4]

4.1.3 Hinged hubs

A hinged hub is basically a rigid hub with flanges for the blades but it is more complex in nature due to the hinge assembly used. It is considered as a combination of teetering & rigid hub. Since teetering hubs are majorly used in two bladed turbines, the blades tend to balance each other, so it does not require a centrifugal stiffener for low rpm operation. However, as there is no such counterbalancing on a hinged blade there has to be some kind of mechanism to prevent the blades from flopping over during low rotational speed. Normally, springs and dampers are used for this purpose. Figure 21 shows a hinged hub. [5]

Figure 21 Hinged hub [5]

4.2 Methods of Hub Attachment

In a pitch regulated wind turbine, the hub must have bearings at the blade root for the process of pitching of the blade. The hub is attached to the main shaft in such a way that it will not slip or spin on the shaft. Smaller turbines make use of keys, with keyways on the shaft and the hub. The shaft is also threaded and the mating surfaces are machined for a tight fit. This kind of attachment is not preferred in case of large wind turbines as keyways weaken the shaft and machining threads on a larger shaft is inconvenient.

Another method which can be used to attach the hub to wind turbine shafts is through Ringfeder shrink disc arrangement. In this arrangement, a projection on the hub slides over the end of the main shaft. The diameter of the hole in the hub projection is slightly larger than the diameter of the end shaft. The Shrink Disc consists of two discs and a ring. The inner surface of the ring slides over the outside of the hub projection. The outside of the ring is tapered in both axial directions. The two discs are placed on either side of the taper, and then pulled together with bolts. As they approach each other, the ring is compressed and this, in turn, compresses the hub projection. The compression of the hub projection clamps it to the hub. The Ringfeder shrink disc arrangement is shown in the figure 22.

The hub can also be attached to the main shaft by making use of a permanent flange at the end of the shaft. The flange may be either integral to the shaft or added later. The hub is attached to the flange through bolts. [5]

Figure 22 Ringfeder shrink disc arrangement. [5]

4.3 Material

Rotor hubs are likely to fail due to the process of fatigue hence, it is necessary to carefully select its material. The materials which are normally used are.

1. Steel.
2. Ductile Cast Iron.

4.4 Manufacturing Process

There are different ways in which a rotor hub can be manufactured; some of them are forging, welding and casting.

Forging helps us to create a high strength component as the forging operation can be carried out in such a way that the crystals are stretched in the direction of the stress. They are considerably lighter when compared to welding or casting. [15] However, it is not suitable to manufacture hubs of multi mega-watt turbines due to the high costs involved.

The Hubs which are produced by the process of welding involves minimum investment in production tools. They are normally used to manufacture hubs in small quantities. However, using the process of welding to manufacture hubs in large scale is not preferred as the production costs are high. [15]

Modern day rotor hubs have complex, three dimensional geometry and are produced by casting as it is economical when produced on a large scale.

4.5 Geometry

For three bladed machines, two types of rotor hubs can be identified

1. Tri-cylindrical or star shaped hub.
2. Spherical hub.

Tri cylindrical or star shaped hubs consist of three cylindrical shells which are concentric with the blade axes. These cylindrical shells flare into each other when they meet whereas, spherical hubs consist of a spherical shell with cut outs at the blade mounting positions. The spherical hub is also called as a ball hub. Figures 23, 24 show the spherical and tri-cylindrical hub. [16]

Figure 23 Tri-cylindrical or star shaped hub

Figure 24 Spherical hub [16]

4.5.1 Topology optimized rotor hub

Earlier it was common to use either spherical hubs or star shaped hubs, but nowadays it is possible to obtain topologically optimized rotor hubs. An optimization algorithm is used in FEM to vary the wall thickness according to the local stress and obtain a solution with minimum use of the material. These hubs have the shape of an apple with additional openings in zones where no material is needed. Figure 25, shows the comparison between the spherical hub, star hub and topologically optimized hub. [3]

Figure 25 Comparison between the spherical, star shaped and topologically optimized hub [3]

4.6 Size of the Rotor Hubs

The size of the rotor hub increases with the increase in the capacity of the wind turbine. Over the years there has been a tremendous increase in the size of wind turbines, from a modest 12kW wind turbine which was installed at Cleveland, Ohio in 1888 to an 8MW Vestas V164 prototype wind turbine which is installed offshore in 2015.

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Chapter 5

MODELLING OF THE ROTOR HUB

A generic 7.5MW reference rotor hub needs to be modelled in order to carry out fatigue analysis on it. The rotor hub which was modelled by SCHWACK [17] is recreated for this purpose. This chapter gives detailed information about the construction of a parametric rotor hub model.

5.1 Parametric Model

The construction of the CAD-model of the rotor hub is based on large number of sketches, reference co-ordinate systems, Boolean-operations, reference planes and their mutual dependencies. Since the model of the rotor hub has to be used for further research and the dimensions of various components that interact with the rotor hub are not completely defined, all the dimensions of the model are made parametric in nature. [17]

The aim of this parametric CAD-model is that, a change in the topology of the model can be implemented quickly and effectively without profound knowledge about the model. However, it is important to know the different types of parameters and their effects on the model to ensure an error-free operation of the model.

In general, three types of parameters are used for the construction of CAD-model of the rotor hub. These are illustrated by the first letter of the parameter.

5.1.1 V-parameter

All parameters which have the first letter as 'V' are the variable parameters. These parameters can be chosen freely, taking into account the physical and topological dependencies. [17]

5.1.2 C-parameters

The parameters that are identified by the letter 'C' are so-called design parameters. These parameters are required to form the parametric model and are based on

trigonometric relationships. The user of the model should not generally change these parameters as they have an impact on the complete structure of the model. The most important design parameter is the *Ball diameter*, as this includes the dimensions of the hub radius and the flanges. [17]

5.1.3 I-parameters

The parameters which are identified by the letter ‘I’ are the Interactive parameters. These parameters are both constructive and variable parameters. [17]

Table 1 gives an overview of the dimension of the parameters used in the creation of the model.

Parameters	Dimensions	Units
I_Flange edge blend radius	250	[mm]
C_Ball diameter	5421,87	[mm]
C_Stress reducing element edge blend radius	100	[mm]
V_Angle	2	[deg]
V_Flange outside diameter	3800	[mm]
V_Number of bolt holes on flange	24	[-]
V_Flange bolt hole diameter	45	[mm]
V_Flange height	0	[mm]
V_Flange inside diameter	3400	[mm]
V_Flange pitch circle diameter	3600	[mm]
V_Generator side(GS) bearing outside diameter	2400	[mm]
V_GS bearing outside hole number	20	[-]
V_GS bearing inside hole number	20	[-]
V_GS bearing outside hole diameter	100	[mm]
V_GS bearing inside hole diameter	75	[mm]
V_GS bearing outside hole depth	150	[mm]
V_GS bearing inside hole depth	200	[mm]

V_GS bearing width	250	[mm]
V_GS bearing inside diameter	1600	[mm]
V_GS bearing outside hole distance from center	1100	[mm]
V_GS bearing inside hole distance from center	900	[mm]
V_GS Bearing edge blend radius	250	[mm]
V_Wind Side (WS) bearing outside diameter	1700	[mm]
V_WS bearing width	200	[mm]
V_WS Bearing edge blend radius	250	[mm]
V_WS bearing inside diameter	1400	[mm]
V_Hub radius	2000	[mm]
V_Wall thickness	50	[mm]
V_Stress reducing element thickness	50	[mm]
V_Stress reducing element inside diameter	2000	[mm]

Table 1 Dimensions of various parameters of the model. [17]

5.2 Creation of the Parametric Model

The following steps are used to create the parametric model of the rotor hub.

5.2.1 Hollow sphere

- A hollow sphere with diameter of 5393.4mm which is obtained from the formula (see appendix) is created. It forms the basic structure of the rotor hub. Figure 26 gives the graphical representation of the hollow sphere and its cut section.

Figure 26 Graphical representation of hollow sphere and its cut section.

5.2.2 Creating wind side (WS) bearing.

- A datum plane (plane A) is created on the surface of the sphere. On the created datum plane, a circle with the diameter corresponding to the wind side bearing inside diameter is drawn and extruded into the sphere with Boolean operation *subtract* till the center of sphere (figure 27a).
- A new datum plane (plane B) is created on the surface of the sphere by using curve and points feature. The center of circles which have been created due to the process of extrusion is selected as the two points.
- Two concentric circles with diameters of wind side bearing outside and inside diameter (figure 27b) is created on plane B. The sketch is extruded into the sphere with a value corresponding to WS bearing width with Boolean operation *unite* (figure 27c).
- The edge blend function is used on the outer circle of the WS bearing to obtain a smoother finish.

Figure 27 shows the steps involved in the creation of the wind side bearing.

Figure 27 Steps involved in the creation of a wind side bearing.

5.2.3 Creating generator side (GS) bearing

- A datum plane (plane C) is created on the surface of the sphere exactly opposite to plane A. Plane C is used to draw a circle with diameter whose dimensions corresponds to GS bearing inside diameter. The sketch is extruded till the center of the sphere with the Boolean operation *subtract* (figure 28a).
- A new datum plane (plane D) is created on the surface of the sphere by using curve and points feature; the center of circles which have been created due to the process of extrusion is selected as the two points.
- On Plane D, two circles with dimensions corresponding to WS bearing outside and inside diameter is drawn (figure 28b). The created sketch is extruded into the sphere with the Boolean operation *unite* to obtain the GS bearing (figure 28c).
- A datum plane E is created on the surface of GS bearing. On the created datum plane, a circle at a distance of 1100mm from the center of GS bearing is drawn. The diameter of the circle corresponds to the dimension of GS bearing outside hole diameter. The created sketch is extruded into GS bearing using Boolean operation *subtract* to a depth corresponding to GS bearing outside hole depth dimension.
- Another circle is created at a distance of 900mm from the center of GS bearing. The diameter of the circle corresponding to the dimension of GS bearing inside hole diameter. The created circle is extruded it into the GS bearing using Boolean function *subtract* to a depth corresponding to GS bearing inside hole depth dimension.
- The pattern feature function is used to replicate the holes throughout the bearing (figure 28d).

Figure 28 shows the process of generating a generator side bearing.

Figure 28 Steps involved in the creation of a Generator side bearing

5.2.4 Creating a flange

- A datum plane (plane F) is created perpendicular to plane A or C on the surface of the sphere. On the created datum plane F, a circle with diameter of dimension equal to flange outside diameter is drawn and extruded into the sphere with the Boolean operation *subtract* till the center of the sphere. Figure 29 shows this process.

Figure 29 Boolean operation carried out on the sphere

- A datum plane G is created by using the function curves and points.
- Datum plane G is used to draw two concentric circles whose diameters corresponds to stress reducing element inside diameter and stress reducing element outside diameter.
- The created sketch is extruded away from the sphere using the Boolean operation *unite* to a distance specified by the value corresponding to stress reducing element thickness as shown in figure 30.

Figure 30 Creation of a stress reducing element.

- On datum plane G, two concentric circles with diameters equal to flange inside and outside diameter are drawn as shown in the figure 31.

Figure 31 Sketch showing flange inside and outside diameter.

- A datum co-ordinate system A is created on datum plane G. A second datum co-ordinate system (co-ordinate system B) is created keeping coordinate system A as a reference at a distance of 157.44mm and it is given by the formula (see appendix). The Y axis is tilted by an angle of two degrees (figure 32a).
- Datum plane H is created using coordinate system B (figure 32b).
- On the created datum plane H, two concentric circles, with diameters equal to flange inside and outside diameter are drawn (figure 32c).
- The sweep function is used and the curves created on the datum plane G and datum plane H are selected to obtain the flange of the rotor hub (figure 32d).
- The pattern feature function is used to replicate the flange at an angle of 120 degree to each other in order to obtain 3 flanges.
- The edge blend feature is used to give the flange a smoother finish.

Figure 32 shows the entire process of creation of the above mentioned steps.

Figure 32 Process of flange creation

On the created flange, a circle with diameter of 45mm is drawn and extruded into the flange to a depth of 50mm with the Boolean operation *subtract*. The pattern feature function is used to replicate this hole throughout the flange at an angle of 18 degree from each other. This process is carried out on the other flanges as well. The complete model of the rotor hub is shown in the figure 33.

Figure 33 Complete model of the rotor hub.

This chapter explained the detailed process involved in the creation of the parametric rotor hub model. The created model is imported to ANSYS for further analysis. The prerequisites for setting up the analysis are explained in detail in the next chapter.

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Chapter 6

ANALYSIS SETUP

Once the model is created, it is imported into ANSYS 15 to carry out further analysis. The material used for the analysis is EN-GJS-400-18 LT. The properties of this material are inserted into the material database. The extreme load simulations which are discussed in the work carried out by SCHWACK [17] are re-simulated.

6.1 Sensitivity Analysis

Sensitivity analysis is carried out to determine the appropriate mesh size that can be used for the analysis. ANSYS by default generates a particular mesh size and this is referred to as basic mesh. The basic mesh cannot be used to carry out the analysis as it's very coarse and in order to improve the accuracy of the results, it is necessary to decrease the size of the mesh and refine it. Figure 34 shows the stresses and deformations obtained on application of extreme loads using basic mesh.

Figure 34a) Basic mesh

Figure 34b) Stresses acting on the hub with basic Mesh

Figure 34c) Deformations experienced by the hub with basic mesh

6.1.1 Mesh refinement 1

The mesh is refined from the default size. The flanges, bearings and stress reducing elements are given a size of 40mm and the rest of the model has a mesh size of 90mm. Patch conforming method using tetrahedron elements are used on the whole body in order to obtain a better quality mesh. The computation time for such a mesh

size is around 45 minutes. The results obtained are shown in figure 35. It is be seen that the stresses and deformations with mesh refinement 1 are deviated by more than 5% compared to the stresses and deformations obtained from analysis with basic mesh and hence it has to be further modified.

Figure 35a) Mesh refinement 1

Figure 35b) Stresses for mesh refinement 1

Figure 35c) Deformations for mesh refinement 1

6.1.2 Mesh refinement 2

In order to improve the accuracy, the mesh is further refined. The flanges, bearings and stress reducing elements are given a size of 40mm and the rest of the model has a mesh size of 80mm, the patch conforming method using tetrahedron elements is used on the model. The results obtained are shown in figure 36. It is seen that the stresses and deformations obtained from the analysis are deviated by less than 5% when compared to the results obtained from analysis with mesh refinement 1. The mesh takes a computation time of about 60 minutes but in order to use this mesh size for the current analysis, it is necessary to prove that a further reduction in mesh size will have negligible or no effect on the results obtained.

Figure 36a) Mesh refinement 2

Figure 36b) stresses acting on the hub with Mesh refinement 2

Figure 36c) Deformations experienced by the hub with Mesh refinement 2

6.1.3 Mesh refinement 3

A further refinement in mesh size is carried out to check if there are any considerable changes in the values of the results obtained with further reduction in mesh size from mesh refinement 2. The flanges, bearings and stress reducing elements are given a size of 40mm and the rest of the model has a mesh size of 70mm with patch conforming method using tetrahedron elements applied on the whole body. The

results from the analysis are shown in figure 37. It is observed that the results obtained are deviated by less than 5% when compared to the results from analysis with mesh refinement 2. However, the computation time is increased from 60 minutes to 90 minutes.

Figure 37a) Mesh refinement 3

Figure 37b) Stresses experienced by the hub with Mesh refinement 3

Figure 37c) Deformations experienced by the hub with Mesh refinement 3

It has been seen through the above process that mesh refinement 2 gives us accurate results with computational time of 60 minutes and a further decrease in the mesh size yields no considerable change in the results but takes more computational time. Hence, the mesh size used in mesh refinement 2 is applied to the hub and further analysis is carried out on it.

6.2 Co-ordinate System

In order to achieve consistency and enable easier comparison between different simulation results, a uniform co-ordinate system has to be used. For detailed analyses of the hub, it is necessary to have two co-ordinate systems as described by Germanischer Lloyd [18].

1. Hub co-ordinate system
2. Blade co-ordinate system

A Hub coordinate system has its origin at the center of the rotor and does not rotate with the rotor. The X-axis (XN) points in the direction along the main shaft in the downwind direction, the Z axis (ZN) vertically upwards and the Y – axis (YN) is perpendicular in accordance with the right hand rule. The hub co-ordinate system is shown in the figure 38. [18]

Figure 38 Hub co-ordinate system [18].

The Blade coordinate system has its origin at the blade root and rotates with the rotor. The X-axis (XB) points in the direction along the main shaft in the downwind direction, the Z-axis (ZB) is in the radial direction and the Y-axis (YB) is perpendicular in accordance with the right hand rule. The blade co-ordinate system is shown in the figure 39. [18]

Figure 39 Blade co-ordinate system [18].

6.3 Synthesizing an S-N Curve

Using the formulae mentioned in section 3.3.1 an effort is made to synthesize an S-N curve for the material EN-GJS-400-18LT.

The value of $R_{m,min}$ as obtained from the handbook [10] is 370 MPa.

The value of σ_b is given by the equation 3.6

$$\sigma_b = 1.06 * 370 = 392.2 \text{ Mpa} \quad (6.1)$$

The next step is to obtain the fatigue strength of the material and since EN-GJS-400-18LT is Nodular cast iron, σ_w is given by equation 3.8, hence

$$\begin{aligned} \sigma_w &= 0.27 * 382.2 + 100 \text{ MPa} \\ &= 212.25 \text{ MPa} \end{aligned} \quad (6.2)$$

As the surface roughness and stress gradient effects have to be considered we need to calculate them. We know from equations 3.12-3.15 that,

$$F_o = 1 - 0.22 * (\log R_z)^{0.64} * (\log \sigma_b) + 0.45 * (\log R_z)^{0.53}$$

$$n = 1 + 0.32 * X^{0.77}$$

$$\beta_k = \frac{\alpha_k}{n}$$

$$X = \frac{1}{thickness} \left(\frac{1}{mm} \right)$$

R_z is the surface roughness and is equal to 25 μ m as the material used is nodular cast iron.

Hence,

$$\begin{aligned} F_o &= 1 - 0.22 * (\log 25)^{0.64} * (\log 392.2) + 0.45 * (\log 25)^{0.53} \\ &= 0.86 \end{aligned} \quad (6.3)$$

$$X = 1/50 = 0.02 \left(\frac{1}{mm} \right) \quad (6.4)$$

$$\begin{aligned} n &= 1 + 0.32 * 0.018^{0.77} \\ &= 1.01 \end{aligned} \quad (6.5)$$

The value of thickness is considered to be 50mm as model 1 has hub body thickness of 50mm but, this value has minimum effect on the synthetic S-N curve obtained, the value of 'X' given in (6.4) is valid even when there is a small change (around 20-30mm) in the thickness of the hub body in future models.

During the process of meshing, the entire body will be divided into numerous tiny cubes. The stress concentration factor for a cube is equal to 1. Hence, the average stress concentration of the entire body is considered to be equal to one. Therefore, β_k given by equation 3.14 is,

$$\beta_k = \frac{1}{1.01} = 0.99 \approx 1 \quad (6.6)$$

The total influencing factor given by equation 3.11 is calculated below

$$\begin{aligned} F_{ok} &= \sqrt{1^2 - 1 + \frac{1}{0.86^2}} \\ &= 1.16 \end{aligned} \quad (6.7)$$

The fatigue strength of the material after considering the surface roughness and stress gradient effects is given by equation 3.10 and is as follows,

$$\begin{aligned} \sigma_{wk} &= \frac{212.25}{1.16} \\ &= 182.82 \text{ MPa} \end{aligned} \quad (6.8)$$

For alternating stress, the stress ratio $R = -1$ hence, $F_m = 1$.

The stress amplitude at the foot of the S-N curve σ_A , is given by equation 3.16,

$$\sigma_A = 182.82 * 1 = 182.2 \text{ MPa} \quad (6.9)$$

σ_A calculated above is for an S-N curve which has 97.7% survival probability. The slopes of the synthetic S-N curve are given by the formulae 3.19, 3.20 and the values are calculated below

$$m_1 = \frac{5.5}{1.16^2} + 6$$

$$= 10.08 \quad (6.10)$$

$$m_2 = 2 * 10.08 - 1 = 19.16 \quad (6.11)$$

Number of load cycles at the knee of the synthetic SN curve N_D obtained from equation 3.21 is calculated below

$$N_D = 10^{6.8 - \frac{3.6}{10.08}} \text{ cycles}$$

$$= 10^{6.44} \text{ cycles} \quad (6.12)$$

The upper limit of the fatigue life line, σ_1 is given by

$$\sigma_1 = 392.2 * \left(\frac{1 - (-1)}{2} \right)$$

$$= 392.2 \text{ MPa} \quad (6.13)$$

The Number of cycles to failure at the upper limit of fatigue life (i.e. at 392.2 MPa) is calculated by using formula 6.14,

$$\frac{N_1}{N_D} = \left(\frac{\sigma_1}{\sigma_A} \right)^{-m_1} \quad (6.14)$$

$$\frac{N_1}{10^{6.44}} = \left(\frac{392.2}{182.82} \right)^{-10.08}$$

$$N_1 = 1255 \text{ cycles} \quad (6.15)$$

The different points of synthetic S-N curve for the material EN-GJS-400-18 LT is calculated and is shown in table 3. The graph representing the synthesized SN curve can be seen in figure 40.

Stress (MPa)	Life (in number of cycles)
392.2	1253.07
319.17	10000
254	100000
202.12	1000000
182.2	2750000
170.90	10000000
151.55	100000000

Table 2 Synthetic S-N curve calculated values

Figure 40 Synthetic S-N curve.

The synthetic SN curve shown in figure 40 is inserted into the material properties of EN-GJS-400-18LT and the analyses are carried out.

6.4 Load Application and Boundary Conditions

Once the mesh size is determined; the next step in the analysis is to determine the loads and boundary conditions which need to be applied on the model in order to carry out the analysis.

The loads in the form of Damage Equivalent Loads (DELs) are obtained from the turbine simulation tool HAWC2 by assuming Rayleigh distribution of wind for a period of 20 years and a lifetime of 1×10^7 cycles. The DELs signify that if the rotor hub is able to withstand the DELs for 1×10^7 cycles without undergoing failure, then, it will have a lifetime of 20 years as the wind data for 20 years is used as one of the input to obtain the DELs. The damage equivalent loads consist of forces and moments in X, Y and Z directions.

The values of forces and moments constituting the DELs are obtained for different S-N curve slopes from the simulation tool. In section 6.3, slope of the S-N curve synthesized is calculated and it is around 10, the corresponding life time is around 2.75×10^6 cycles. The value calculated is in agreement with slope of the S-N curve as calculated for pressure vessels in the design code EN 13445-6 for material EN-GJS-400-18LT with life time greater than 2×10^6 cycles [19]. The values of forces and moments obtained from the simulation tool for an S-N curve slope of 10 is given in table 3.

F_{XB}	825 kN
F_{YB}	426 kN
F_{ZB}	839 kN
M_{XB}	2.02×10^4 kN-m
M_{YB}	1.89×10^4 kN-m
M_{ZB}	4.24×10^2 kN-m

Table 3 Damage Equivalent Loads (DELs) consisting of forces and moments in X, Y and Z direction

The forces and moments in X, Y and Z directions were applied to all the three flanges of the rotor hub. The generator side bearing and Wind side bearing were constrained such that it is completely fixed in all the directions and the simulations are carried out to obtain equivalent

stress and total deformation. Figure 41 shows the graphical representation of DEL application (Resultant forces and moments).

Figure 41 Graphical representation of DEL application to the rotor hub.

Chapter 7

RESULTS AND DISCUSSIONS

After the process of meshing, extreme load simulations are carried out on the rotor hub. If the hub model withstands the extreme loads, then it is subjected to DELs. Fatigue tool is inserted into the analysis set up and stress life approach is chosen as we are dealing with high cycle fatigue. Goodman criterion is used as the mean stress theory since the material used is nodular cast iron and it does not have definite yield strength. The analyses are carried out to determine the equivalent stress, deformation and life of the material. The results of the analyses are discussed in this chapter.

7.1 Model 1

Model 1 is subjected to both extreme load situations and DELs which were discussed in section 6.3. The results from extreme load simulations are not discussed here as it was observed that the results were same as that obtained by SCHWACK [17] in his work. It was found from the extreme load simulations that model 1 would not fail when subjected to extreme loads; therefore it was further subjected to DELs. The results obtained from DELs simulations for model 1 are discussed below.

7.1.1 Stress

The material EN-GJS-400-18 LT has a tensile strength of 400MPa, so any stress acting on the material greater than that will cause the material to undergo fracture and finally failure. Once the analysis was completed, the stress acting on the hub was determined. Figures 42 and 43 show the stress acting on the hub (model 1). It can be seen that the maximum stress acting on the model is 340.61 MPa which is less than the allowed 400MPa hence; the material is not going to fail due to the applied loads. Maximum stress acts on the manholes.

Figure 42 Stress acting on the rotor hub (model 1) (view 1).

Figure 43 Stress acting on the rotor hub (model 1) (view 2).

7.1.2 Deformation

The deformation of the rotor hub after applying the DELs is shown in figures 44 and 45. It can be seen that the deformation experienced by the rotor hub is around 8 mm. Maximum deformation occurs on the stress reducing element.

Figure 44 Deformation of the rotor hub due to the applied DELs (view 1).

Figure 45 Deformation of the rotor hub due to the applied DELs (view 2).

7.1.3 Life

A wind turbine rotor hub is subjected to alternating loads and is expected to withstand them for at least 1×10^7 cycles i.e. the design life of a rotor hub is expected to be at least 1×10^7 cycles. Considering this, the analysis was carried out to determine the life of the rotor hub. From figure 46, it can be seen that the minimum

life is 5550 cycles which is less than the expected design life. Hence, it is necessary to modify certain parts of the rotor hub in order to improve its fatigue life. Figure 46 and 47 shows the life cycle of different parts of the hub.

Figure 46 Life of different components of rotor hub after being subjected to DELs (view1).

Figure 47 Life of different components of rotor hub after being subjected to DELs (view2).

7.2 Optimization 1

The analysis carried out on model 1 had a design life of less than 1×10^7 cycles, a new model (optimized model 1) of similar construction and dimensions with minor changes is proposed in this section. The thickness of the wall is increased from 50mm to 78mm. While carrying out the analysis for model 1, it was observed that the region around the manholes was susceptible to failure. In order to prevent this, shoulders are added around the manholes.

To create a shoulder, two concentric circles of diameters 700mm and 900mm are drawn around the manhole. The created sketch is extruded into the sphere with the Boolean operation *unite* to a thickness of 170mm. This feature is patterned around other manholes.

The reconstructed model is shown in figure 48.

Figure 48. Optimized model 1

Extreme load simulations are carried out on the optimized model. If the model withstands the extreme loads then it will be subjected to DELs. There are six extreme load cases which have to be applied to each flange of the rotor hub. The magnitudes of forces and moments that constitute the extreme loads are tabulated in the appendix. Extreme loads are applied to the rotor hub in a similar way to DELs application as described in section 6.3. The variation of stress and deformation acting on the rotor hub (optimized model 1) with the different load cases are shown in figures 49 and 50.

Figure 49 Graph showing the variation of stress with different load cases for optimized model 1

From figure 49, it can be observed that the maximum stress acting on the body due to the extreme loads is around 115 MPa and it is less than the tensile strength of the material used. Hence, it can be concluded that the rotor hub will not undergo failure when subjected to extreme load situations. In the figure 49, the load cases are represented on the X axis and each time step represents a different load case. The green curve in figures 49 and 50 represent the maximum value for that particular case and the red curve indicates the minimum value. Figure 50 shows the variation of deformation experienced by the body due to the different load cases acting on it.

Figure 50 Plot representing the variation of deformation with different load cases for optimized model 1.

As the optimized rotor hub 1 withstood the extreme load conditions, it is now subjected to DELs. The DEL analyses are carried out without changing any input data from the DEL analyses of model 1 to determine the effect of wall thickness on the fatigue life, equivalent stress and deformation.

7.2.1 Stress

From figures 51 and 52, it can be seen that the maximum stress that the model is subjected to is 162.7 MPa. The material has a capability to withstand 400MPa. So this implies that the optimized model 1 will not fail due to excessive stress. Also, by comparing model 1 and optimized model 1 we can infer that the by increasing the thickness from 50mm to 78mm, the maximum stress acting on the hub is reduced from 340 MPa to 162.7Mpa.

Figure 51 Stress acting on optimized model 1 (view 1).

Figure 52 Stress acting on optimized model 1 (view 2).

7.2.2 Deformation

It is noticed from figures 44, 53 and 54 that the deformation experienced by the hub is reduced from 8 mm to 3.7 mm due to increase in the wall thickness from 50mm to 78mm.

Figure 53 Deformation experienced by optimized model 1 due to the applied DELs (view 1).

Figure 54 Deformation experienced by optimized model 1 due to the applied DELs (view 2).

7.2.3 Life

Fatigue analysis is carried out on optimized model 1 using the S-N curve which was synthesized earlier to determine the life of the hub. The result obtained is shown in figure 55. It can be observed that the minimum life is 2.65×10^7 cycles which is greater than the design life of 1×10^7 cycles. The minimum life occurs at the manholes and can be seen in figure 56.

Figure 55 Life of the optimized model 1 after being subjected to DELs.

Figure 56 Part of the rotor hub with minimum life

Although optimized model 1 withstands the DELs and has a design life of more than a 1×10^7 cycles, it is heavy as its weight accounts to 54,294 kg and efforts are made to bring down the weight to less than 45,000 kg.

7.3 Optimization 2

In order to reduce the weight of the rotor hub, efforts are made to modify the rotor hub by removing the unnecessary material.

Model 1 is kept as the base model and changes are made to it so that it withstands the DELs applied and does not fail during its design life. It was noticed that an increase in wall thickness helped in reducing the stress acting on the hub. Also an increase in the wall

thickness increases the weight; hence a compromise has to be made. After a series of trial and error, a thickness of 68mm was found to be optimum.

Increasing the thickness of the rotor hub from 50mm to 68mm would solve one of the problems associated with the model. Few more modifications have to be made in order to make sure that the model does not fail under applied DELs. The modifications are discussed below.

During the process of trial and error to find the optimum thickness, it was noticed that the region connecting the stress reducing element to the hub body was subjected to maximum stress. In order to prevent this, the thickness in this region has to be increased. The process by which the thickness is increased is described below.

Arcs with radius of 2685mm and 2605mm are drawn and are connected by straight lines to form the image which is shown in the figure 57. This sketch is revolved such that the thickness in the region connecting stress reducing element and the hub body is increased. By doing this, the stress acting in this particular region is reduced while adding minimum weight to the overall structure. Edge blend features are applied in order to have a smooth transition. The revolve feature is then patterned to all 3 rotor bearings.

Figure 57 Sketch drawn in order to increase the thickness of the region susceptible to failure

Figure 58 Strengthening of the region susceptible to failure by adding material.

The other major problem which reduces the overall life of the hub is the manhole; the stress acting in this particular region is above the tensile strength of the material. In order to solve this problem shoulders are inserted around the manholes. Previous experiences have shown us that increasing thickness will help in reduction of stresses.

To create a shoulder, draw concentric circles with diameters of 675mm and 900mm around the manhole. Extrude the sketch towards the center of hub to a distance of 170mm with the Boolean operation unite (figure 59 a, b). Use the edge blend feature to blend the edges and have a smoother transition. Pattern this feature around all the three manholes (figure 59 c, d). The process of shoulder generation is shown in the figure 59.

Figure 59 Process of manhole generation

Addition of shoulders to the manholes increase the overall weight of hub but our objective is to design a lightweight rotor hub which can withstand the DELs applied for a million cycles. Therefore, it is necessary to identify the parameters that can be modified which in turns leads to removing the unwanted material. The rotor hub has very few parameters that can be changed, one such parameter is the diameter of the stress reducing element. The inside diameter of the stress reducing element is increased from 2000mm to 2500mm, this helps in decreasing the material used in stress reducing element and thus the overall weight of the hub. The modified stress reducing element is shown in figure 60.

Figure 60 Modified stress reducing element

After all the modifications made to model 1, we get a model which looks similar to model 1 on the first look, but when examined closely the various modifications done to it can be identified. The model with modifications (optimized model 2) is shown in the figure 61.

Figure 61 Optimized model 2

The optimized model 2 is subjected to extreme load cases. Figures 62 and 63 show the graphs representing the variation of stress and deformation experienced by the hub body due to different load cases. From figure 62, it can be seen that the maximum stress experienced by the body is 120.9 MPa while the tensile strength of the selected material is around 400MPa. So, it would be safe to say that the material will not undergo failure in extreme load conditions. Figure 63 shows a plot representing the changes in deformation with different load cases.

Figure 62 Graph showing the variation of stress with different load cases for optimized model 2

Figure 63 Plot representing the variation of deformation with different load cases for optimized model 2.

Once the optimized model 2 withstood the extreme load situations, it was further subjected to DELs. The DEL analyses are carried out on the optimized model 2 without changing any input data from the DEL analyses of model 1. The results of the analysis are discussed below.

7.3.1 Stress

From figures 64, 65 and 66, it can be seen that the maximum stress acting on the hub body is around 166 MPa. It is observed that there has been a slight increase in the magnitude of stress acting on the hub compared to the stress acting on optimized model 1 which may be due to the decrease in the wall thickness. Maximum stress acts on the inside portions of the shoulder and in the region between generator side bearing and the manholes.

Figure 64 Stress acting on optimized model 2 (view 1).

Figure 65 Stress acting on optimized model 2 (view 2).

Figure 66 Stress acting on optimized model 2 (view 3).

7.3.2 Deformation

Due to the reduction of material in the stress reducing element, there has been an increase in the value of maximum deformation experienced by the rotor hub from 3.7mm to 4.8mm. Figure 67 and 68 shows the graphical representation of deformations which the rotor hub is subjected to.

Figure 67 Deformation experienced by optimized model 2 due to the applied DELs (view 1).

Figure 68 Deformation experienced by optimized model 2 due to the applied DELs (view 2).

7.3.3 Life

The optimized model 2 is subjected to DELs and fatigue data as discussed in section 6.3 and 6.4. It is seen that the model is able to withstand the loads without undergoing failure for more than 1×10^7 cycles. Figure 69 and 70 shows the life of different parts of the rotor hub.

Figure 69 Life of different components of optimized model 2 after being subjected to DELs (view 1).

Figure 70 Life of different components of optimized model 2 after being subjected to DELs (view 2).

From the above results it can be seen that the optimized model 2 withstands the applied DELs for more than 1×10^7 cycles and functions even in extreme conditions without undergoing failure. The weight of the optimized model 2 is around 43700 kg which is less than the maximum weight of 45000 kg that was set as an upper limit before starting the optimization process.

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Chapter 8

CONCLUSIONS AND FUTURE WORK

In recent decades, there has been an increase in the use of wind turbines to generate electricity because of which the capacities of the wind turbines are continuously increasing. The problems associated with wind turbines increase with the increase in their capacity. In order to deal with such problems, it is essential to have well-documented literature on all components of wind turbine. However, it is observed that there is less information available on some components like rotor hubs.

In the present work, an effort is made to contribute literature on the fatigue properties of the rotor hub by designing a generic 7.5MW reference rotor hub and carrying out fatigue analysis on it. The rotor hub designed is initially subjected to extreme loads and if the rotor hub withstands these loads without undergoing failure, it is further subjected to DELs wherein its fatigue properties are determined.

The model described by SCHWACK in his work was recreated and was optimized in order to obtain a model which could withstand the DELs for more than 1×10^7 cycles without undergoing failure and work even in extreme situations. The optimization process was carried out with a goal to restrict the maximum weight to less than 45000kg. After designing a series of models which failed, an optimized model which fulfilled the above mentioned criteria was obtained.

The optimized model created has a design life of 1.7×10^7 cycles weighing about 43,700 kg and is capable of withstanding the extreme load conditions. Table 4 gives information about the models created, their minimum life and weight. The table summarizes the entire work.

Model	Weight (kg)	Minimum life
Original model created by SCHWACK	36571	5450 cycles
Optimized model 1	54294	2.65×10^7 cycles
Optimized model 2	43653	1.72×10^7 cycles

Table 4 Models created with their life cycle and corresponding

This project was started with the notion of adding literature to the field of fatigue analysis of rotor hubs as there is a lack of information regarding the SN curve or the fatigue properties of the material EN-GJS-400-18LT, which is one of the widely used materials to manufacture rotor hubs. Also, as per the literature available, a 5 MW reference rotor hub without taking fatigue life into consideration weighs about 56,000 kg [20] while the rotor hub created in this project is a 7.5MW reference rotor hub which takes fatigue life into consideration and weighs about 44000 kg . Hence, the objective of obtaining a relatively light weight rotor hub and contributing literature to the field of fatigue analysis of rotor hubs is achieved through this project.

The fatigue life of the model designed in this thesis can be validated by carrying out accelerated fatigue tests on a test bench. Manufacturing the entire rotor hub for the process of testing would be expensive. Hence, a component with similar properties as that of a rotor hub (i.e. same stiffness and load bearing capacity) is designed for the test bench.

Chapter 9

APPENDIX

- The radius of the hollow sphere used to create the rotor hub model is given by,

$$R_{Ball} = \sqrt{\left(R_{Hub} - H - \left[\sin(\alpha) * \frac{D_{Flange}}{2}\right]\right)^2 + \left[\frac{D_{Flange}}{2}\right]^2}$$

Where,

- R_{ball} – Radius of the hollow sphere to be created (mm).
- R_{Hub} – Radius of the rotor hub (mm).
- H – Flange height (mm).
- α – Cone angle (degree).
- D_{Flange} – Diameter of the flange (mm).

- The distance (d) between co-ordinate system 1 and co-ordinate system 2 is given by the formula

$$d = \tan(\delta) * R_{flange\ pitch\ circle}$$

Where, δ is the tilt angle (angle between the horizontal axis and the rotor axis) and is equal to 5 degrees.

- Magnitude of the forces and moments that constitute the extreme loads are given below:

		Load cases					
		1	2	3	4	5	6
Blade 1	M_X [N-mm]	-1.44e10	-1.64e10	-1.21e9	1.86e10	1.17e10	5.5e9
	M_Y [N-mm]	-6.24e8	4.56e9	-1.59e9	1.86e9	-4.56e8	4.46e9
	M_Z [N-mm]	6.19e7	4.21e8	5.24e8	-7.33e6	1.07e7	-8.57e7
	F_X (N)	-5.03e5	-3.99e5	-4.44e5	5.72e5	2.41e5	1.83e5
	F_Y (N)	70880	-7150	61860	-40080	2.78e5	-2.2e5
	F_Z (N)	10347	9.72e5	1.41e5	-2.76e5	7.03e5	1e6

Blade 2	M _X [N-mm]	2.88e10	-1.63e10	4.98e9	-6.77e9	1.16e10	3.44e8
	M _Y [N-mm]	4.19e9	5.81e9	1.64e8	2.49e9	3.12e9	-2.4e9
	M _Z [N-mm]	6.18e7	3878.2	-3.77e8	-3.33e7	-2.4e7	-7.58e7
	F _X (N)	86670	-3.75e5	3.03e5	-1.1e5	2.75e5	12740
	F _Y (N)	-1.24e5	-81710	-16440	-30530	-1.46e5	1.45e5
	F _Z (N)	-2.72e5	4.4e5	-1.43e5	1.99e5	4.3e5	1.32e6

Blade 3	M _X [N-mm]	-1.51e10	-1.47e10	7.15e9	-1.24e10	1.18e10	4.64e9
	M _Y [N-mm]	3.09e9	1.343e10	-1.01e9	2.44e8	3.84e9	-4.69e9
	M _Z [N-mm]	1.6e8	5.17e8	3.78e8	6.54e7	-1.43e7	-9.44e7
	F _X (N)	-3.52e5	-2.34e5	2.52e5	-4.56e5	2.73e5	1.13e5
	F _Y (N)	-41530	4.69e5	15020	53950	-1.76e5	2.54e5
	F _Z (N)	2.3e5	7.7e5	3e5	1.19e5	9.38e5	8.44e5

Chapter 10

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