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DESIGN AV NACELLE FOR EN 10 MW VINDTURBIN

Ebbe Berge Smith

Master i produktutvikling og produksjon
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Hovedveileder: Bjørn Haugen, IPM

Norges teknisk-naturvitenskapelige universitet
Institutt for produktutvikling og materialer

MASTEROPPGAVE VÅR 2012
FOR
STUD.TECHN. EBBE SMITH

DESIGN AV NACELLE FOR EN 10 MW VINDTURBIN
Design of a nacelle for a 10 MW wind turbine

Utviklingen av vindturbiner har vært formidabel i de siste år. Det finnes kommersielt tilgjengelig 5 MW turbiner med diameter over 120 meter. Disse er også installert offshore på bunnfaste installasjoner. Morgendagens turbiner er muligens enda større og kan være flytende. Dette er et av temaene for forskningssentret NOWITECH. Det jobbes med å utvikle en referanseturbin på 10 MW som skal benyttes i forskningssammenheng på offshore flytende turbiner.

Oppgaven innebærer mekanisk design av en nacelle for en 10 MW offshore vindturbin

Oppgaven bearbeides ut fra følgende punkter:

1) Litteratursøk
   a) Gjøre seg kjent med "state of the art" av vind turbiner
   b) Gjøre seg kjent med design av vindturbiner og hvordan nacellen er dimensjonert
   c) Gjøre seg kjent med standarden IEC 61400

2) Software kjennskap
   a) Gjøre seg kjent med DAK-programmet; NX
   b) Gjøre kjent med styrkeberegning vha Nastran

3) Gjennomføre mekanisk design og dimensjonering av følgende komponenter:
   a) Bunnplate
   b) Hovedaksling
   c) Hub

4) Finne løsning for hovedlager og yaw-lager

Før kandidaten går i gang med forsøkene, skal han/hun sette opp en oversiktlig forsøksplan som inneholder alle opplysninger om forsøksmetodikk og forsøksbetingelser. Planen skal foreligge til godkjenning innen tre uker etter utlevering av oppgaven.

Besvarelsen skal ha med signert oppgavetekst, og redigeres mest mulig som en forskningsrapport med et sammendrag på norsk og engelsk, konklusjon, litteraturliste, innholdsfortegnelse, etc. Ved utarbeidelse av teksten skal kandidaten legge vekt på å gjøre teksten oversiktlig og velskrevet. Med henblikk på lesning av besvarelsen er det viktig at de nødvendige henvisninger for korresponderende steder i tekst, tabeller og figurer anføres på begge steder. Ved bedømmelse legges det stor vekt på at resultater er grundig bearbeidet, at de oppstilles tabellarisk og/eller grafisk på en oversiktlig måte og diskuteres utførlig.

Senest 3 uker etter oppgavestart skal et A3 ark som illustrerer arbeidet leveres inn. En mal for dette arket finnes på instituttets hjemmeside under menyen undervisning. Arket skal også oppdateres ved innlevering av masteroppgaven.

Besvarelsen skal leveres i elektronisk format via DAIM, NTNUs system for Digital arkivering og innlevering av masteroppgaver.

Kontaktperson:
Professor Ole Gunnar Dahlhaug, Institutt for Energi og Prosessteknikk, NTNU

Torgeir Welo
Instituttleder

Bjørn Haugen
Faglærer

NTNU
Norges teknisk-
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Preface

This master thesis was written for the institute of product development and material science (IPM) at the Norwegian university of science and technology (NTNU) in the spring of 2012.

The report includes work on the bedplate and main shaft bearings for the NOWITECH reference turbine which is a 10MW three bladed upwind turbine. The assignment specifies mechanical design and dimensioning of central rotor nacelle components such as the main shaft, main bearings, yaw bearing, bedplate and hub. It became clear at an early stage that it was necessary to limit the assignment. The hub and main shaft became the main focus for an overlapping master thesis while I focused on the bedplate and bearings.

Development of the bedplate was performed using the computer assisted design (CAD) software NX 7.5 with the integrated finite element method (FEM) solver Nastran. My previous experience with this software was obtained in the courses TMM4135 (Dimensioning 101) and TMM4155 (Product development and materials) where I was introduced to basic simulations with Nastran and surface modeling.

Most of the time for this project went into learning the software tools and developing simulations scenarios based on relevant standards. My personal goal was to develop a lean bedplate design and this caused important facets of the assignment to be somewhat neglected. Examples are fatigue testing and design details regarding the yaw solution with emphasis on yaw bearing selection.

During this project I have enjoyed bi-weekly meetings with the "wind group" that consisted of Bjørn Haugen (main supervisor), Ole Gunnar Dahlhaug (co supervisor), Lars Frøyd (Phd) and Peter Kalsaas Fossum (master student). These meetings provided insights in related subjects beyond my assignment as well as invaluable feedback and suggestions. I would especially like to thank Bjørn Haugen, who patently provided feedback and guidance over the course of the entire semester. Finally, I would like to thank Remi Pedersen and Hans Magnus Johnsen for providing corrections and suggestions to the manuscript.

Wind turbines was a completely new subject for me and I found it to be a difficult yet an exciting challenge. At this point I have more questions regarding wind turbine design in general than when I started, and I am eager to learn more about the subject.

Trondheim, 11.06.2012

Summary

A solution containing two main shaft bearings and a new bedplate design for the NOWITECH 10MW reference turbine is proposed based on extreme loading. The extreme loads was determined for power production under normal and extreme turbulence conditions specified in IEC61400-3:2008 - Design requirements for offshore wind turbines [1].

The bedplate consist of two components with a total weight of 82.4tons and does not contain a yaw-system to transmit loads to the tower. Ultimate strength analysis was performed for the determined extreme loads, where the highest von Mises stress was found to be 208.3MPa.

A configuration of a floating spherical roller bearing and a fixed double tapered roller bearing was selected from the SKF product catalog and is briefly discussed. Finally, future work for the bedplate is discussed.

Sammendrag

En løsning for to hoved lager og en bunnplate for NOWITECH 10MW referanse turbinen er foreslått basert på dimensjonering i henhold til ekstremlaster. Ekstremlastene er basert på ekstrapolerte laster under vanlig strøm produksjon for normale og ekstreme turbulens forhold spesifisert i IEC61400-3:2008 - Design requirements for offshore wind turbines [1].

Bunnplaten består av to komponenter med en total vekt på 82.4 tonn, bunnplaten har hittil ingen yaw-løsning for overføring av laster til tårnet. Styrkeanalyser er utført basert med ekstremlastene og den høyeste von Mises spenningen ble funnet til å være 208,3MPa.

En konfigurasjon av et kulelager og et dobbelt konisk rullelager ble valgt ut ifra SKF sin produktkalatog. Tilslutt er videre arbeid for bunnplaten diskutert.

Contents

1. Introduction

The first wind turbines was designed to automate time-consuming tasks as grinding grain and pumping water. The earliest known turbine design is a vertical axis system developed in Persia 500-900 A.D [2]. Since then wind turbines have evolved to turn kinetic wind energy into electrical energy for every day consumption. At the end of 2011 the total nameplate capacity for wind energy was rated 238,3 GW [3] and have experienced a doubling every three years since 2006.

A wind farm consist of a group of wind turbines placed in close proximity and connected in a grid. Wind farms are today to be found both on- and offshore. As wind energy production increases, two main directions provides driving forces for future development, increased efficiency in power production over area consumption and utilization of offshore areal. The latter option is commonly preferred, as wind turbines is in the public eye viewed as visual pollution. Those living close to wind farms may also be bothered by the noise levels, which also may have an impact on the local ecological environment.

Table 1.1 - Wind turbine classes [4]

	Onshore	Near shore	Offshore
Wind speed class	III	II	I
Annual avererage wind speed	6,5 – 7,5 m/s	7,5 – 8,5 m/s	8,5 – 10,0 m/s
Capacity factor	20 – 30 %	30 – 40 %	40 – 50 %
Rotor diameters	90 – 120 m	100 – 120 m	115 – 130 m
Nominal turbine power	1,5 – 3,0 MW	3,0 – 4,0 MW	4,0 – 6,0 MW
Wind power plant cost	1,2 – 1,6 M€/MW	1,4 – 1,8 M€/MW	2,0 – 4,0 M€/MW
Availability	> 90 %	> 95 %	> 98 %
Tower Head Mass	< 140 t	200 – 300 t	300 – 500 t
Generating cost	60 – 90 €/MWh	50 – 70 €/MWh	70 – 150 €/MWh

The offshore wind conditions are considered to be preferred compared to onshore conditions (see table 1.1) as the turbulence caused by local topography is virtually non-existing. The offshore operation environment is much harsher and creates new technical challenges. The support structures are exposed to challenges related to water depth and soil quality. Remoteness of wind farms causes the investment and maintenance cost to increase which requires the turbines to be much more reliable. Offshore wind farms typically consist of the largest available turbines on the market.

Offshore wind turbine technology is still considered to be relatively immature, as the first demonstration project was established in 1991 and the total installed capacity barrier of 1000MW was reached in 2007/2008. Increased interest in the field has led to many interesting projects, initiated by both private and government founded projects.

Figure 1.1 - Aeral view of Lillgrund Wind Farm, Sweden [3]

The current driving factors specific for offshore wind turbines are

- Nacelle mass reduction
- Large rotor technology and advanced composite engineering
- Foundation design

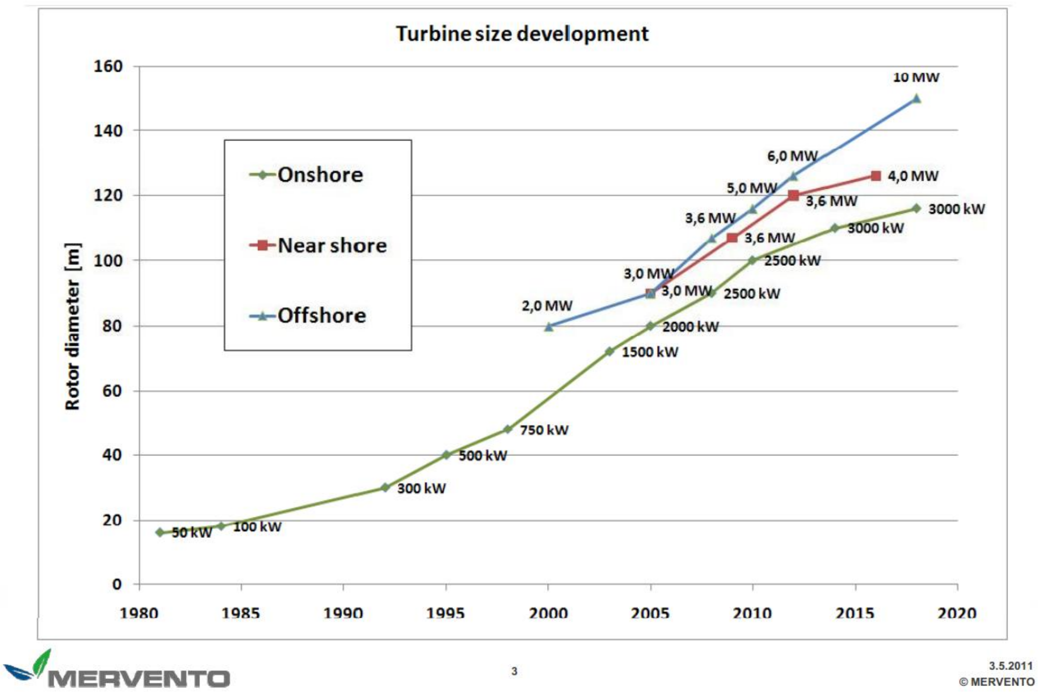


Figure 1.2 - Rotor diameter trend [4]

New foundation and support structure solutions aims to enable large new deep water wind farms. Investment costs are cut by reducing the mass of the rotor nacelle assembly (RNA) through the development of new and lighter composites for the rotor blades. The nacelle is optimized introducing innovative solutions which requires less components resulting in higher reliability.

Offshore wind turbines increase their energy yield by having larger rotors, as illustrated in figure 1.2. This trend continues as new and lighter materials allow increasingly larger rotors blades.

Direct drive permanent magnet generators have become popular as they have an increased reliability and higher efficiency compared to traditional generators which operate at higher speeds and requires more components. Advantages and disadvantages for direct drive (DD) generators are summarized in table 1.2.

Table 1.2 - Direct drive generators [4]

Advantages of DD	Disadvantages of DD
• No gearbox and related wear an tear on mechanical components	• Larger diameter of generator/nacelle complicates transportation and installation.
• Simpler turbine with fewer parts	• Higher top-mass weight *)
• Higher electrical (PM) and overall drive train efficiency, producing higher energy yield	• Expensive PM material – potentially uncertain security of supply
• Lower maintenance and greater reliability with less downtime	• More complex assembly of generator using PMs
• Improved thermal characteristics due to absence of field losses	• Demagnetisation of PM at high temperature
• Full power conversion improves the turbine's grid compatibility	• More advanced cooling system required
	• Full power rather than partial power conversion makes the turbine more expensive
Comments: *This comparison is based on a traditional drive train comprising a doubly feed induction generator and a 3-4 stage gearbox.* **) The general trend towards higher top-mass weight for direct drive turbines seems to have been broken by the new DD turbine SWT 3.0 from Siemens Wind Power* *When comparing the PMG solution to the Enercon DD design, the latter is heavier and the absence of a permanent magnet creates excitation loss when magnetising the coils, but the concept has an impressive track record from more than 20 GW of capacity in operation.*	

Source: BTM Consult - A Part of Navigant Consulting - March 2011

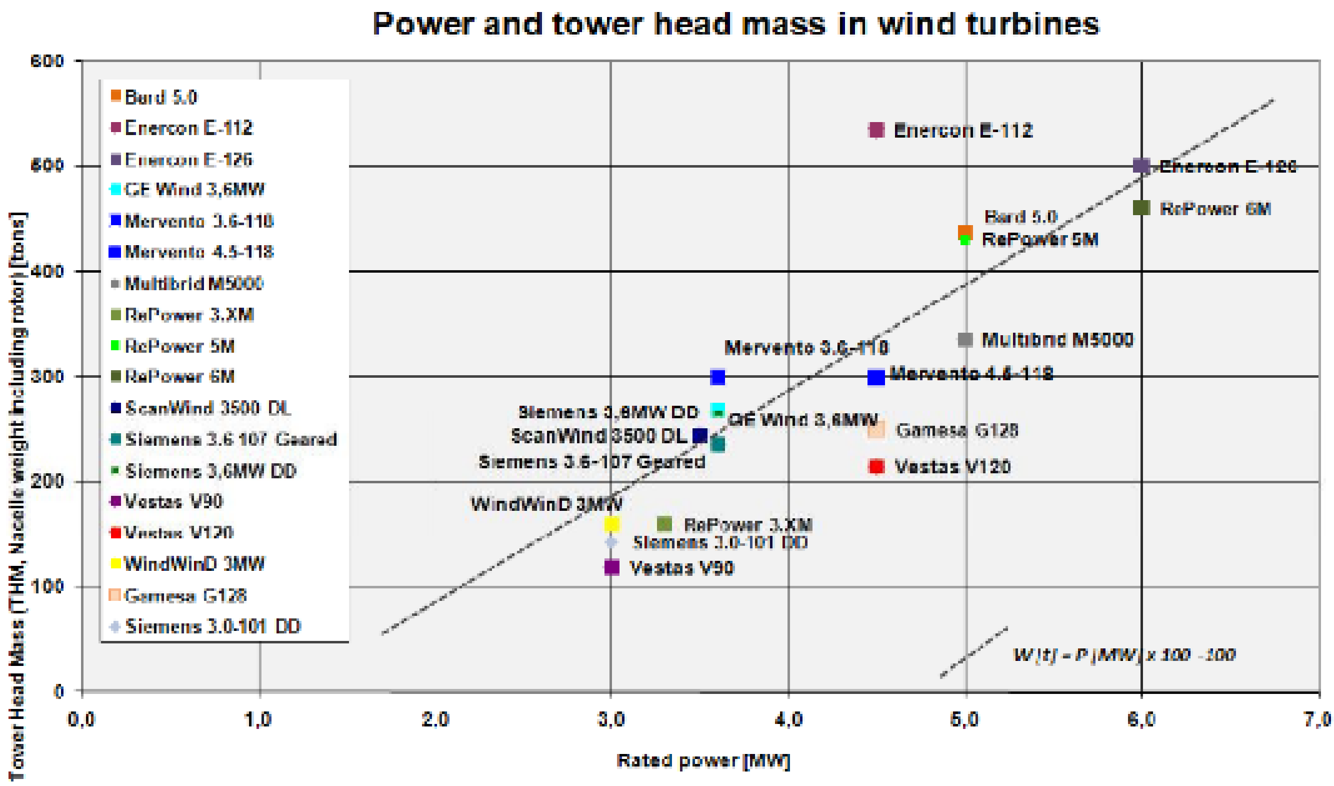


Figure 1.3 - Tower head mass in wind turbines [4]

Tower head mass estimations for modern wind turbines are presented in figure 1.3. The tower head mass seems to follow a linear relationship between rated power and mass. Initial investment cost (IIC) increases as larger and heavier components complicates manufacturing, installation and assembly. The rated power and the low operating costs continues to drive this trend forward. In order to both reduce investment risk and further the competitiveness of wind energy the tower head mass plays a significant role.

1.1 NOWITECH 10MW Reference turbine

The design process of a wind turbine is a multidisciplinary engineering task, and it is difficult to get enough information from manufacturers to create comprehensive and independent studies. The NOWITECH reference turbine is a multidisciplinary cooperation with NTNU in order to create a state of the art wind turbine design and create a platform in which students at NTNU and researchers within NOWITECH can collaborate. The project makes it possible to create detailed studies within an open project where information about every detail regarding the turbine design is accessible.

The wind turbine is a 10MW is a three bladed horizontal axis wind offshore turbine that has a bottom fixed foundation at 60meter water depth [5]. The nacelle is based on an state of the direct drive outer rotor generator design. This master thesis will review and modify rotor nacelle components of the turbine.

Figure 1.4 - NOWITECH 10MW Reference Wind Turbine

1.2 Structure

The remaining text is intended to be read in order, and is structured as follows:

- In chapter 2, the design basis for the wind turbine is presented. This includes a detailed description of the environmental conditions and design considerations based on previous work for the rotor nacelle assembly.

- Chapter 3 presents safety factors and design load cases (DLC) considered for dimensioning from relevant IEC standards.

- Chapter 4 shows how the loads from the design load cases was obtained and applied to the main shaft bearings and the bedplate. This chapter aims to provide transparency in how the FEM simulations was developed.

- In chapter 5, the design process of the bedplate from the naïve implementation to the optimized design is shown.

- In chapter 6, the selection of main shaft bearings based on the SKF's product catalog is presented.

- In chapter 7, results from the ultimate strength analysis based on the cases described in chapter 5 are performed on the optimized design presented in chapter 6.

- Finally, in chapter 8 the results are summarized in the conclusion and some ideas for future work is outlined.

2. Design basis

The following text is partly taken from the specification of the NOWITECH 10MW reference wind turbine [5].

"The NOWITECH reference wind turbine is designed for a fictitious wind farm located offshore in the North Sea, characterized by strong average winds, low wind shear and low turbulence. The reference site is chosen to be relevant for the wind farms that will be installed at Doggerbank. The main design basis is presented in Table 2.1.

The turbine is designed according to IEC design class I_C. The class I wind parameters represent a strong wind regime with reference wind speed of 50 m/s. The turbulence parameters are chosen according to turbulence category C, which is representative for low turbulence locations similar to those found offshore. The choice of design parameters have been adapted from UpWind project [6], which investigated turbine design on a deep water depth location offshore on the Dutch continental shelf.

The turbine is intended to be placed within a wind farm, which means that the effective turbulence will be higher than in the ambient wind flow. This will likely cause an increase in blade fatigue, especially in the flapwise direction. However, because wind speed are generally lower within a wind farm, it is not assumed that the maximum operating gusts or extreme wind speeds will increase. "

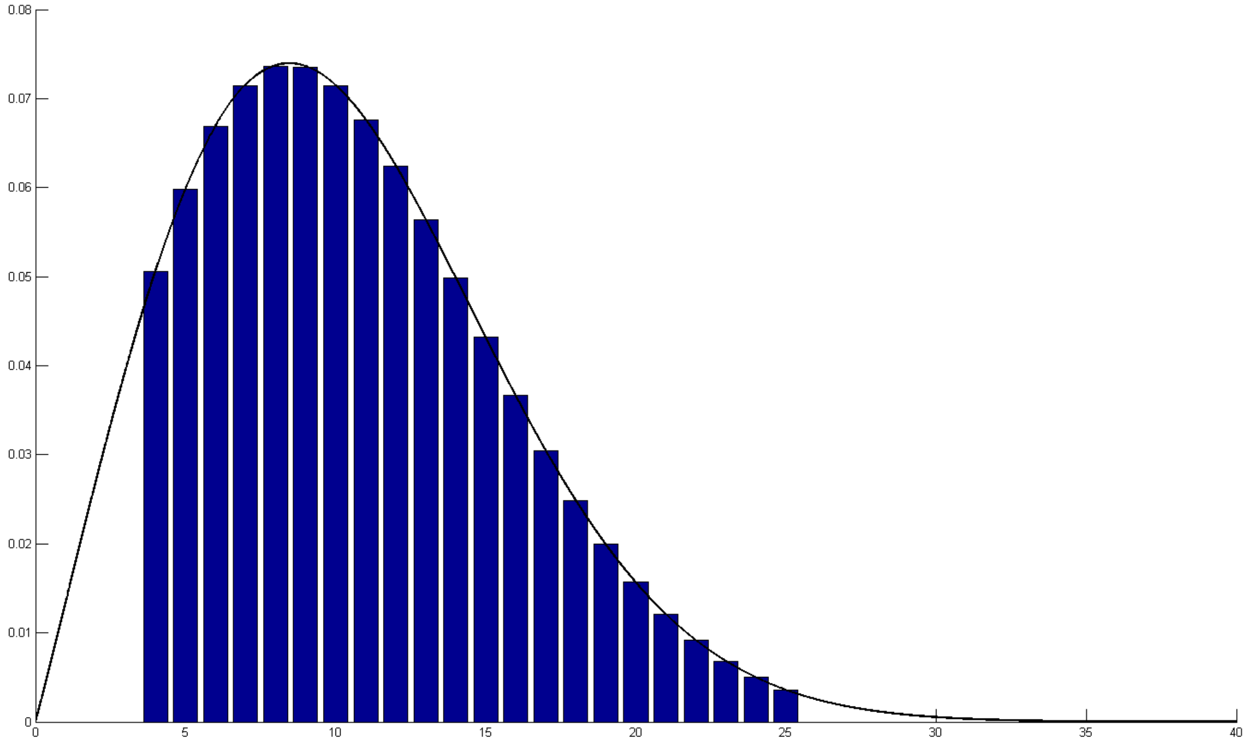


Figure 2.1 - Wind speed distribution from the UpWind weibull parameters, graph provided by Peter Fossum. Vertical axis: Probability [%], Horizontal axis: wind speed [m/s]

Table 2.1. Main design criteria [5]

	Symbol		Unit
Extreme wave height		30	m
Maximum sea current velocity		1.2	m/s
Water depth		60	m
Rated power output	P	10	MW
Electrical Frequency	f_n	50	Hz
Weibull parameter	A	11.75	-
Weibull parameter	k	2.04	-
Density of air	ρ_{air}	1.225	kg/m^3
Density of seawater	ρ_{sea}	1025	kg/m^3
Water salinity		3.5	%
Water temperature (min/max)		0 / 22	^oC
IEC turbulence parameter	I_{ref}	0.12	-
IEC reference wind speed	U_{ref}	50	m/s
Average wind speed at hub height	U_{ave}	10.4	m/s
IEC wind shear exponent	α	0.14	-
Rotor diameter	D	141	m
Number of blades		3	-
Hub diameter	D_{Hub}	4.94	m
Length between blade tip and the tower	L	13	m
Maximum rotor speed	n	13.54	rpm
Maximum allowed tip speed		100	m/s
Extreme wind speed, 50 years	U_{e50}	70	m/s
Extreme wind speed, 1 year	U_{e1}	56	m/s
Design wind speed	U_{Design}	13	m/s
Rated wind speed	U_{Rated}	~ 15	m/s
Cut-in wind speed	U_{Cut-in}	4	m/s
Cut-out wind speed	$U_{Cut-out}$	30	m/s
Optimum tip speed ratio	TSR_{Opt}	7.8	-
Blade pre-curvature		3.06	m
Turbine blade coning angle		2	degrees
Main shaft tilt angle		5	degrees

2.1 Rotor nacelle concept

The rotor nacelle assembly was selected from a collage of direct drive outer rotor designs presented by Mervento, shown in figure 2.2. The designs presented are based on current implementations in the industry. All the solutions have the generator mounted upstream of the tower.

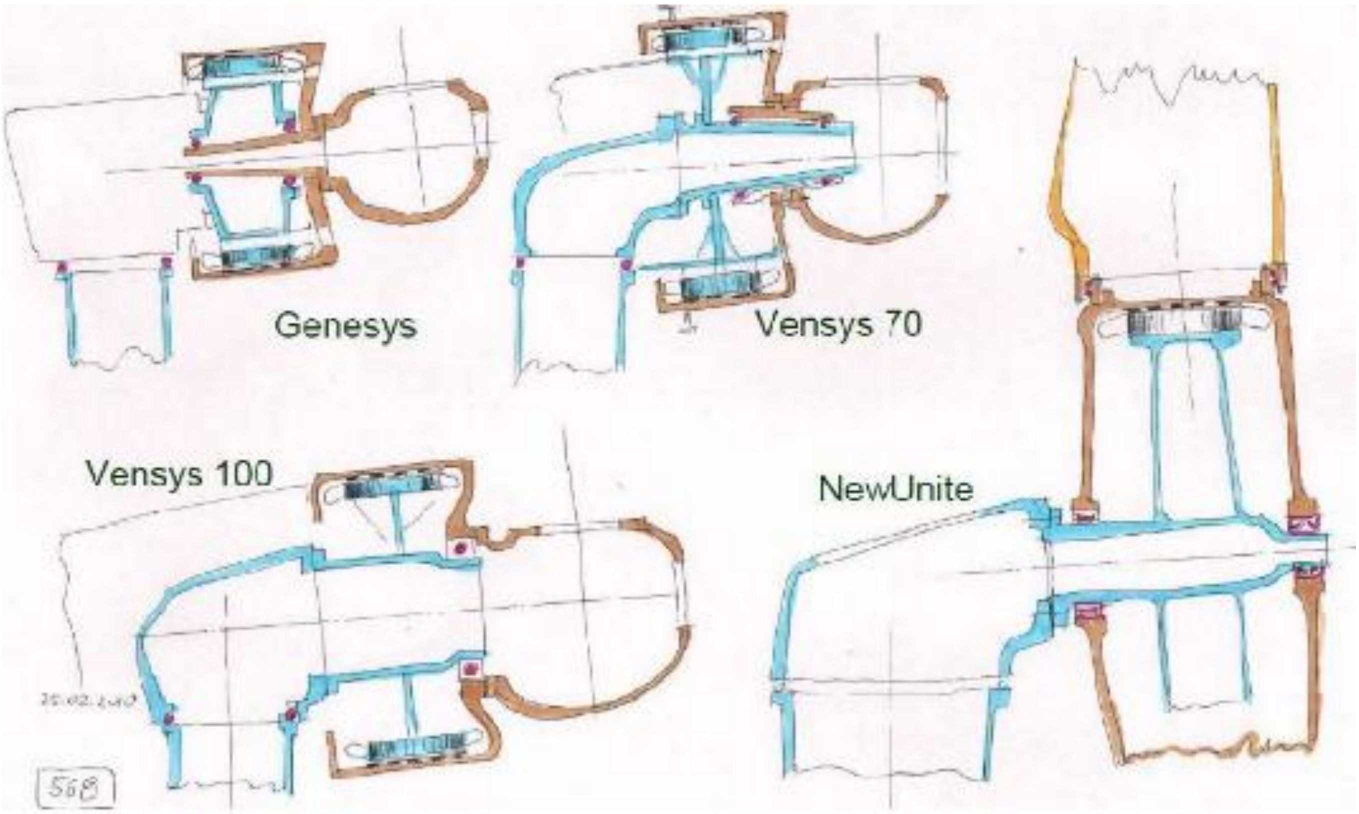


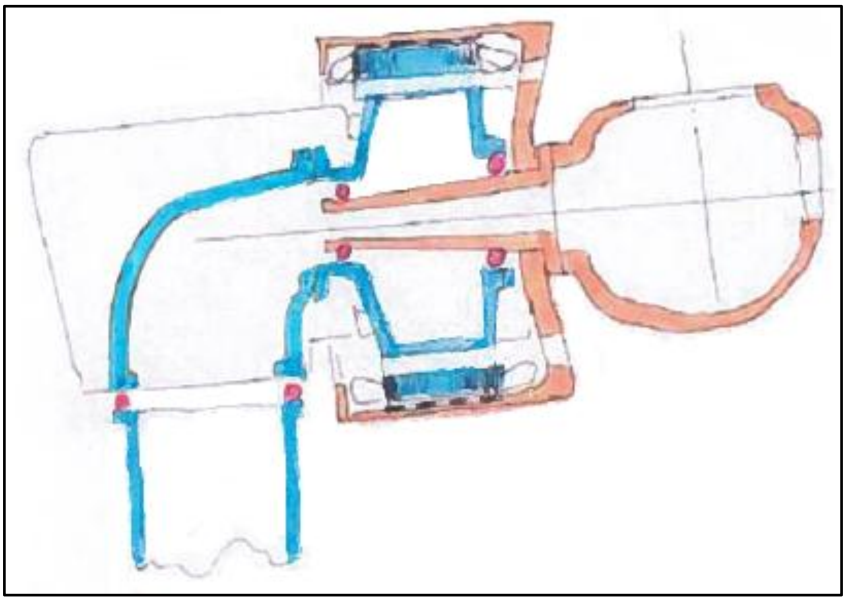
Figure 2.2 - Direct drive outer rotor concepts [4]

The outer designs allows for a bigger generator than the alternative approach where the generator is mounted on the inside.

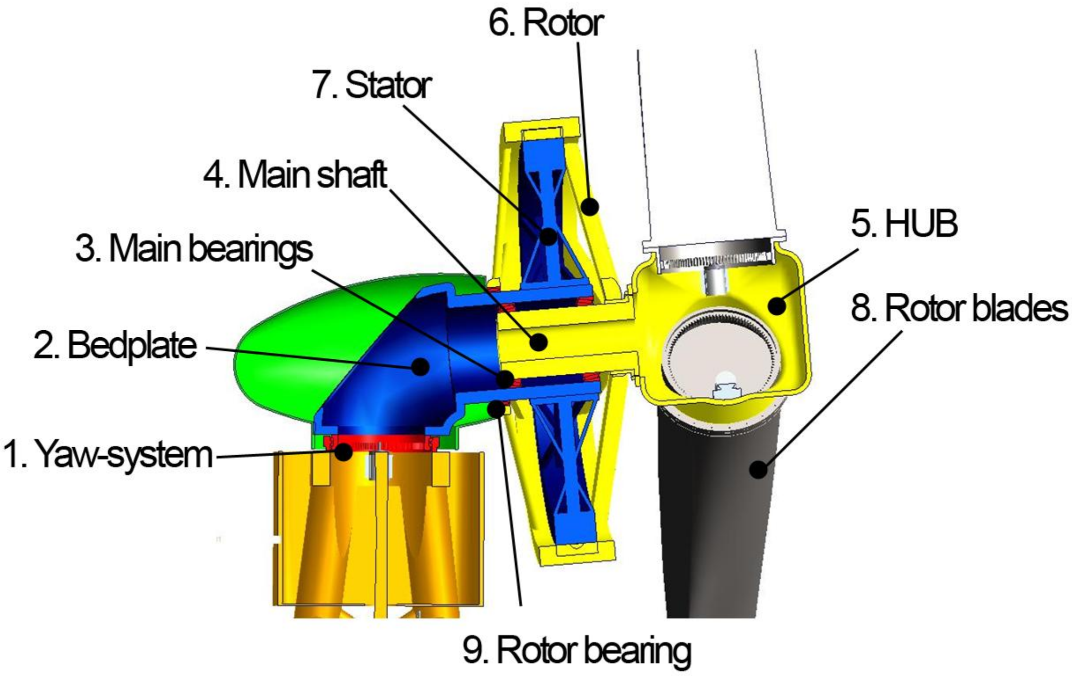
The selected solution was the Gensys principle, shown in figure 2.3, which requires a main shaft and two main shaft bearings. This principle requires a bedplate to fully support the

- generator
- rotor blades
- hub
- main shaft
- various components inside the nacelle

The design must mount the yaw system to the lattice tower.



Figur 2.3 - Selected nacelle assembly [4]



Figur 2.4 - Nacelle assembly explained

The rotor nacelle layout consists of:

1. Yaw-system - *aligns the turbine upwinds.*
2. Bedplate - *supports the whole assembly (except yaw) and loads the tower.*
3. Main bearings - *supports the mainshaft*
4. Main shaft - *supports the hub and translate torque to the rotor.*
5. Hub - supports the rotor blades, and a pitch motor for the blades.
6. Rotor - *attatched to the mainshaft*
7. Stator - *Electrical generator, a radial flux permanent magnet synchronous generator*
8. Rotor blades - *Converts kinetic energy into mechanical energy*
9. Rotor bearing - *supports the mainshaft*

2.3 Estimations and assumptions

The generator suggested by Hilde Liseth [7] is a compact generator with an outer diameter of 12m, stack length of 1.2m and an efficiency of 96.2%. The weight of the generator is estimated to be 200tons in a worst case scenario, where 60tons are expected to be active materials. The active materials is assumed to be the generator rotor, and inactive materials as the generator stator.

The specification [5] states that the generator is 310tons, a contradicting estimate compared to those proposed by Hilde Liseth [7]. Conservative estimates including a 60ton rotor and a 200ton stator have been selected.

The rotor blades are currently estimated to weigh 27.8tons and the hub weight has been estimated to three times the rotor blade weight.

The tower head mass totals to nearly 600tons which is considered to be very light compared to the tower head mass trend shown in figure 1.3.

Table 2.2 - Preliminary weight estimats

Weight estimates	
Component	Weight [tons]
Generator rotor	60
Generator stator	200
HUB	83,6
Bedplate	100
Main Shaft	45
Rotorblade	27.5
Fixed bearing	11
Floating bearing	3
Rotor bearing	3

2.3.1 Geometrical considerations

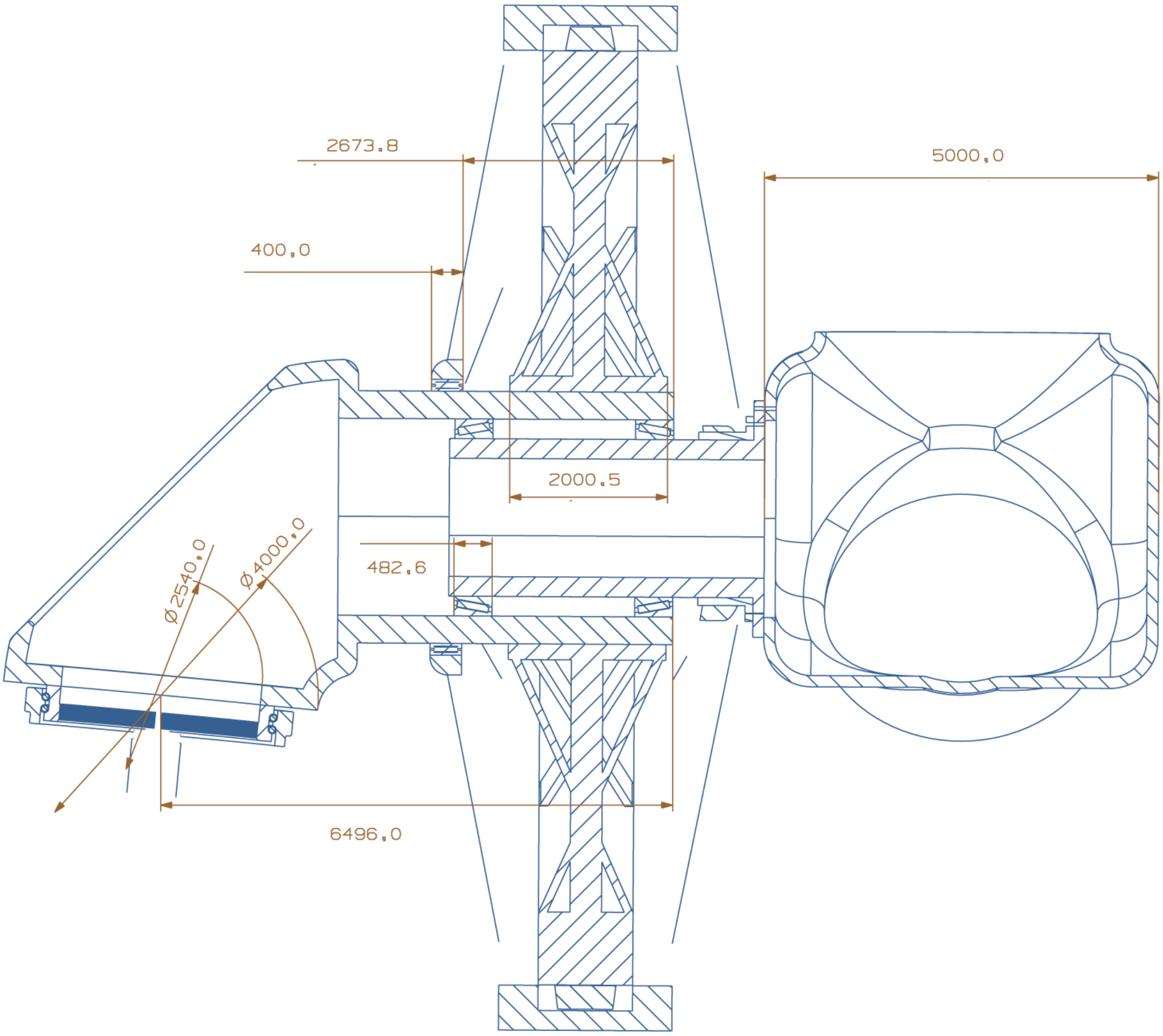
Geometrical constraints are derived from the naïve implementation to ensure

- That the rotor blades won't crash into the tower
- Generator compatibility

The parameters listed in table 2.3 are derived from the naïve rotor nacelle assembly provided by NOWITECH (see figure 2.4) and illustrated in figure 2.5.

Table 2.3 - Geometrical constraints

Distance Description	Length [mm]
Base to nose (origo to origo)	**6500**
Outer nose diameter	3200
Stator contact surface	2000
Rotor bearing distance from nose	2700
Base Inner diameter	2540
Base outer diameter	4000
Main bearing seperation (max)	4700
HUB length (front to back)	5000



2.4 Summary

The reference turbine will be designed according to the IEC design class I_C. Environmental parameters are derived from the UpWind project [6] which provides design parameters for conditions measured at the K13-Alpha site (Doggerbank) at 90.4m height. These design parameters will be used for load estimation using aero elastic analysis, described chapter 3.

The rotor nacelle assembly is still in an early development phase. Specifications for important rotor nacelle components such as the generator is still in a preliminary state and conservative estimations provide the basis for preliminary gravitational load estimation for the bedplate. Partial dimensions for the outer nose profile and bottom base has been established as design requirements in order to ensure compatibility for the generator and rotor.

3. Structural Design

3.1 Design Method

Ultimate loading of wind turbines [8] states that "Verification of the structural integrity of wind turbine structures involves analysis of fatigue loading as well as extreme loading arising from the environmental wind climate. With the trend of persistently growing turbines, the extreme loading seems to become relatively more important."

The design is developed by a max stress criteria (design load effect) according to ultimate loads obtained by design load cases from IEC61400-3:2008 - Design requirements for offshore wind turbines [1]. Development of the structural design was done by direct simulation of combined load effects of simultaneous load processes. The following subchapters explains how both the design load and design loads effects are determined in addition to resulting requirements for material selection.

3.1.1 Design Format

The safety level of a structural component is considered satisfactory when the design load effect S_d does not exceed the design resistance R_d [1]

$$S_d \leq R_d \tag{0.1}$$

3.1.2 Design load effect

The design load effect S_d is established by structural analysis according to the second approach described in design requirements for offshore wind turbines [1]. This approach establishes the design load effect as a result of a structural analysis where the design load F_d is obtained by multiplication with the characteristic loads F_k and the specified partial safety factor γ_m specified in IEC61400-3:2008 - Design requirements for offshore wind turbines[1]

$$F_d = \gamma_f F_k \tag{0.2}$$

The alternative approach is to multiply the characteristic load effect with the partial load factor. The selected approach is usually applies to the design load effects for the support structure, which the bedplate structure is considered to be. Both approaches are illustrated in figure 3.1.

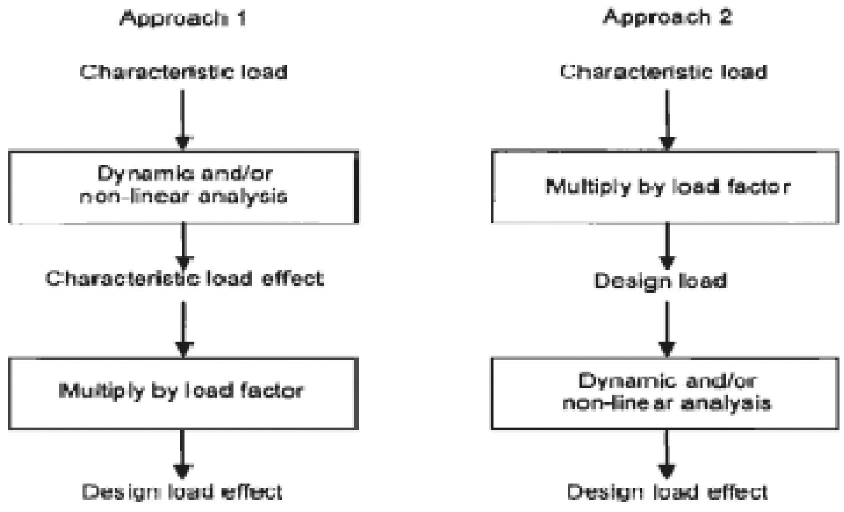


Figure 3.1 - Two approaches to calculate the design load effect [1].

3.1.3 Design load resistance

Design requirements for wind turbines [1] determines the design load resistance from the characteristic resistance for the particular component as

$$R_d = \frac{1}{\gamma_m} R_k \tag{0.3}$$

with γ_m as the material safety factor for the component and R_k as the characteristic resistance. The characteristic resistance is the ultimate load with a probability factor applied. Alternatively, the design load resistance can be determined from characteristic material strength f_k

$$R_d = R(\frac{1}{\gamma_m} f_k) \tag{0.4}$$

3.2 Ultimate strength analysis

Every wind turbine structure must satisfy the limit state function given in (0.1). As the resistance generally corresponds with material strength and the design load effect (shown in equation 0.2) from the maximum structural response from a combination of the highest characteristic loads the limit state function can be written as

$$\gamma_f F_k \leq \frac{1}{\gamma_m \gamma_n} f_k \tag{0.5}$$

where γ_n is the failure factor. The ultimate and characteristic loads are obtained by aero elastic analysis according to design load cases specified in the standard [1]. The selected design load cases are described later in this chapter.

3.2.1 Partial safety factors

The load safety factor account for [1]:

- Possible unfavourable deviations of the loads from their characteristic values.
- The limited probability that different loads exceed their respective characteristic values simultaneously.
- Uncertainties in the model and analysis used for determination of load effects.

The material load factor account for [1]:

- Possible unfavourable deviations in the resistance of materials from the characteristic value.
- Uncertainties in the model and analysis used for determination of resistance.
- A possibly lower characteristic resistance of the materials in the structure, as a whole, as compared with the characteristic values interpreted from test specimens.

The partial load safety factor γ_f for normal design situations are [1]

- 1,35 for unfavourable loads
- 0,9 for favourable loads

where favourable loads are considered those gravitational loads and pretensions which relieve the total load response on the structure.

The general partial safety factor for materials must satisfy $\gamma_m \geq 1{,}1$ to account for material variability.

Partial safety factor for component class 1 is $\gamma_n = 0{,}9$

3.2.1 von Mises yield criterion

"In materials science and engineering the von Mises yield criterion can be formulated in terms of the von Mises stress, σ_j, a scalar stress value that can be computed from the stress tensor. In this case, a material is said to start yielding when its von Mises stress reaches a critical value konwn as the yield strength, f_y. The von Mises stress is used to predict yielding of materials under nay loading condition from results of simple uniaxial tensile test. The von Mises stress satisfies the property that two stress stats with equal distortion energy have equal von Mises stress" [9]

The criterion for yielding [10] is

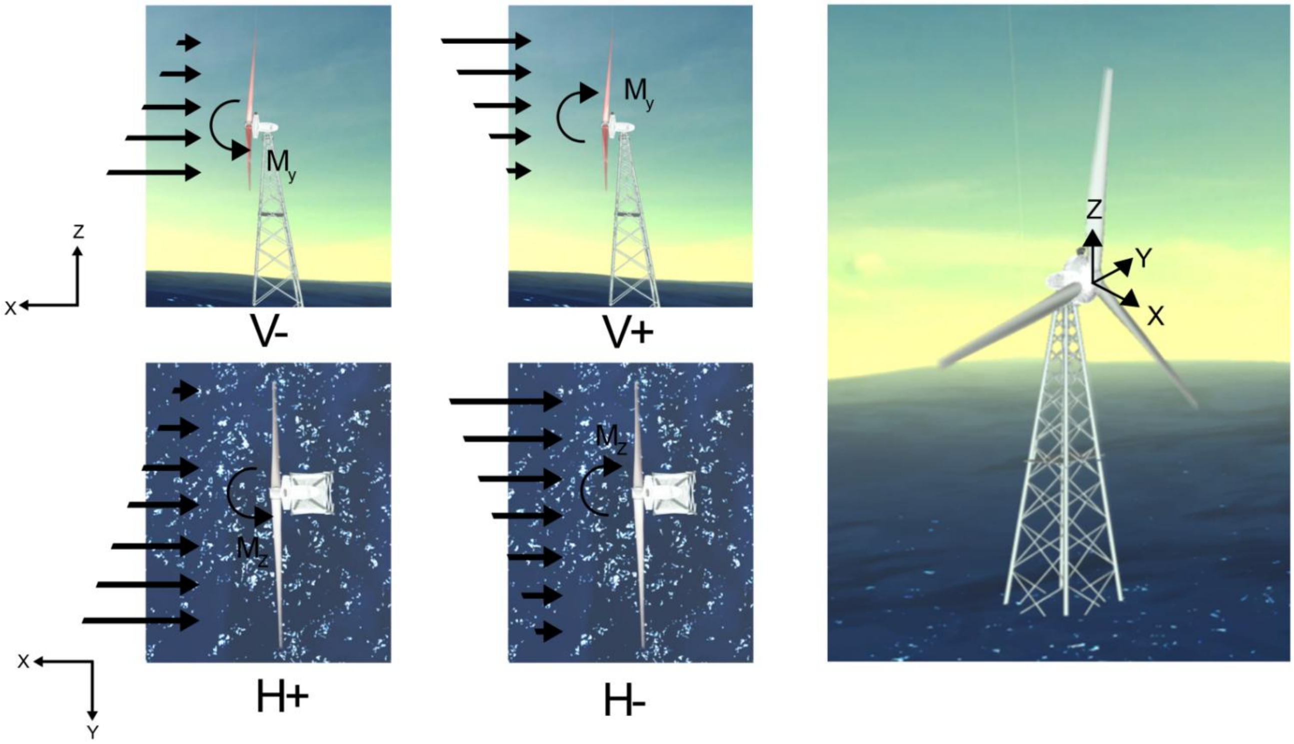
$$\sigma_j = f_y \tag{0.6}$$

where

$$\sigma_j = \sqrt{\sigma_x^2 + \sigma_y^2 - \sigma_x \sigma_y + 3\tau_{xy}^2} \tag{0.7}$$

The bedplate will be made out of cast steel, the grade of the steel has not yet been decided. It has been assumed that a reasonable limit for the von Mises stress is 200MPa.

3.3 Design load cases



Figur 3.2 - Aerodynamic loads on the wind turbine

Design load cases (DLC) are a combination of a specified operation situation for the wind turbine combined with specific environmental conditions (wind and waves for offshore). The IEC standard [1] consist of several hundred load cases (including the required wind speeds). The selected load cases for dimensioning are listed in table 3.1.

Table 3.1 - Selected design load cases

DLC	Wind Condition	Wind Speed Conditions	Type of analysis
N/A	No wind	n/a	Ultimate Strength
1.1	NTM[1]	13 and EXT	Ultimate Strength
1.3	ETM[2]	13 and EXT	Ultimate Strength

These design load cases represent aerodynamic loading (see figure 3.2) under normal and extreme turbulence conditions. Based on DLC 1.1, extreme loads for power production with the normal

[1] NTM - Normal turbulence model
[2] ETM - Extreme turbulence model

turbulence model (NTM) is extrapolated to 50-year return period loads. This extrapolation is represented in the partial load safety factor. Extreme turbulence model (ETM) have a 50 year return period, loads from these conditions are additionally extrapolated with the partial safety load factor.

Two wind speed conditions is evaluated for each design load case:

- 13- The characteristic loads equals the highest transient loads obtained from the aero elastic simulation at 13m/s.
- EXT - The ultimate loads are a conservative combination of the highest transient loads recorded for all wind speeds in the given design load case. These values were derived from the wind speeds 28-30 m/s.

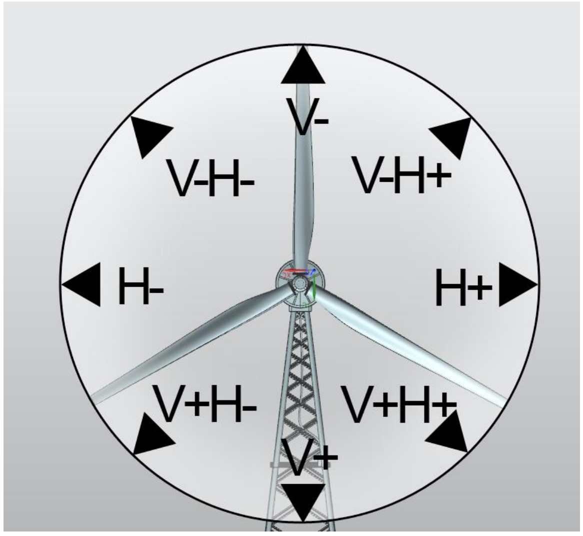


Figure 3.3 - Axial loading

3.3.1 Aero elastic analysis

Loads from the design load cases is obtained with aero elastic simulations using site and turbine specific parameters. The simulations was conducted with the simulation software FAST [11]. Statistics from these simulations obtained through the batch-style postprocessor crunch[3].

These simulations satisfy the statistical reliability required by the IEC standard where the simulations has to cover either six 10-min stochastic realizations or a 60-min continuous time period, the latter method was used.

The FAST simulations was conducted with the following known deviations from the design basis

- Wind turbine class IB (foot: I_{ref} = 0,14 instead of 0,12 (IC))
- Infinite stiff main shaft

The increased turbulence is intended to represent the expected conditions in an offshore wind farm, where the turbulence is expected to be higher than in the ambient air flow.

Aerodynamic loading on the rotor creates three forces and three moments which is transmitted directly over to the main shaft (shown in figure 3.4). These moments and forces was obtained with a virtual strain gauge simulated in the center of the hub, the sensors specified in the FAST user manual [11] are identified as

- LSShftFxa - thrust on rotor
- LSShftMxa - driving torque at rotor
- LSShftFys - side force on rotor and hub
- LSShftMys - tilting moment at rotor
- LSShftFzs - weight of rotor and hub

[3] http://wind.nrel.gov/designcodes/postprocessors/crunch/

- LSShftMzs - yaw moment at rotor

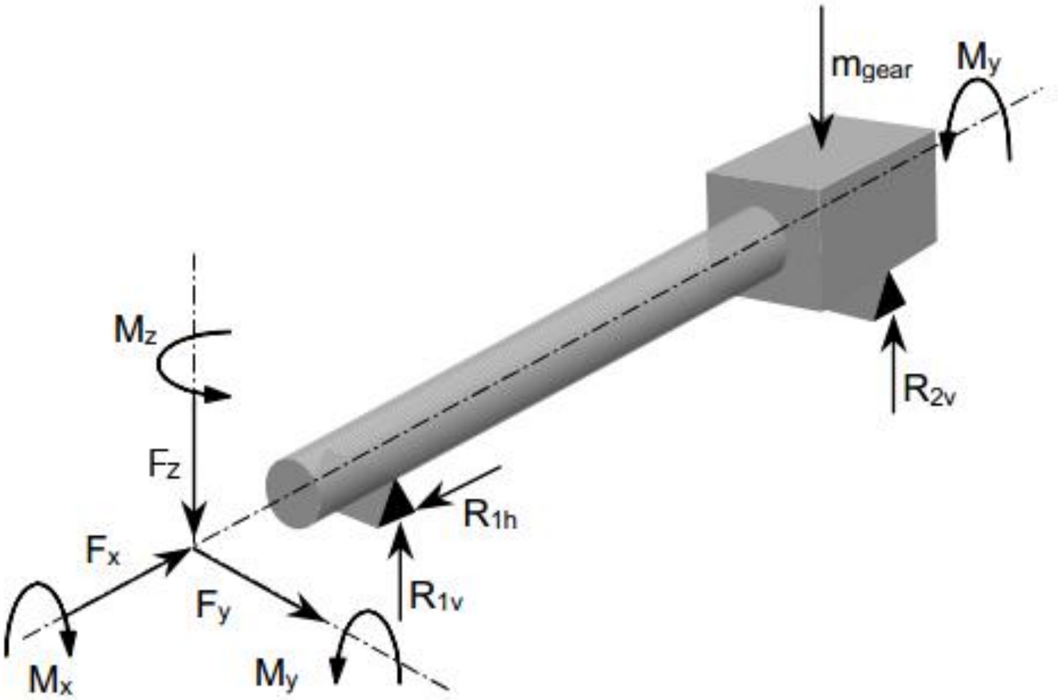


Figure 3.4 - Loads and reactions on the main shaft [12]

A combination of the obtained loads are applied to the structure to create the multi-axial loading which creates the most unfavourable load cases. Figure 3.3 shows load directions, where the diagonal directions represents multi-axial loading.

3.4 Summary

This chapter presented the design load cases used for verification of the structural integrity for the bedplate and main shaft bearings. In addition to equations for material selection directly related to design load effects.

The dimensioning load cases are created as a result of worst-case multi-axial loads obtained from extrapolated aerodynamic loads representing load with a 50 year recurrence period. Additional design load cases provide a wider sample range that contains characteristic loads derived from operation at the turbine design speed (13 m/s).

4. Load determination

4.1 Generator gravitational loads

The generator is assumed to transmit the driving torque directly to the bedplate structure as well gravitational loads. The generator rotor is assumed to load the main shaft and bedplate frame equally. It has been assumed that the bedplate will weigh 100tons (see table 2.2). Center of gravity for the bedplate was obtained with the "measure body" function in NX, and was found to be very close to the backend bearing. Bedplate weight will be applied as a bearing load in the back bearing, as a simplification. The characteristic loads caused by the generator on the bedplate are:

$$T_{Generator} = -M_{xR}$$

$$F_{Rotor} = 30tons * 9.81\frac{m}{s^2} = 294,3kN$$

$$F_{Stator} = 200tons * 9.81\frac{m}{s^2} = 1962kN$$

$$F_{Bedplate} = 100tons * 9.81\frac{m}{s^2} = 981kN$$

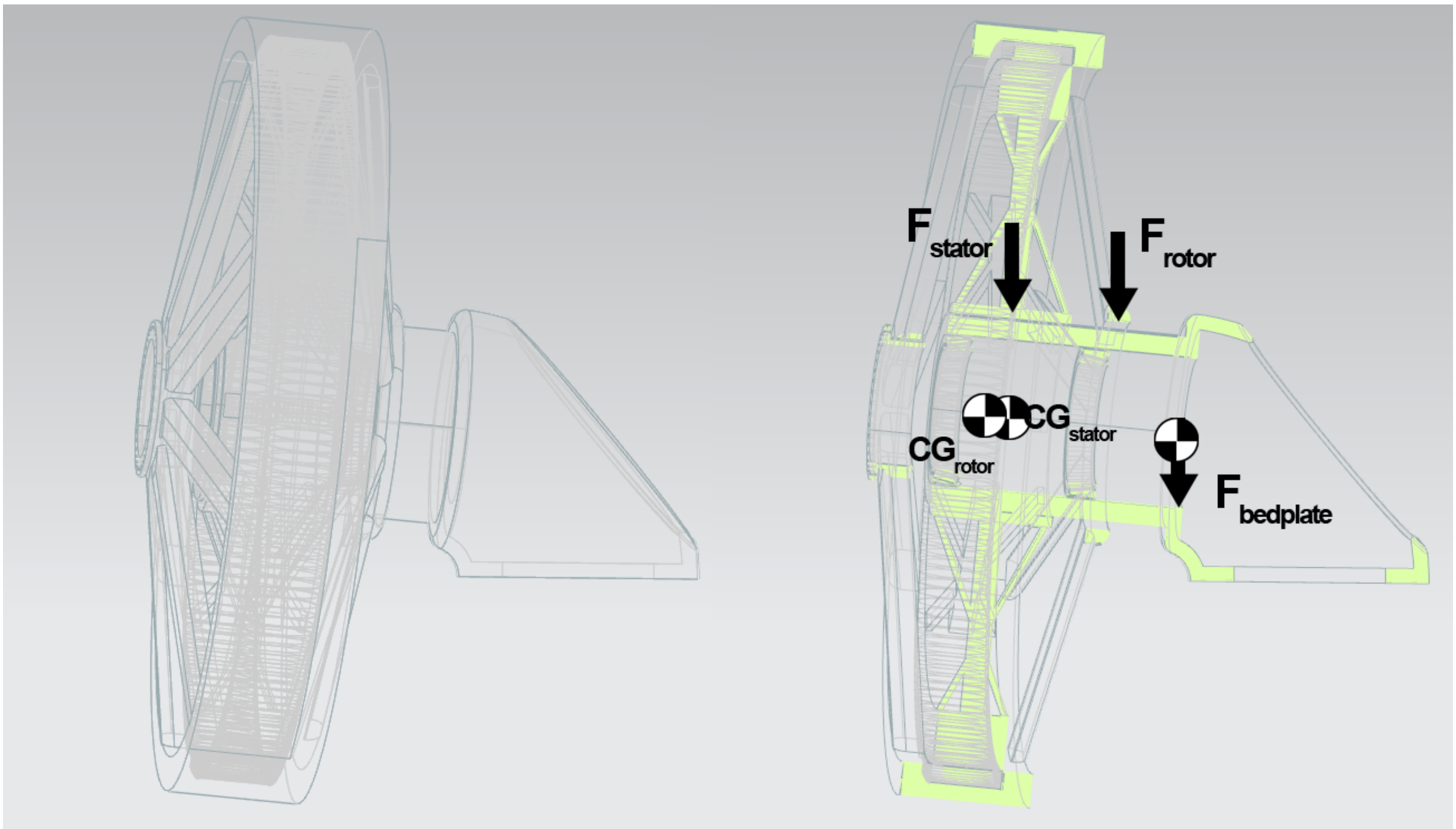
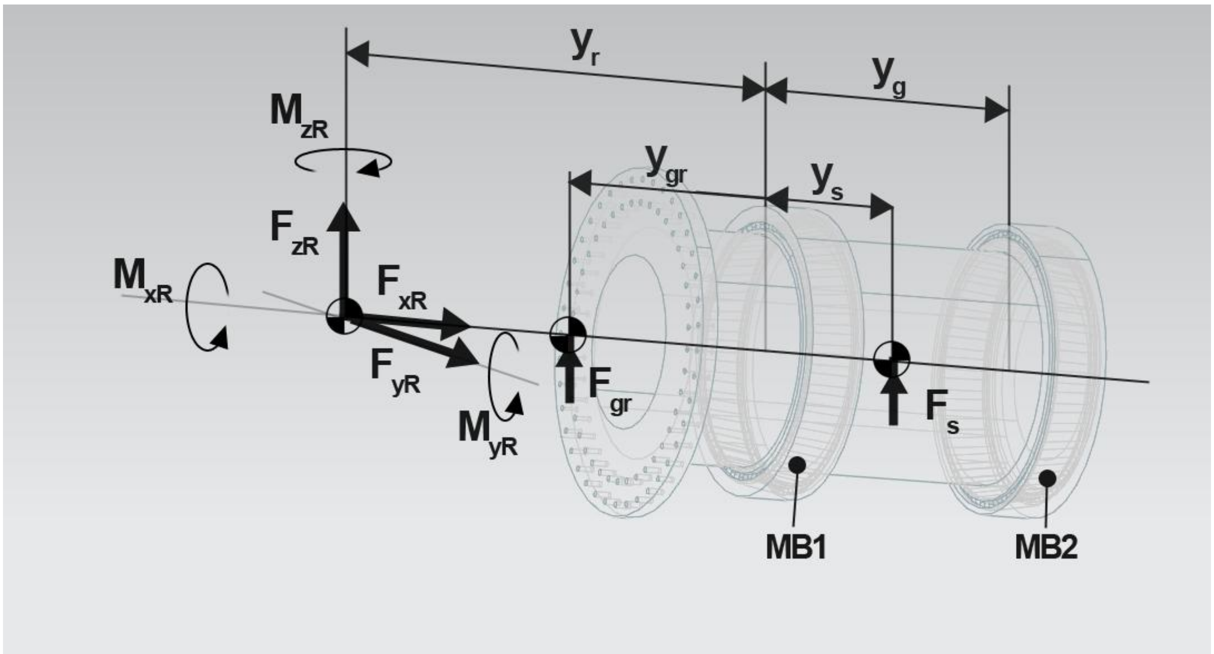


Figure 4.1 - Generator loads

4.2 Main bearing loads

The characteristic loads caused by the specified DLCs recorded at the hub center are available in appendix A. These loads follows the coordinate system shown in figure 5.1 . Bearing loads are identified as the DNV guidelines for Wind Turbines [12] suggest, modified to two bearings and and a generator rotor load on the main shaft.



Figur 4.2 - Bearing loads

y_r - distance rotor center to bearing center
y_g - distance front bearing (MB1) to back bearing(MB2)
y_gr -distance generator rotor to front bearing (MB1) center
y_s - distance shaft center of gravity to front bearing (MB1)
F_xR - thrust on rotor
F_yR - side force on rotor and nacelle
F_zR - weight of rotor and hub
F_gr - partial weight of generator rotor
F_s - weight of main shaft
M_xR - driving torque at rotor
M_yR - tilting moment at rotor

The loads from the aero elastic simulations are converted into bearing forces, which consists of one axial force and two radial forces. These forces are assumed to be directly transmitted to the bedplate structure.

Bearing load calculation follows the method described in DNV guideline for wind turbine design [12]. The radial force for the back bearing is calculated as

$$F_{r2} = \frac{1}{y_g}\sqrt{M_1^2 + M_2^2} \tag{0.8}$$

with

$$M_{21} = M_{yR} + F_{zR} * y_r + F_{gr} * y_{gr} - F_s * y_s \tag{0.9}$$

$$M_{22} = M_{zR} + F_{yR} * y_r \tag{0.10}$$

The front bearing :

$$F_a = -F_{xR} \tag{0.11}$$

$$F_{r1} = F_{r2} + \sqrt{F_{xR}^2 + F_{zR}^2} \tag{0.12}$$

These loads are converted into design loads using the partial load safety factors described in chapter 3. Detailed calculations for the no-wind condition and ultimate loads in DLC 1.3 are available in appendix C with a complete compilations of the design loads in appendix B.

4.3 Calculations by Finite Element Method

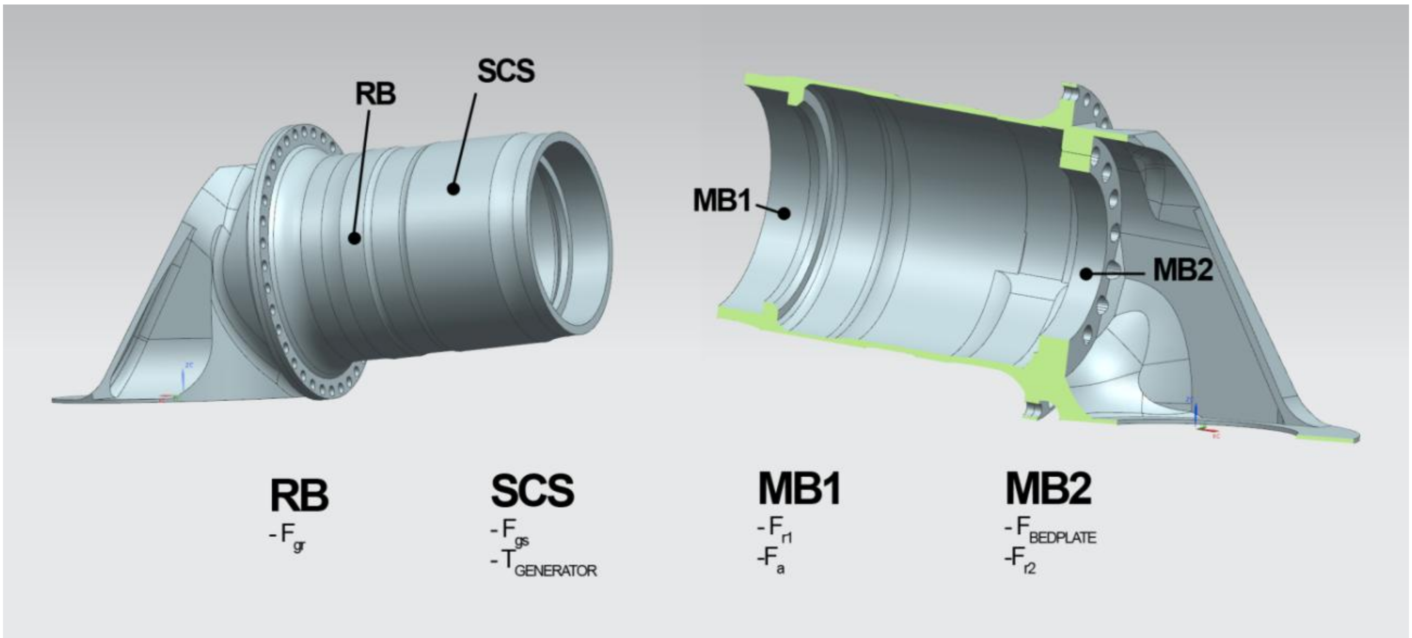


Figure 4.3 - Applied loads

The loads mentioned in this chapter are applied to the bedplate as illustrated in figure 4.3. They are applied to the structure as bearing loads, except the torque and rotor thrust which are applied as a force and a torque load. The radial bearing loads for the main shaft bearings are separated into vertical and horizontal components, as shown in figure 4.4. The design load effects for the various load variations are established by finite element analysis as shown in figure 4.5.

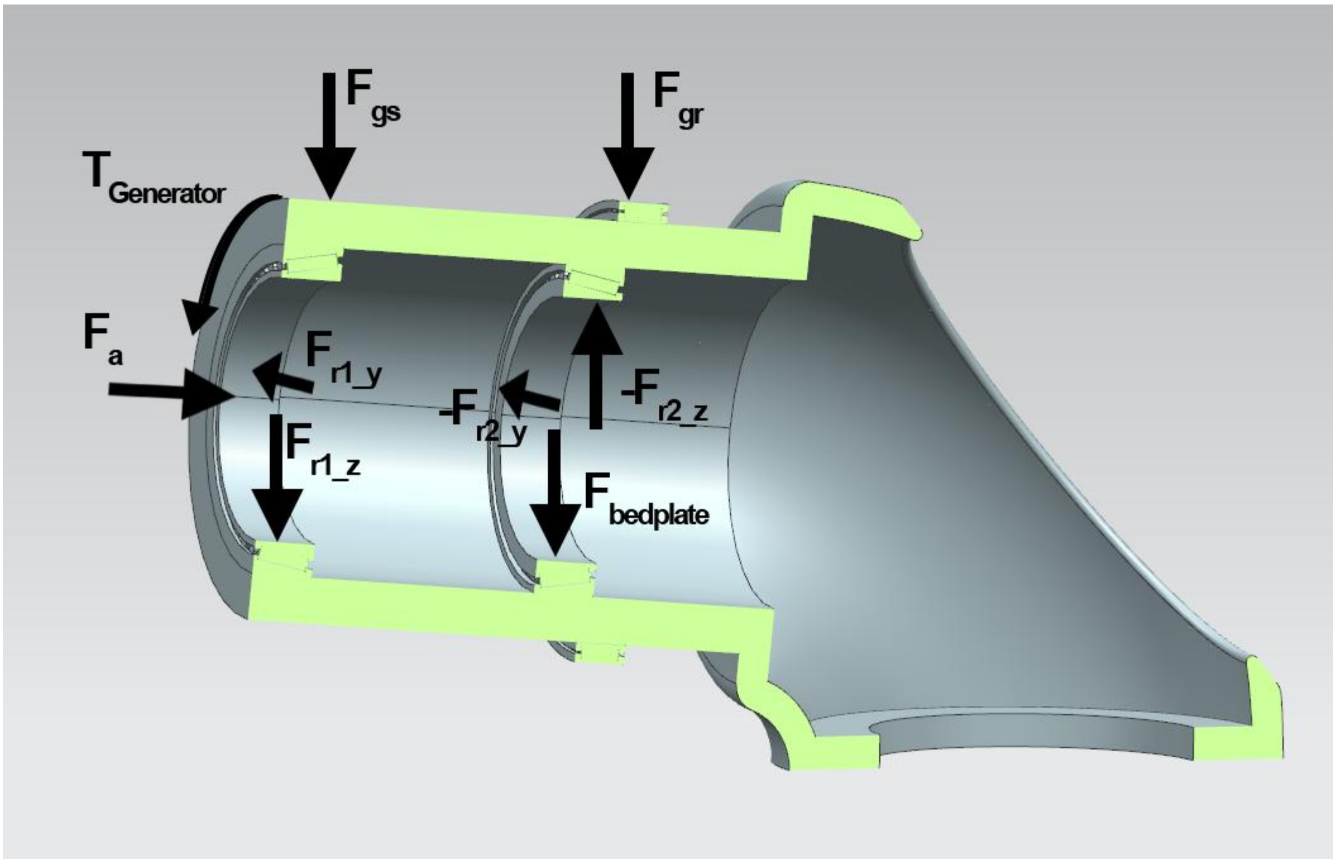


Figure 4.4 - Summary of individual load components on the bedplate

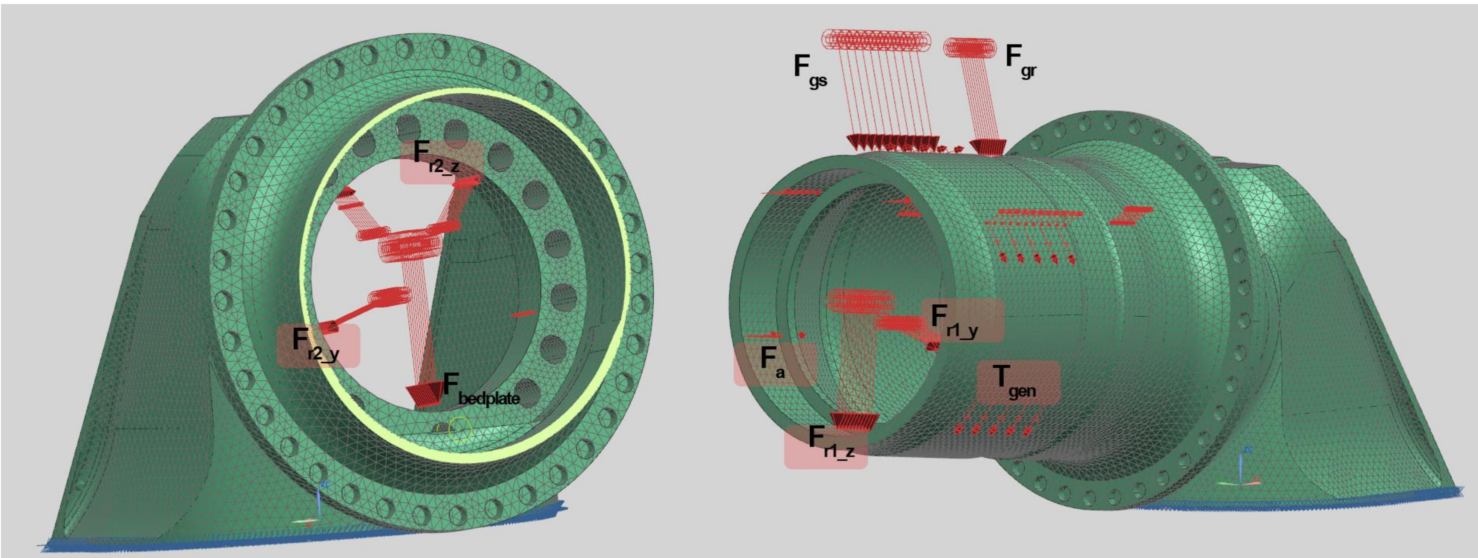


Figure 4.5 - FE analysis

The multi-axial loading is represented by four simulations for each wind condition (13 or EXT) for the design load cases. The selected load directions represent maximum bending moments in directions as shown in figure 3.3 where:

1. V-H+ (R1_z_min, R2_z_min, R1_y, -R2_y)
2. V-H- (R1_z_min, R2_z_min, , -R1_y, R2_y)
3. V+H- (R1_z_max, R2_z_max, -R1_y, R2_y)
4. V+H+ (R1_z_max, R2_z_max, , R1_y, -R2_y)

The structural simulations are identified by the following parameters

- Design load case: [ETM or NTM]
- Wind condition: [EXT or 13]
- Multi-axial loading: [1, 2, 3, 4]

The identifiers for the individual simulations are a combinations of these parameters such as ETMEXT_1[4] and NTM13_4[5].

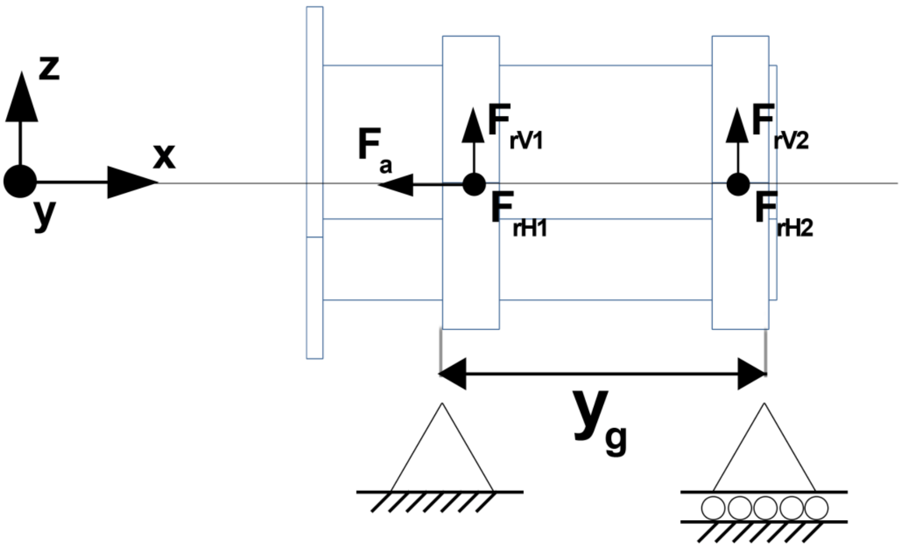
[4] ETMEXT_1 = ETM (DLC 1.3) & Ultimate loads & V-H+
[5] NTM13_4 = NTM(DLC 1.1) & Design wind speed & V+H+

4.4 Summary

The simulated loads are a result of gravitational loads and aerodynamic loads obtained from aero elastic analyses. This chapter has shown how these loads have been implemented as several FEM simulations which covers the worst case load situations for the selected design load cases. These simulations have been used in the development of the bedplate design to verify that the design complies to the stress criteria presented in chapter 2.

5. Bearing selection

5.1 Main shaft bearings options



Figur 5.1 - Bearing configuration

The main shaft is simply supported where the front bearing is fixed and the back bearing is floating. Main shaft bearings are usually a paired solution or a combination of the following bearing types:

- Spherical Roller Bearing (SRB)
- Cylindrical Roller Bearing (CRB)
- Multi-Row Tapered Roller Bearings

Figur 5.2 - From the right - SRB, CRB, Single-row multi-tapered roller bearing, TDI, TDO. [13]

Spherical roller bearings are self-aligning and very robust bearings. Two rows of rollers enables it to carry heavy loads in both radial and axial directions. They are however difficult to produce and therefore expensive. Often used in pairs.

Cylindrical roller bearings can carry very heavy radial loads at moderate speeds because of the large roller surface area. It is usually more sensitive to misalignment, which causes the capacity to drop radically, additionally the capacity is lowered when axially loaded.

Tapered rolling bearings use conical rollers, this enables the bearing to handle very high radial and axial loads. They are however complex to manufacture, which results in high cost. Multi-row tapered roller bearings are prone to handle more pretension than the alternatives which makes them very stiff.

Double-row tapered bearings are commonly used among in large wind turbines. There are two dominant configurations, tapered double inner (TDI) and tapered double outer (TDO). Both carries radial, axial and moment loading. The TDO is considered superior because the load center distance are greater [14]. TDOs are stiffer than the TDI configurations, which causes the TDO to be more prone for angular misalignments of the outer ring relative to the inner ring [15]. This makes a TDI configuration preferable as misalignments are to be heavily expected for a operational wind turbine.

In the previous chapter design load cases shows that the main shaft are exposed to extreme radial and axial loads due to thrust, bending moments from turbulent wind conditions combined with static loads. The priorities behind the selection of a main shaft bearing solution is prioritized by

1. Excellent stiffness
2. High reliability
3. Low cost

The thrust-to-radial loading is crucial for the selection for the fixed bearing. Under normal operational conditions it is reasonable to assume that this ratio is approximate in the 0.6 range. This excludes CRB as a candidate as the fixed bearing.

5.2 Main bearing selection

A quick assessment of bearings is performed, mainly to retrieve realistic dimensions for the bedplate design, a basic lifetime estimation with the ultimate load is considered satisfactory for this preliminary evaluation. The final solution is traditionally developed in cooperation with the bearing manufacturer.

The IEC standard requires the minimum acceptable calculated lifetime for main shaft bearings to be 175000 hours, 20 years. Additionally the safety factor for static load / static equivalent load must be at last 2.0 [16]. The bearings are selected with the following estimates:

- It is assumed that 20yrs equals: 2*10^7 cycles.
- Front bearing loads: Radial force: 12000kN - Axial Force 2200kN
- Back bearing load: Radial force: 8000kN
- Rotational speed: 13.54RPM

The selected bearings must meet the following requirements:

1. $S = \frac{C}{P} = 2$
2. Lifetime is minimum 20Mrev

5.2.1 Front Fixed bearing

TDO/TDI	SRB
+ Excellent capacity + Very good ability for preloading, stiffness + Very reliable + Allows little deflection and displacement - Expensive	+ Excellent lifetime + Good capacity + Reliable - Expensive (less than TDO)

A double-row tapered roller bearing (TDO) is selected as the fixed bearing.

$$P = F_r + F_a \,.$$

$$P = 16172kN$$

$$C \geq 32000kN \,.$$

"BT2B 332497/HA4" from the SKF online product catalog[6] satisfy these requirements. A basic life estimation obtained from the SKF webpage is included in appendix E.

BT2B 332497/HA4 specifications:

- $C = 34700$kN
- $C_0 = 108000$kN
- $P_u = 5000$kN
- Mass=11600kg
- $L_{10} = 23$Mrev

The two requirements are satisfied.

No other manufacturer[7] offers mass produced TDI/TDO bearings in relevant dimensions. A TDO configuration is preferable. The selected bearings is considered to be over dimensioned and very heavy.

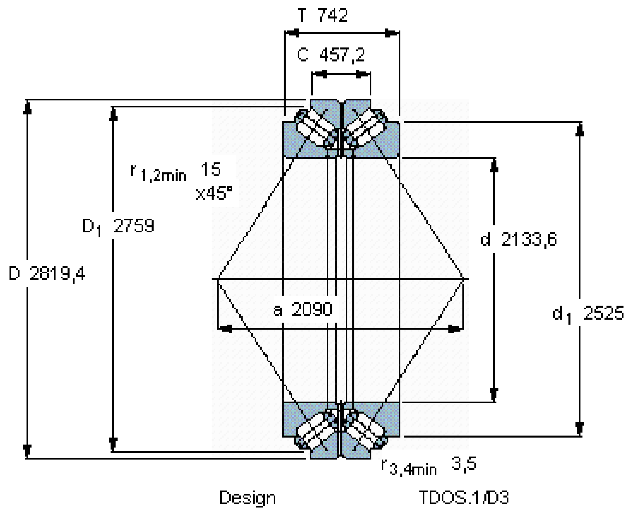


Figure 5.3 - Dimensions for BT2B 332497/HA4

5.2.2 Back floating bearing

It is desirable to have a smaller floating than fixed bearing in order to reduce nacelle weight. A spherical roller bearing is selected from the SKF online product catalog: 248/1800 CAK30FA/W20.

$$P_0 = F_r = 8000kN$$

$$C_0 \geq 2 * P_0$$

[6] www.skf.com
[7] According to www.gigantbearing.com (FAG, NTN, TIMKEN, NSK, KOYO)

The 248/1800 CAK30FA/W20 has desirable dimensions and weights less than the available cylindrical roller bearings. A spherical roller bearings cope better with misalignments than cylindrical bearings.

248/1800 CAK30FA/W20 specifications:

- $C = 20000\text{kN}$
- $C_0 = 63000\text{kN}$
- $P_u = 8000\text{kN}$
- Mass=2850kg
- $L_{10} = 21\text{Mrev}$

The lifetime is considered satisfactory.

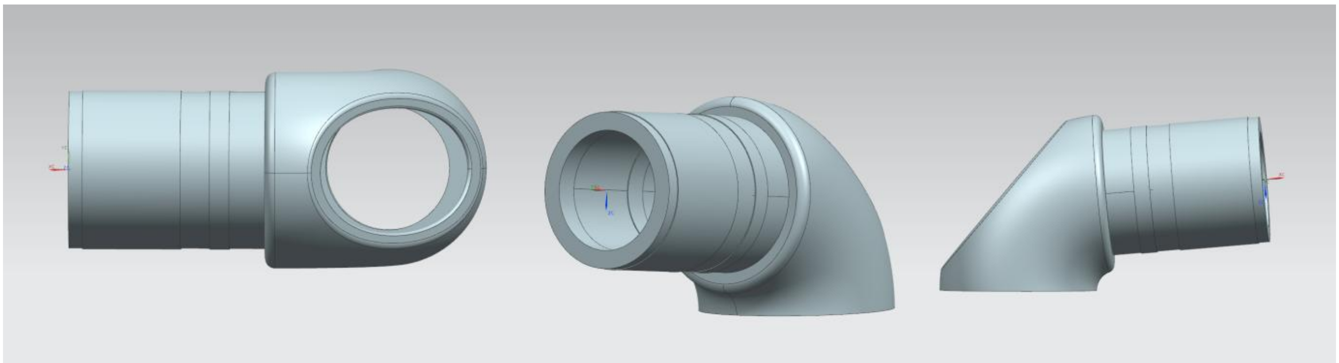
CAD models of the selected bearings was obtained from the SKF web page.

5.3 Summary

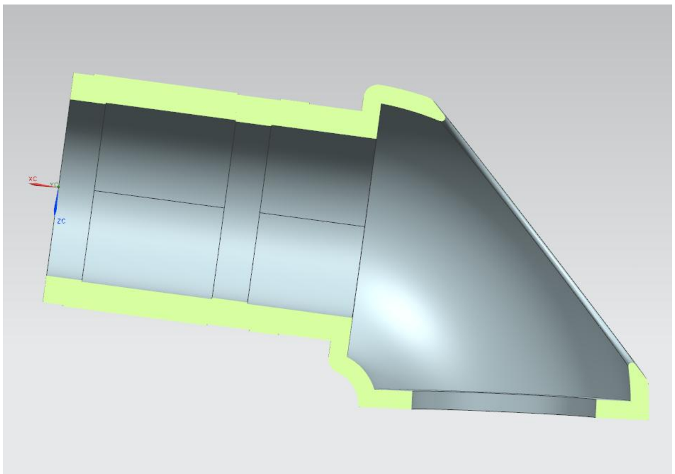
A solution consisting of a spherical and double tapered roller bearing is proposed based on a basic lifetime estimation using the highest encountered design loads obtained from DLC 1.3. A more robust analysis is required, this is traditionally done in cooperation with the bearing manufacturer.

6. Bedplate designs

6.1 Naïve design



Volume: 23,49m^3
Estimated mass[8]: 183,02t
Bearing distance: 3284mm



This is the naïve implementation of the bedplate obtained from NOWITECH at the start of this project. The geometry is heavily over-dimensioned and suffers from several weaknesses. The most apparent problem is the current transition between the nose and the tower, where compression caused by loading causes extreme high stress. This can be dramatically reduced by applying a blend or chamfer.

Equation 0.8 shows that the loads for the bearings are proportional with the distance between the bearings. Not only will an increased distance reduce the loads caused by the bending moments, but the structure will provide a more support as the load distribution is moved closer to the tower. This should be evaluated against the additional weight caused by a longer main shaft.

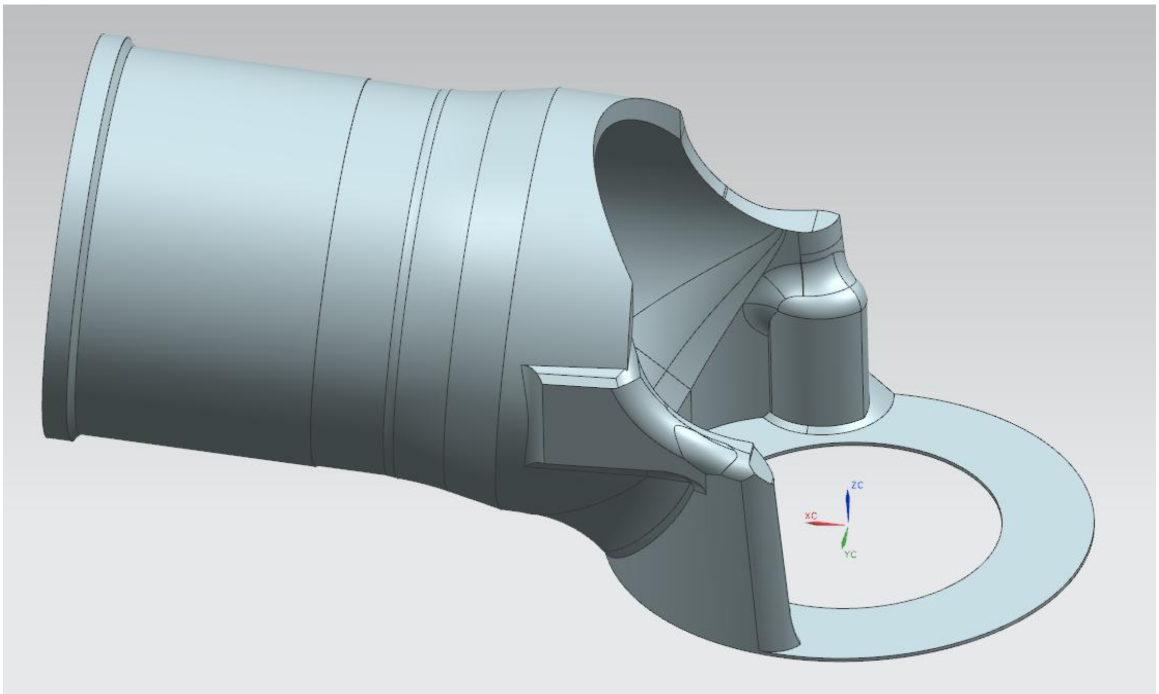
The circular hole at the back does not provide enough stiffness (given a realistic wall thickness) to cope with the horizontal bending moments and torque.

6.2 Design iterations

During the development process of the bedplate, several concepts was explored. The load cases was developed in parallel which caused the benchmarks to change radically throughout the development phase. In this section two design iterations are presented in addition a few points to summarize the design and lessons learned.

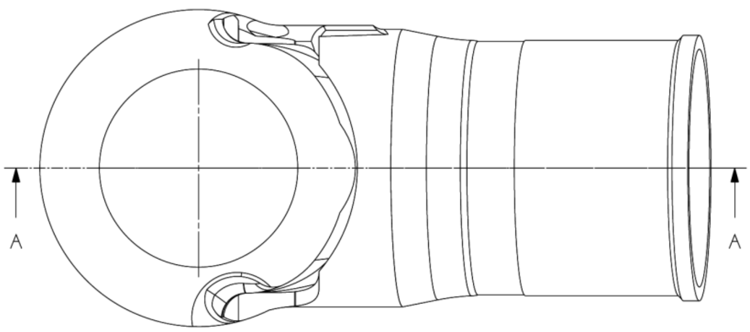
[8] Obtained with the NX's "Measure bodies" function.

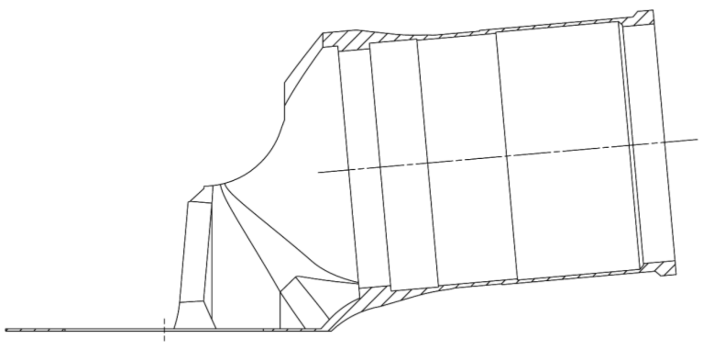
Concept #1 - 49,6tons



- Immature geometry
- Asymmetrical design
- Torque is a problem
- Awful load transfer to the tower.

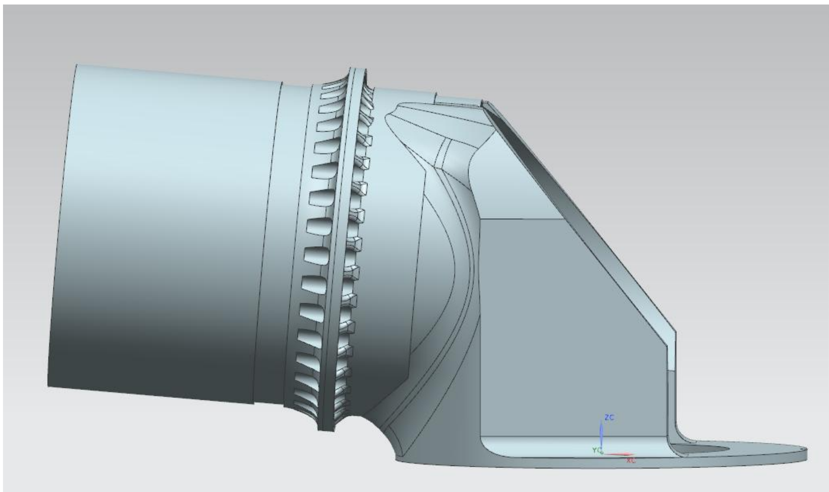
It became clear that the bedplate had to be divided into two components. The design was not able to support the rotor, generator and hub sufficiently (~250MPa). The additional aerodynamic loads was beyond possible with this concept.

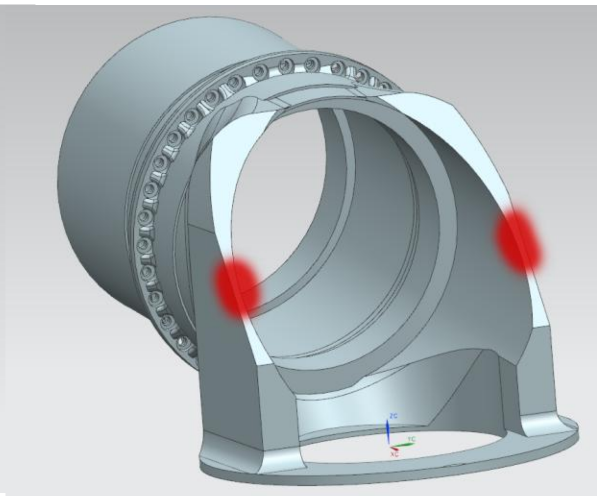




SECTION A-A

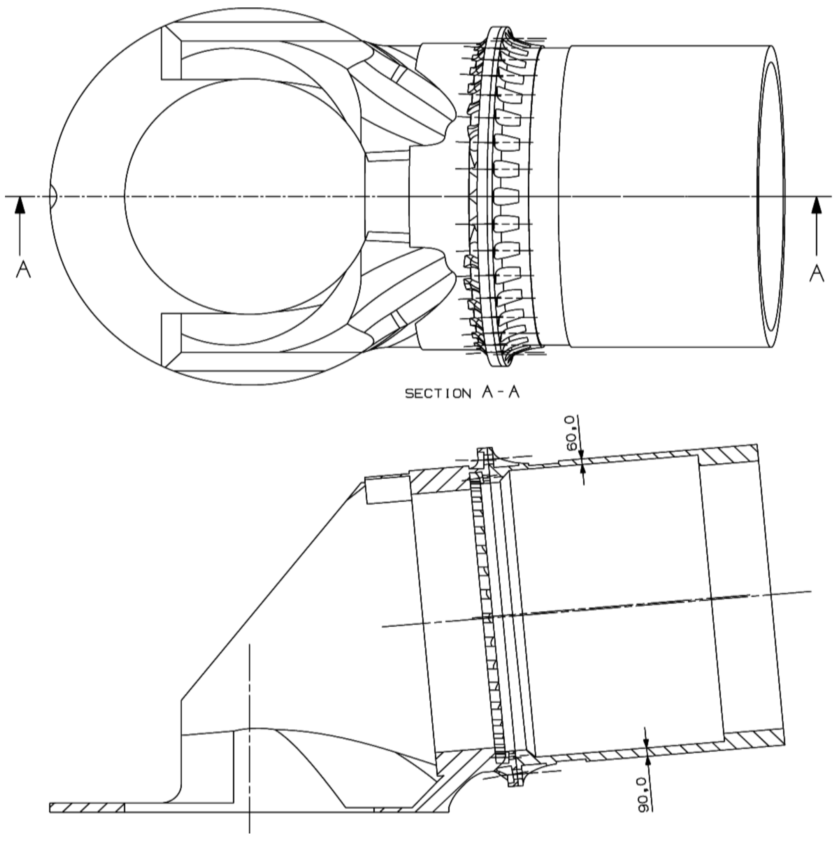
Concept #4 -70tons





- Naïve flange geometry.
- Stress concentrations on the "legs".

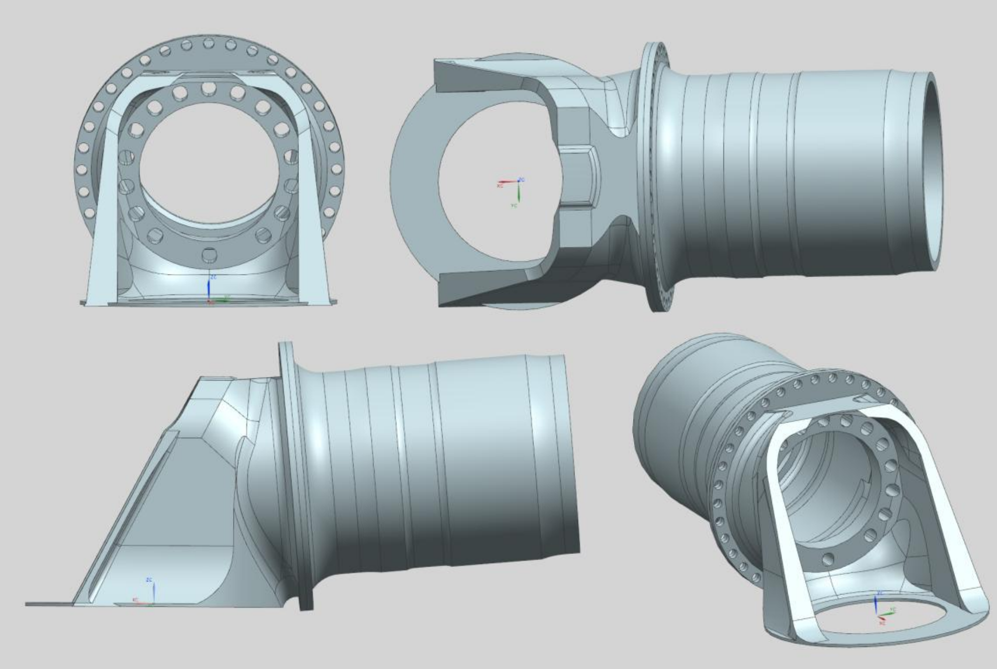
The nose was way too short in this design. The attempt to add stiffness in the flange failed violently, the "fancy" geometry resulted in very high stress concentrations. The support legs were weakest where they should have been at their stiffest (marked in red). This design gave good insights in dimensions for the nose profile. A slight offset for the hole resulted in a 30mm bottom surface which gave very good results compared to a uniform wall thickness.

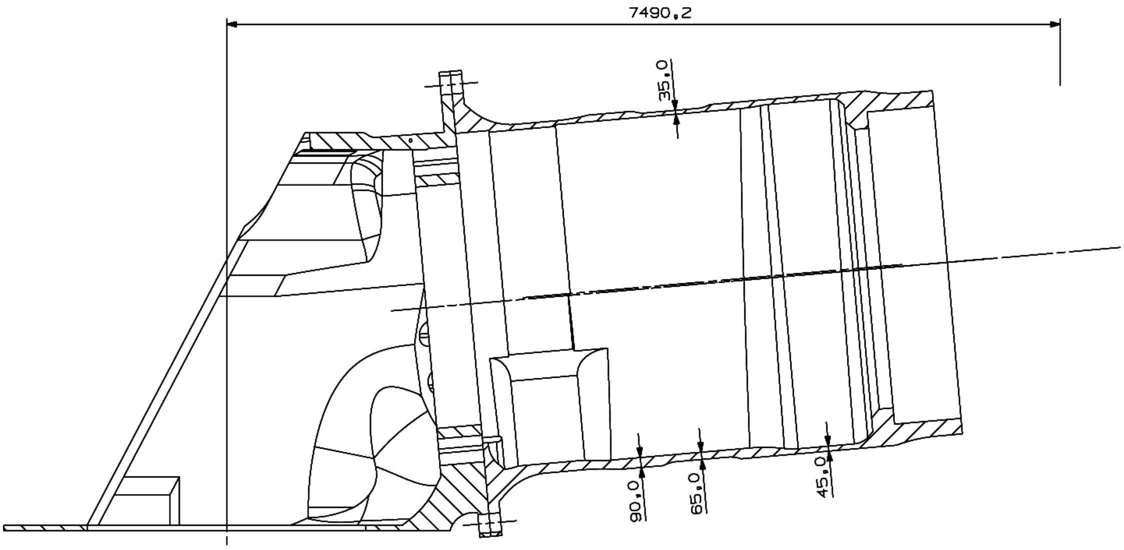


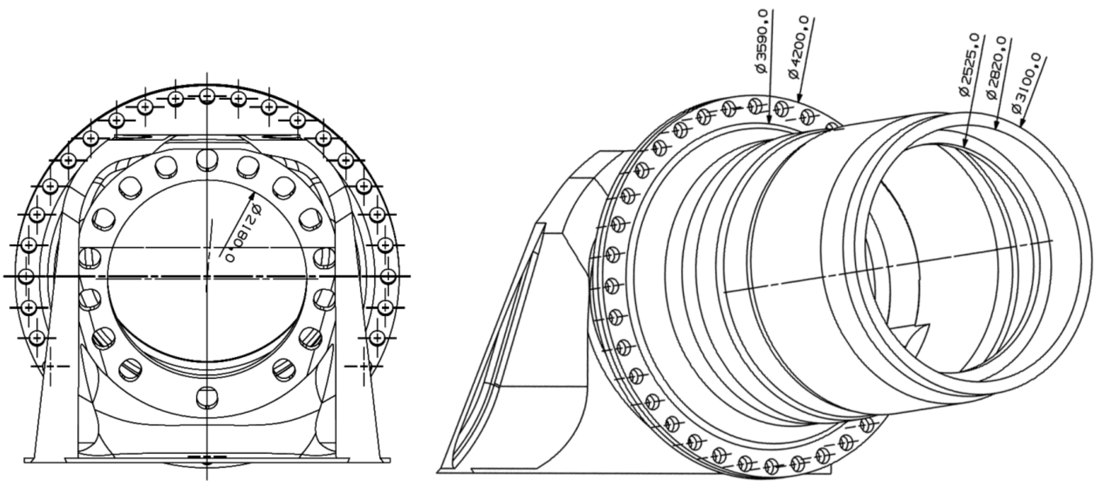
6.3 Optimized design

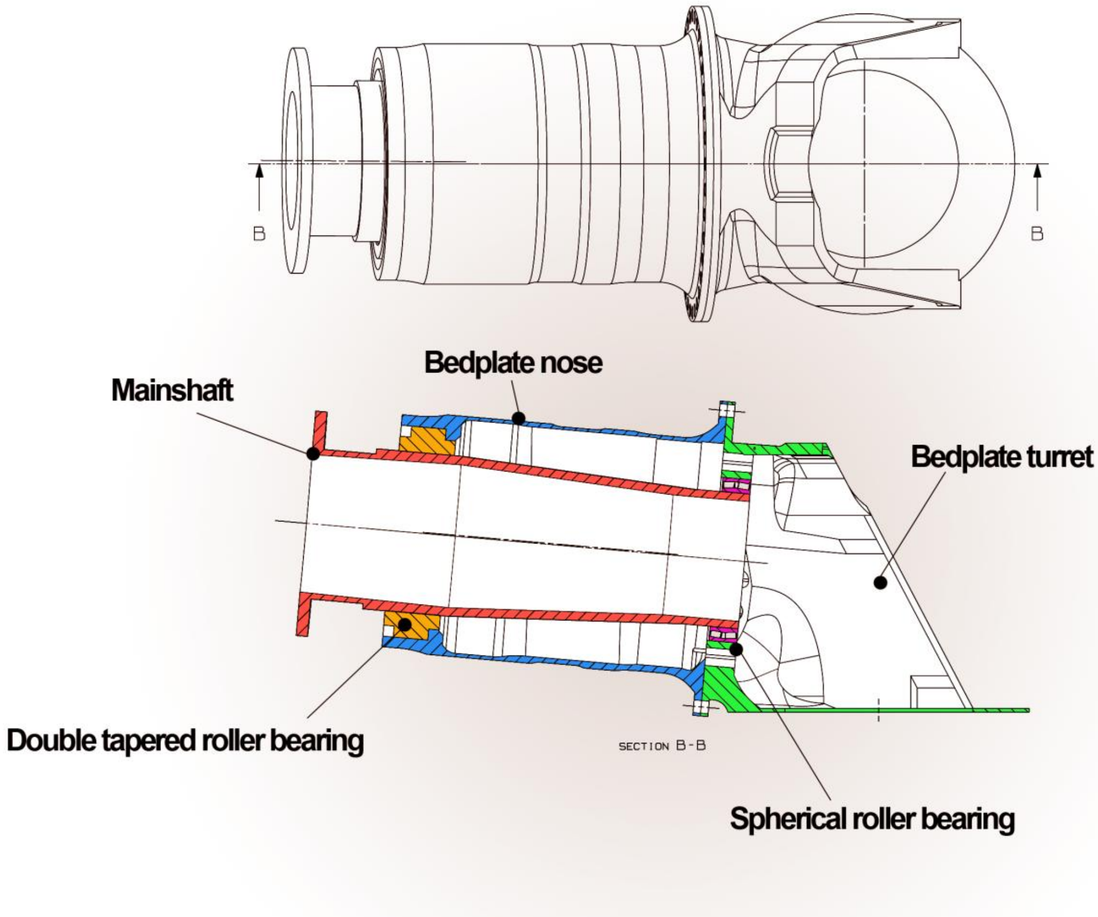
This design is divided into two components, turret and nose. Each component weights under 50 tons, which makes it applicable for casting. This design requires additional stiffness for the transition to the yaw bearing.

Volume: 10,53m^3
Tower: 6,21m^3
Nose: 4,32m^3
Estimated mass: 82,4t
Tower: 48,60t
Nose: 33,84t
Bearing distance: 4700mm



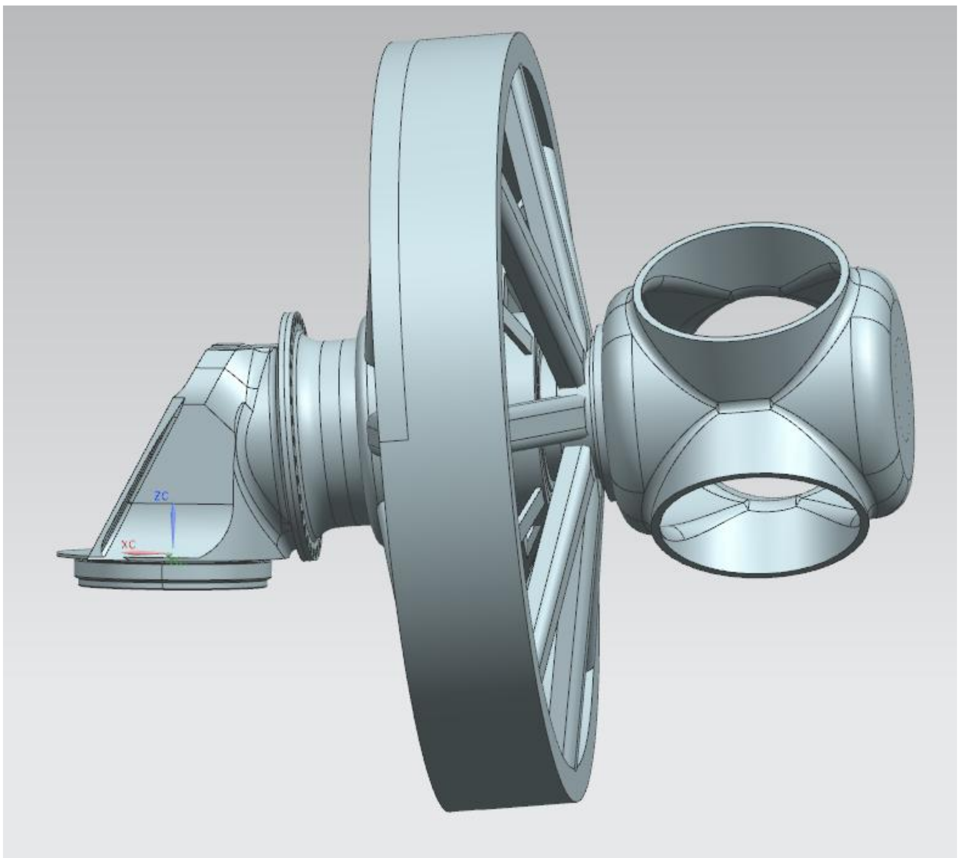






Above - Complete suggested solution

Below - Complete suggested solution with generator and hub



7. Ultimate Strength analysis

The simulated design loads and design load effects are presented. The presented design load effects is represented as von Mises stress. The stress can be obtained from each element, each node in the element and as the average of all the nodes. The nodal results have the highest precision. Results from all these methods are presented to show the consistency of the simulation, as some nodes may be heavily influenced by unfavorable local effects. Additionally, it provides clarity of how the stress affects the structure.

7.1 FEM-mesh

The FEM mesh for the optimized design is a computed with the following features:

- Removed the holes the flange (Idealized Mesh)
- CTETRA(10) elements
- Element size of 100mm
- Mesh Mating between components
- Fixed constraint at the base

The fixed constraint on the base assumes infinite stiffness to the yaw system/tower. The model confidence level for 100mm and CTETRA(10) elements have been in the 97% range for all simulations, and has been considered satisfactory. Mesh mating condition assumes that load transfer in the flange is perfect.

7.1 No wind

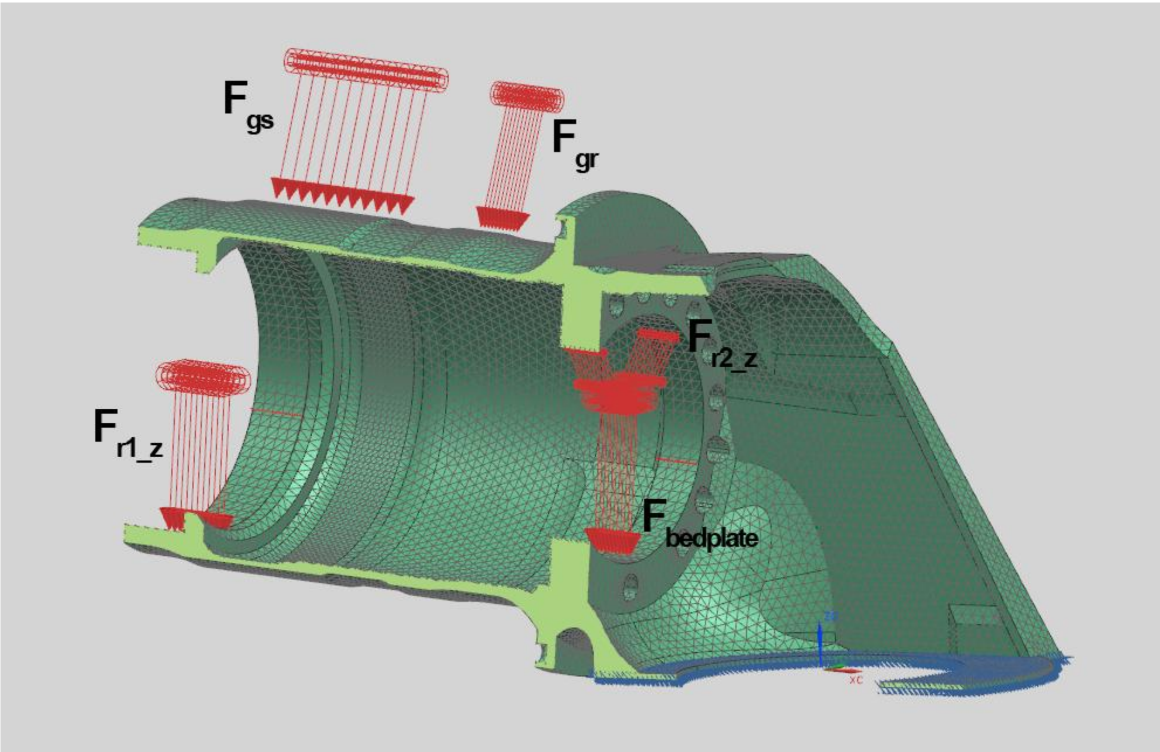
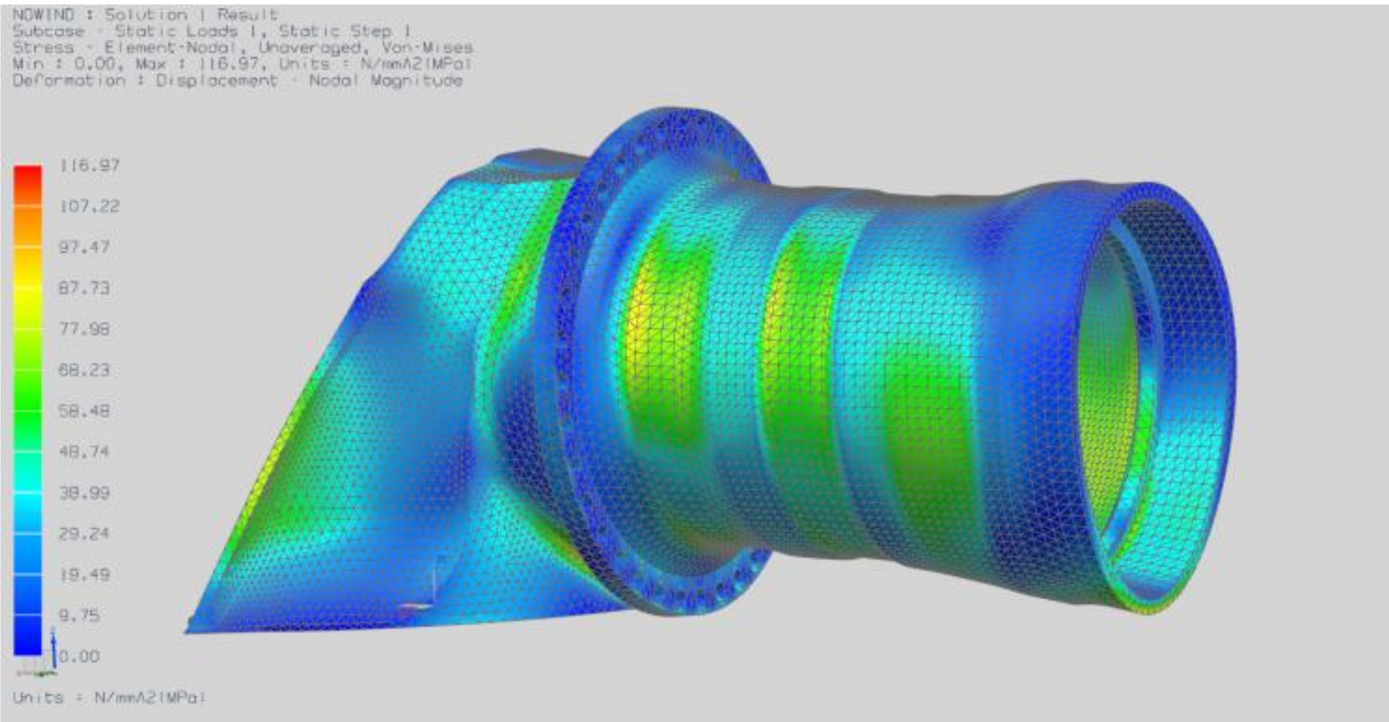
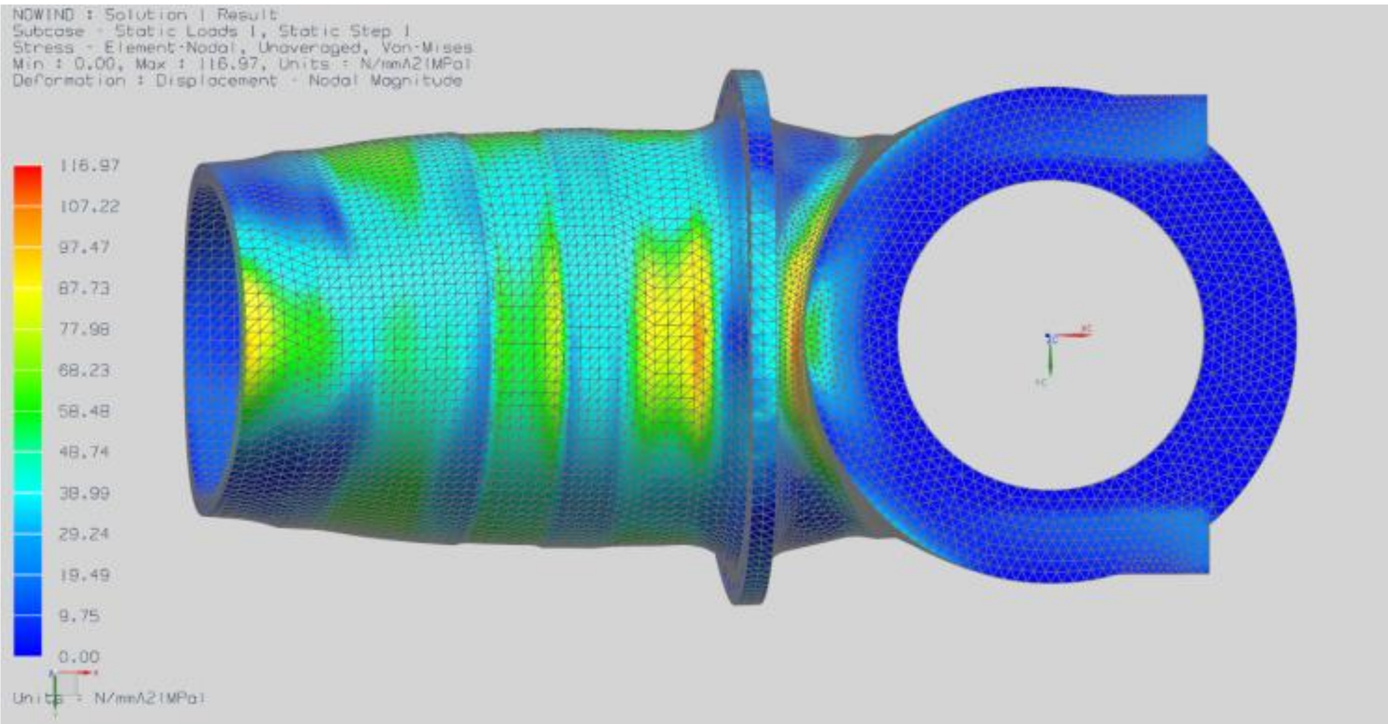


Figure 7.1 - Simulated loads for the no-wind condition





No-wind - Results	
Model Confidence Level	97,50 %
Von Mises Elemental [MPa]	93,52
Von Mises Elemental Nodal (avg) [MPa]	107,26
Von Mises Elemental Nodal [MPa]	116,97

F_gs [kN]	2646
F_gr [kN]	397
F_bedplate [kN]	1324
F_r1_z [kN]	1800
F_r2_z [kN]	5013

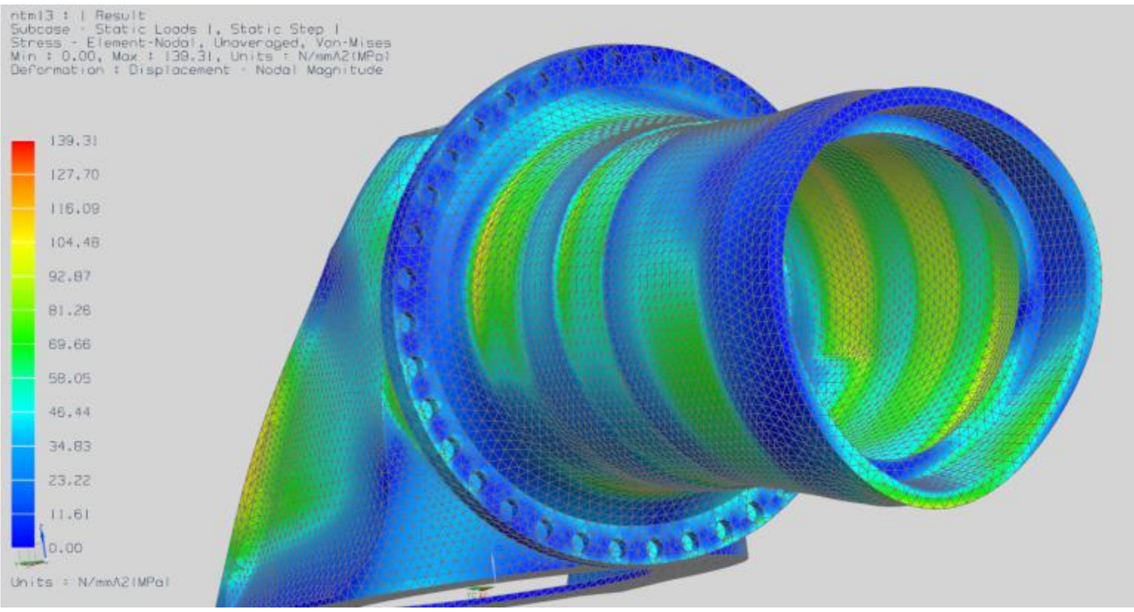
7.2 DLC 1.1

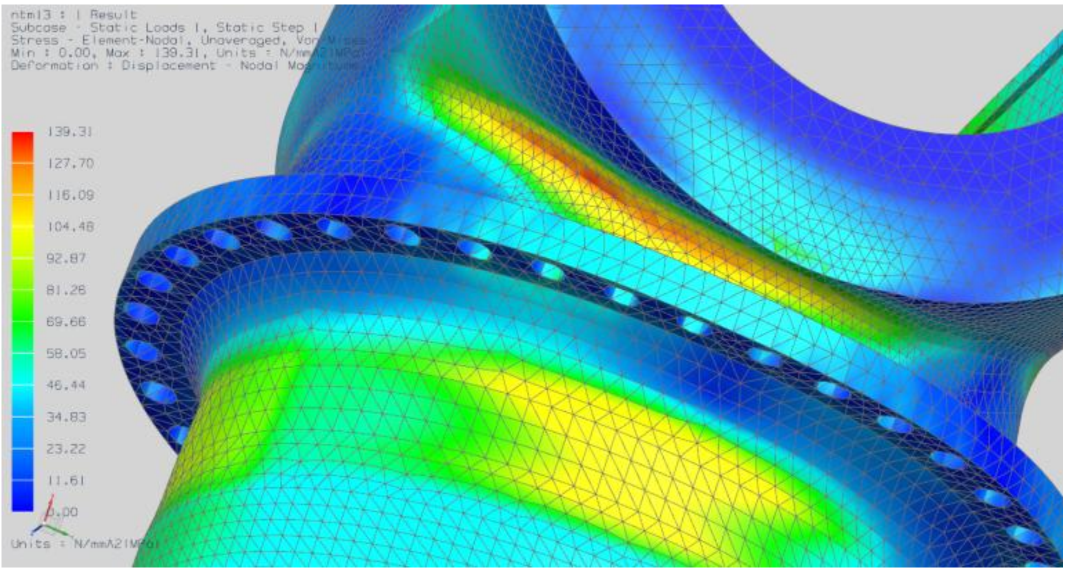
7.2.1 NTM13

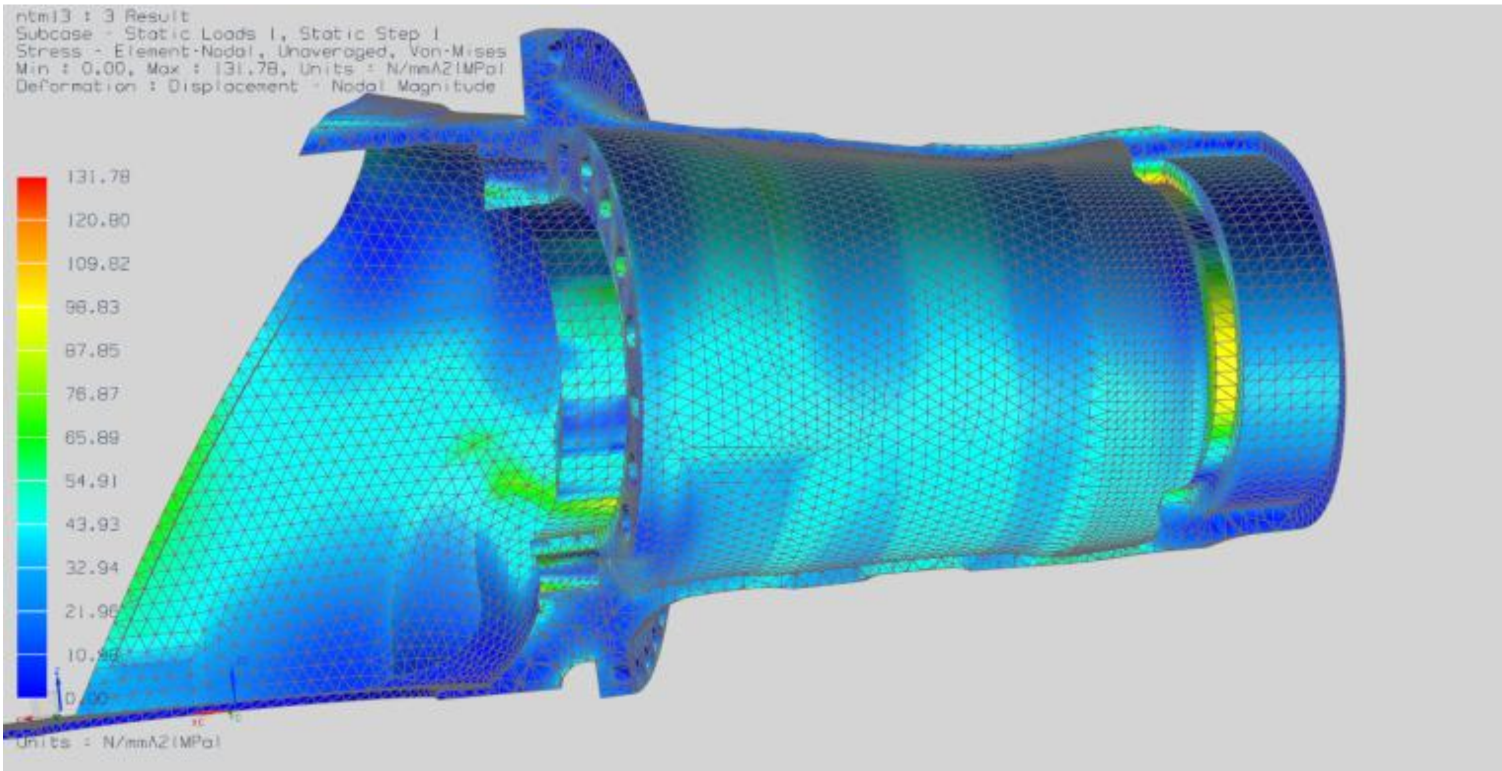
DLC 1.1 - Design loads for the design speed (13m/s)

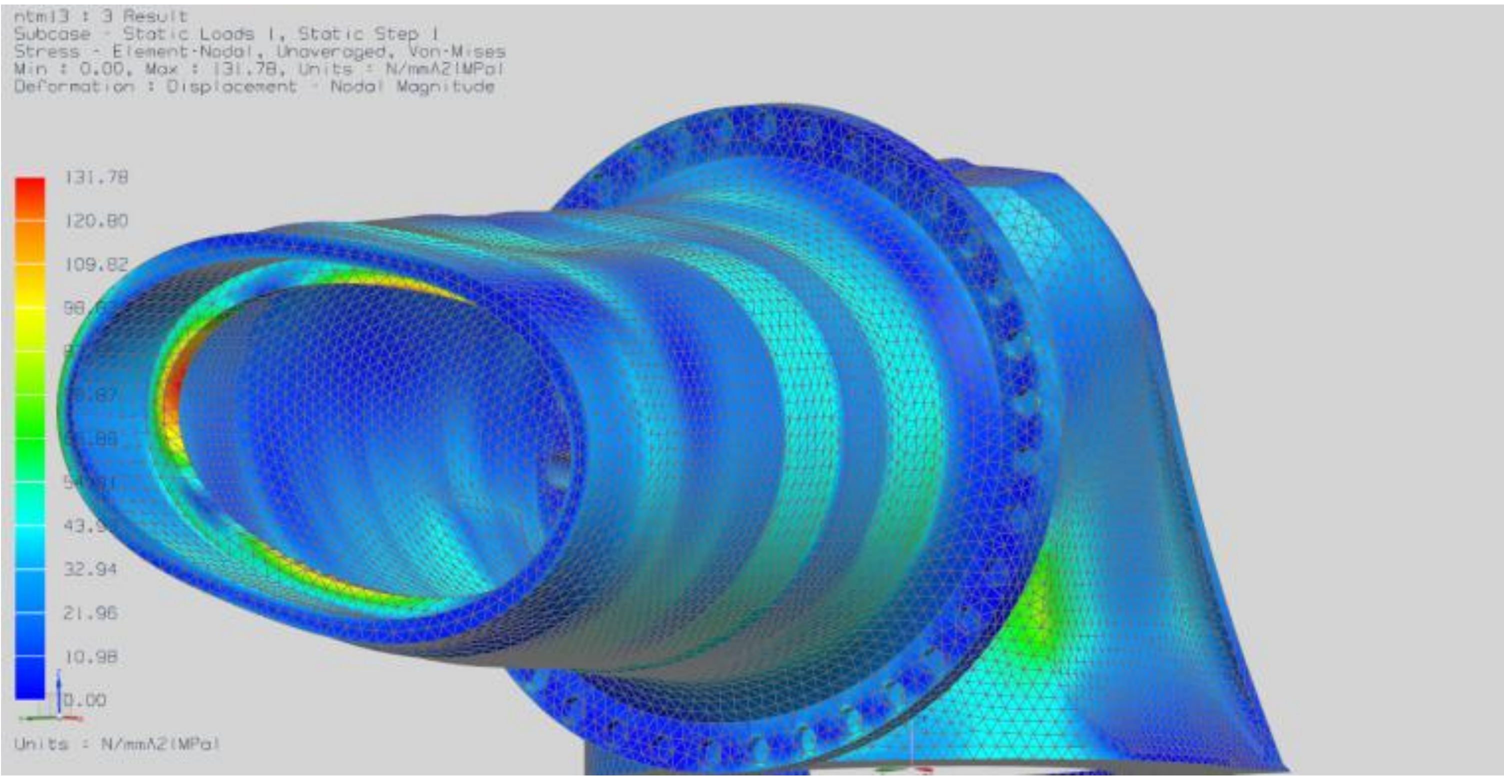
Component	1: V-H+	2: V-H-	3: V+H-	4: V+H+
F_a [kN]	1469			
T_gen [kNm]	11555			
F_r1_z [kN]	-6162	-6162	-603	-603
F_r2_z [kN]	-3848	-3848	1369	1369
F_r1_y [kN]	2707	-2707	-2707	2707
F_r2_y [kN]	2558	-2558	-2558	2558

DLC1.1 - NTM13	**1: V-H+**	**2: V-H-**	**3: V+H-**	**4: V+H+**
Model Confidence Level	97.508%		97,31 %	
Von Mises Elemental [MPa]	109,43	105,74	114,01	112,06
Von Mises Elemental Nodal (avg) [MPa]	134,59	130,92	130,74	127,5
Von Mises Elemental Nodal [MPa]	**139,43**	134,84	131,78	129,28






ntm13 : 3 Result
Subcase - Static Loads 1, Static Step 1
Stress - Element-Nodal, Unaveraged, Von-Mises
Min : 0.00, Max : 131.78, Units : N/mm^2(MPa)
Deformation : Displacement - Nodal Magnitude
131.78
120.80
109.82
98.83
87.85
76.87
65.89
54.91
43.93
32.94
21.96
10.98
0.00
Units : N/mm^2(MPa)

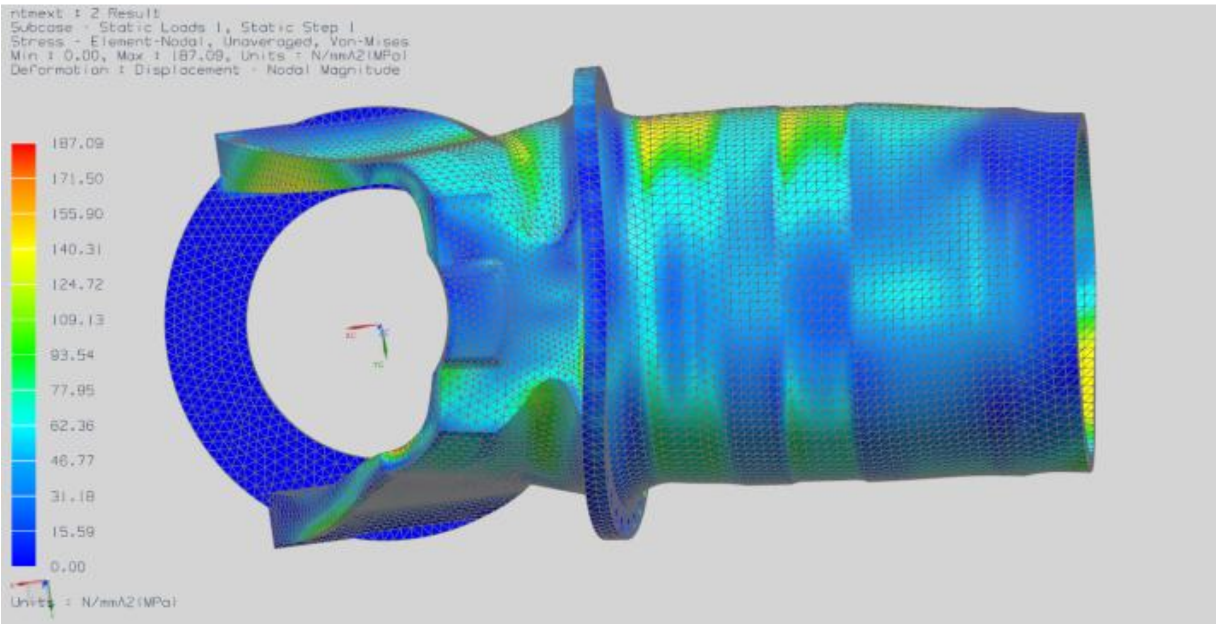

ntm13 : 3 Result
Subcase - Static Loads 1, Static Step 1
Stress - Element-Nodal, Unaveraged, Von-Mises
Min : 0.00, Max : 131.78, Units : N/mm^2(MPa)
Deformation : Displacement - Nodal Magnitude
131.78
120.80
109.82
32.94
21.96
10.98
0.00
Units : N/mm^2(MPa)

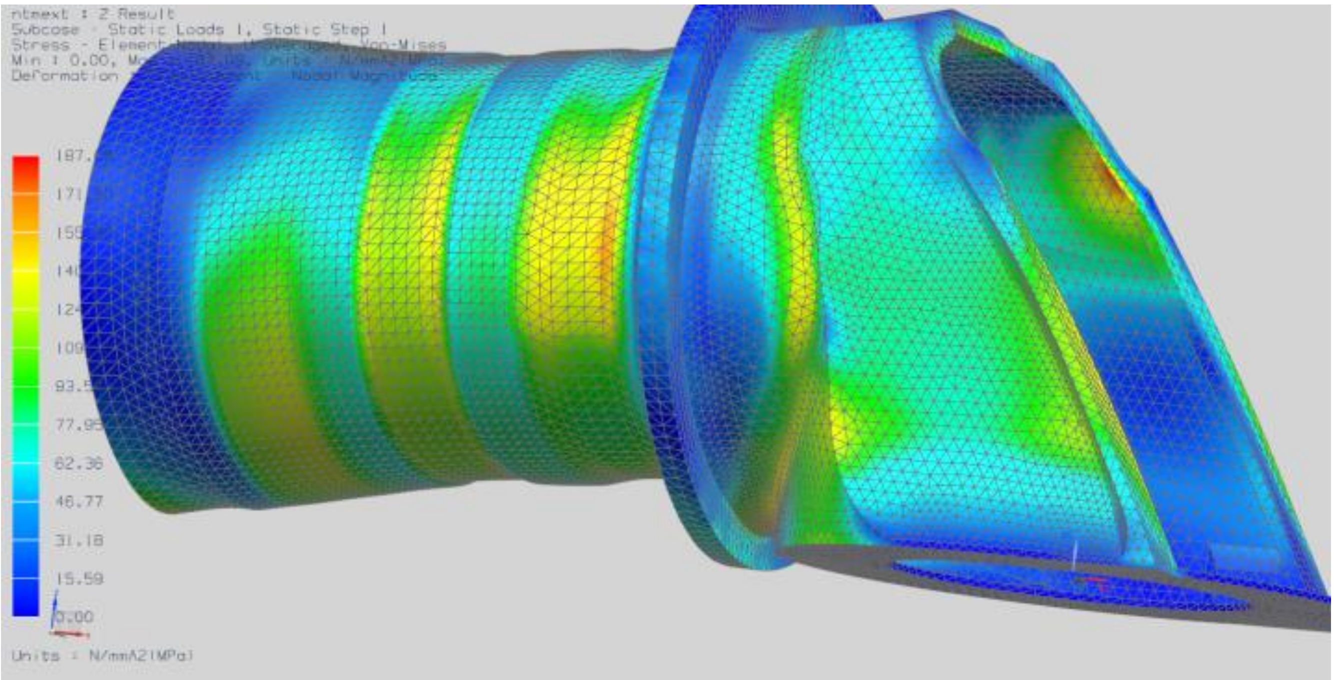
7.2.2 NTMEXT

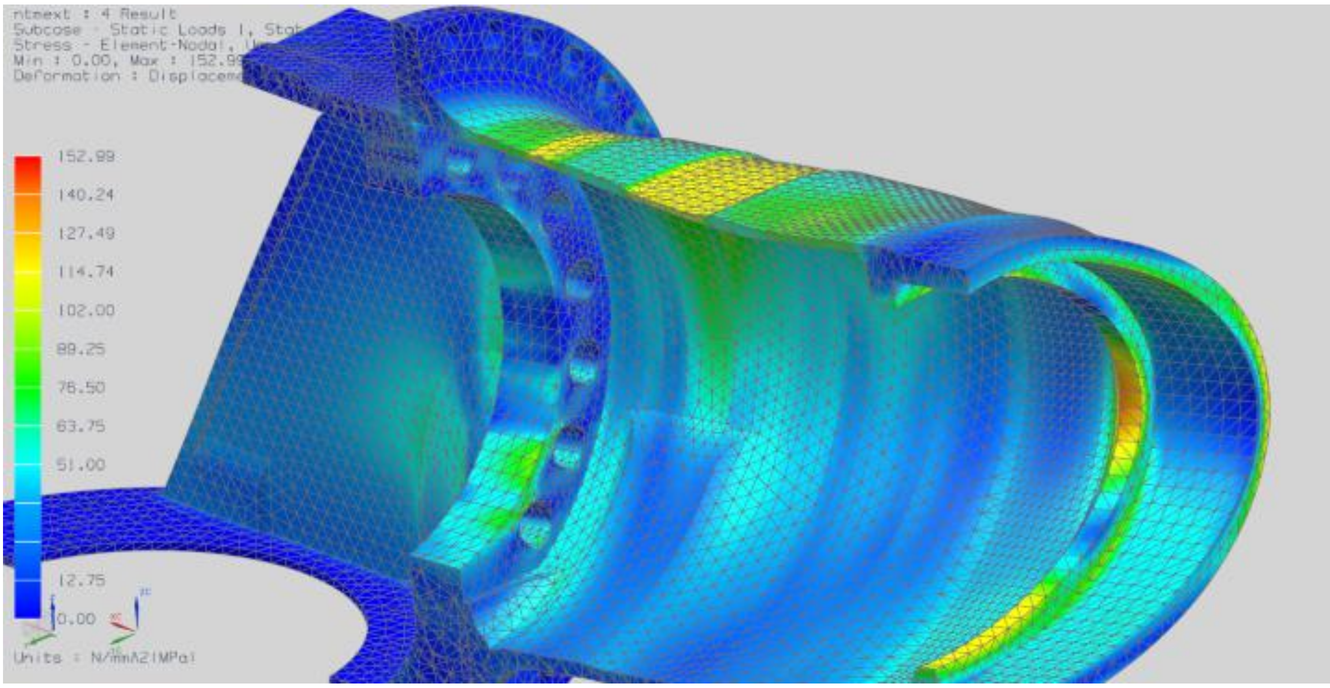
DLC 1.1 - Maximum Design Loads

F_a [kN]	1694			
T_gen [kNm]	14567			
F_r1_z [kN]	-8710	-8710	3567	3567
F_r2_z [kN]	-6191	-6191	5302	5302
F_r1_y [kN]	6242	-6242	-6242	6242
F_r2_y [kN]	5837	-5837	-5837	5837

DLC1.1 - NTMEXT	1: V-H+	2: V-H-	3: V+H-	4: V+H+
Model Confidence Level	97.614%		97.582%	
Von Mises Elemental [MPa]	151,2	148,32	124,21	131,94
Von Mises Elemental (avg) [MPa]	186,73	182,22	146,45	148,59
Von Mises Elemental Nodal [MPa]	**192,76**	187,09	180,83	152,99




ntmext : 2 Result
Subcase - Static Loads 1, Static Step 1
187.
171
155
140
12
109
93.5
77.95
62.36
46.77
31.18
15.59
0.00
Units : N/mmA2(MPa)


ntmext : 4 Result
Subcase - Static Loads 1, Stat
Stress - Element-Nodal,
Min : 0.00, Max : 152.9
Deformation : Displacem
152.99
140.24
127.49
114.74
102.00
89.25
76.50
63.75
51.00
12.75
0.00
Units : N/mmA2(MPa)

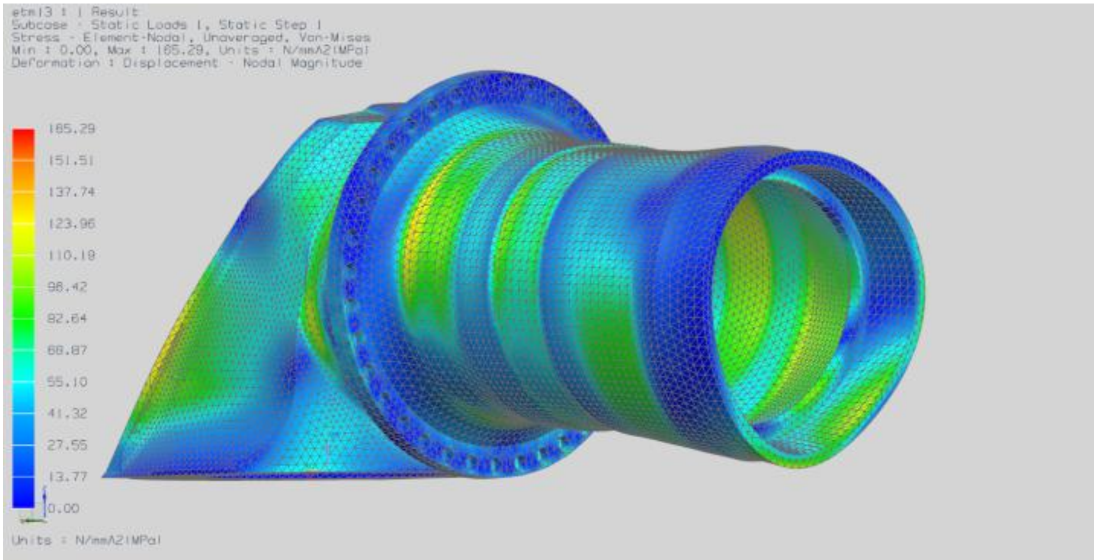
7.3 DLC 1.3

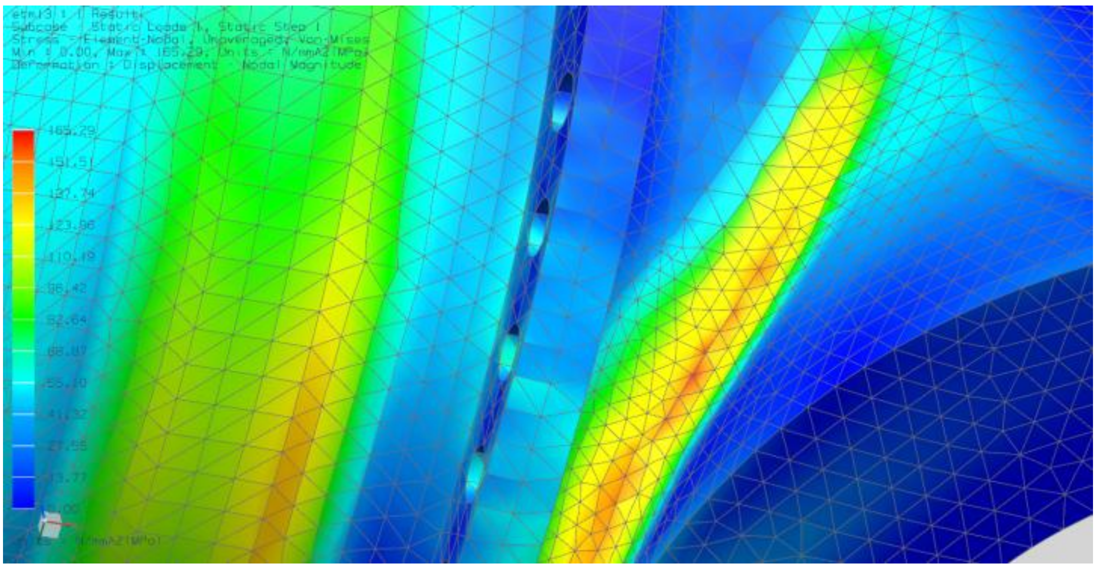
7.3.1 ETM13

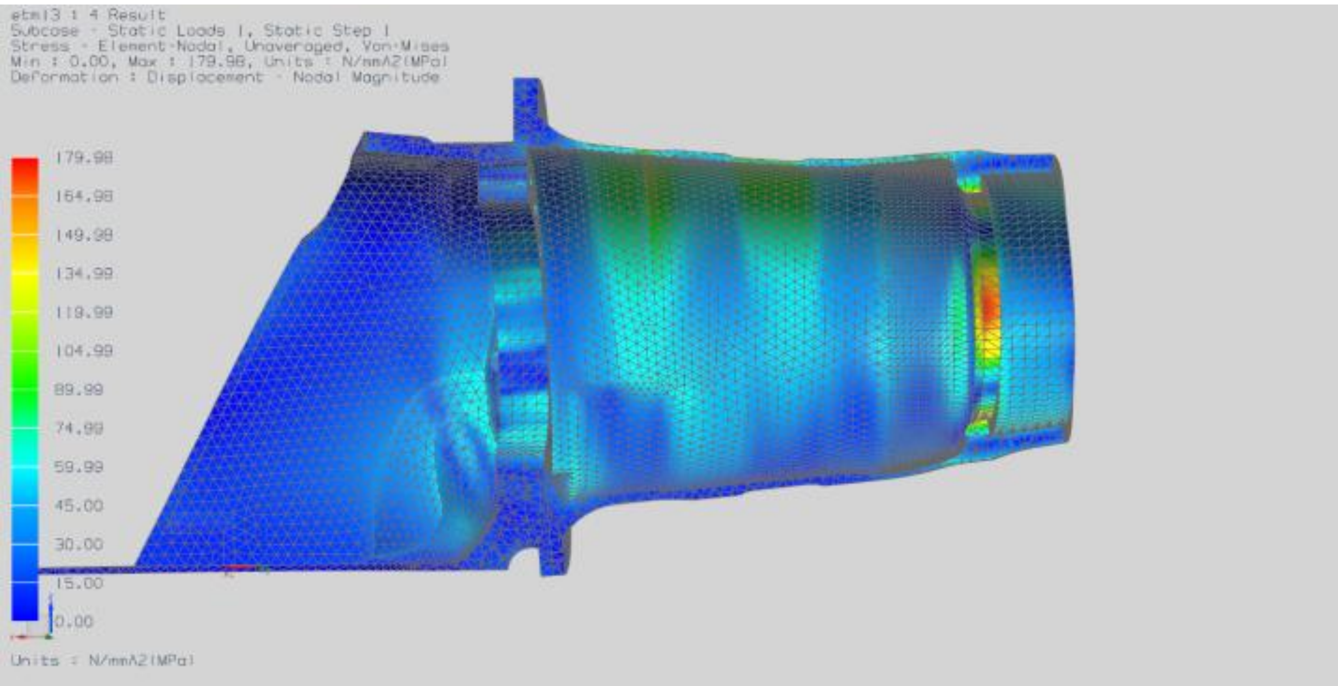
DLC 1.3 - Design loads for the design speed (13m/s)

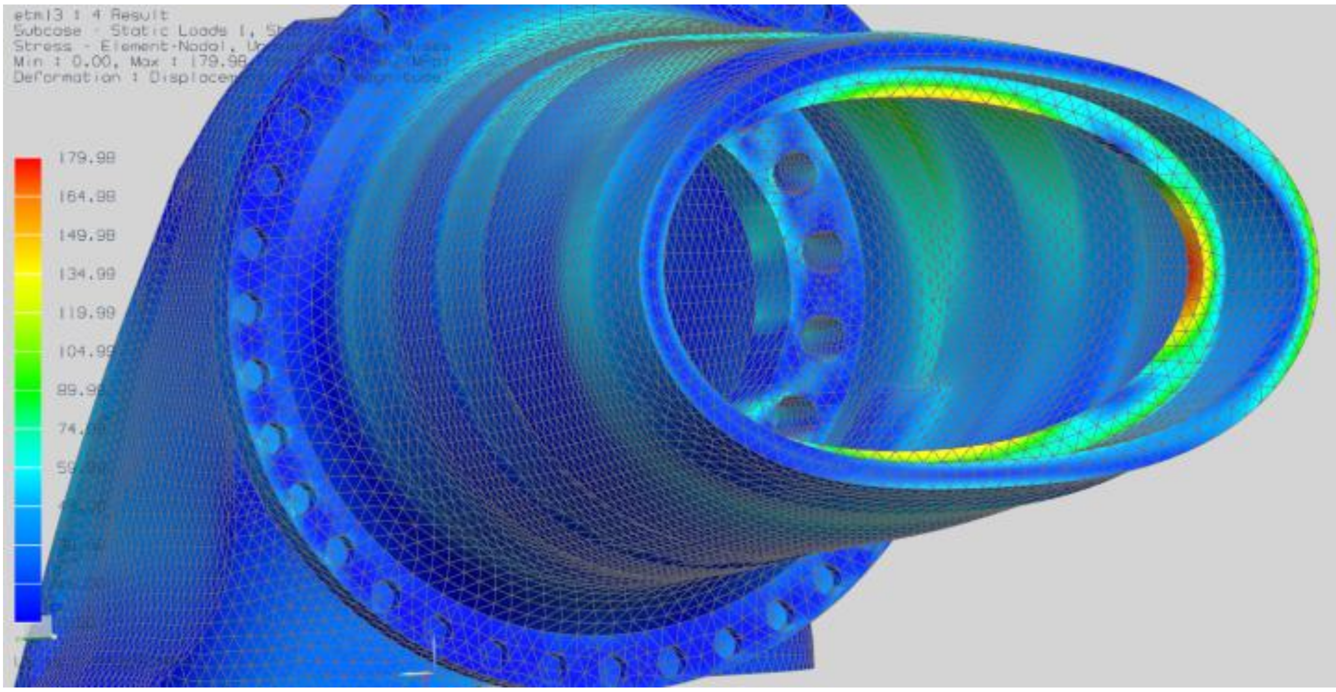
F_a [kN]	1571			
T_gen [kNm]	12324			
F_r1_z [kN]	-7419	-7419	715	715
F_r2_z [kN]	-4981	-4981	2618	2618
F_r1_y [kN]	4722	-4722	-4722	4722
F_r2_y [kN]	4544	-4544	-4544	4544

DLC1.3 - ETM13	1: V-H+	2: V-H-	3: V+H-	4: V+H+
Model Confidence Level	97.555%		97.439%	
Von Mises Elemental [MPa]	128,95	126,51	154,9	152,56
Von Mises Elemental (avg) [MPa]	160,02	156,79	179,88	176,49
Von Mises Elemental Nodal [MPa]	165,29	161,22	**182,02**	179,98






etml3 : 4 Result
Subcase - Static Loads 1, Static Step 1
Stress - Element-Nodal, Unaveraged, Von-Mises
Min : 0.00, Max : 179.98, Units : N/mm^2(MPa)
Deformation : Displacement - Nodal Magnitude
179.98
164.98
149.98
134.99
119.99
104.99
89.99
74.99
59.99
45.00
30.00
15.00
0.00
Units : N/mm^2(MPa)

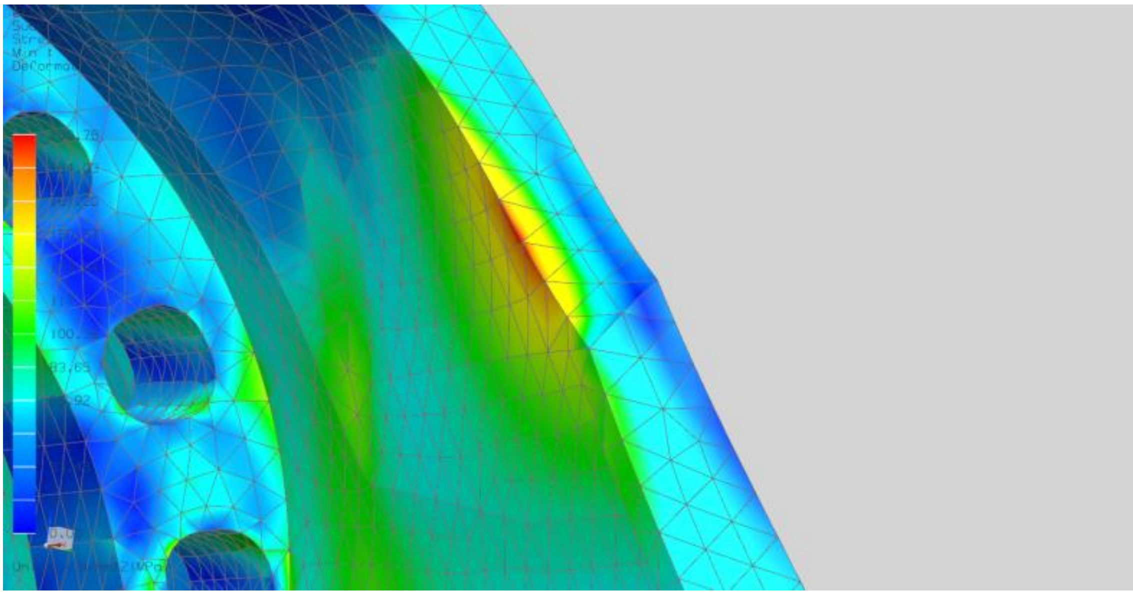

179.98
164.98
149.98
134.99
119.99

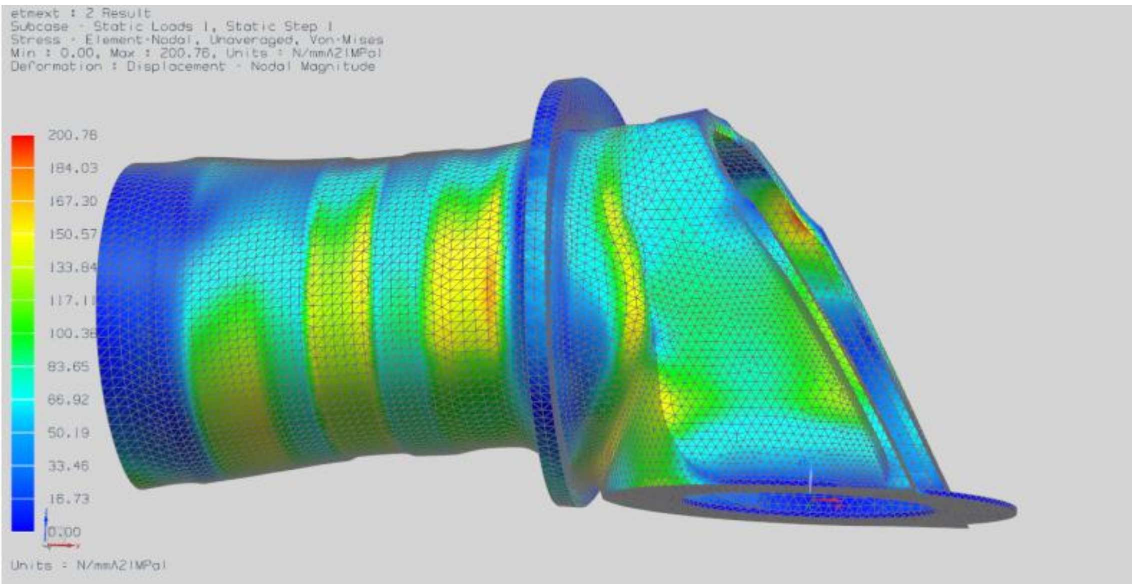
7.3.2 ETMEXT

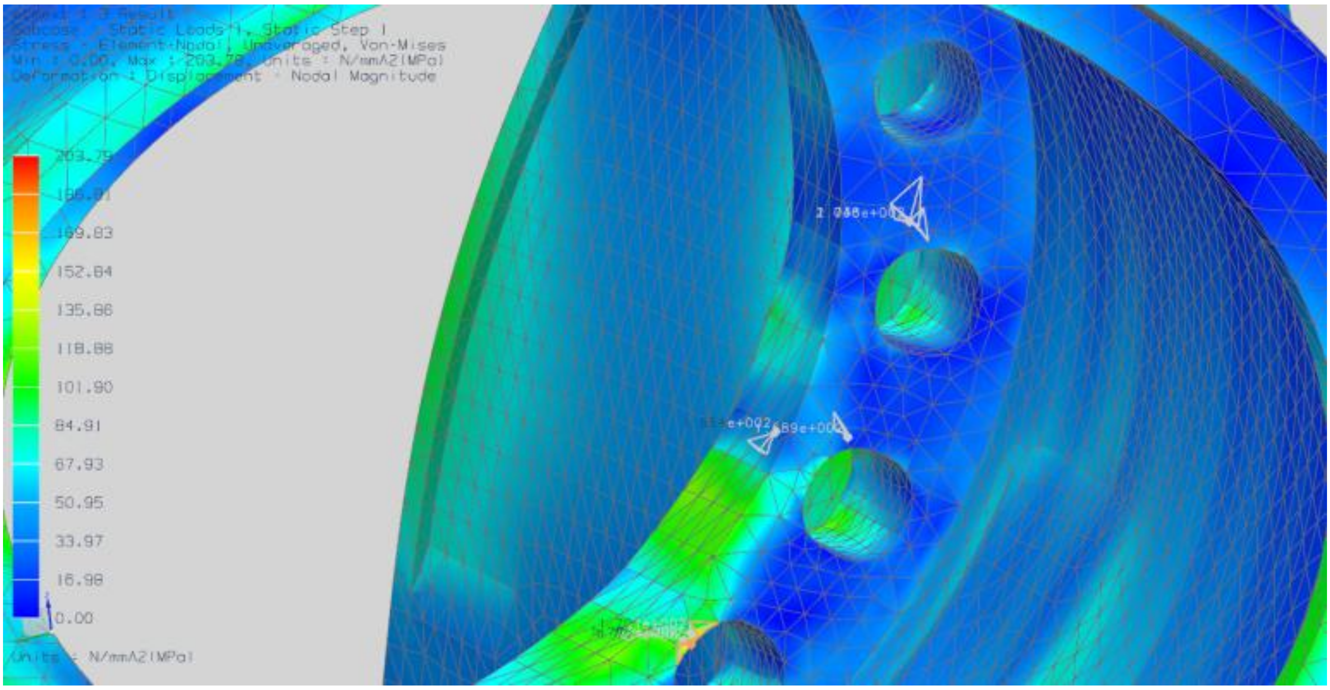
DLC 1.3 - Maximum Design Loads

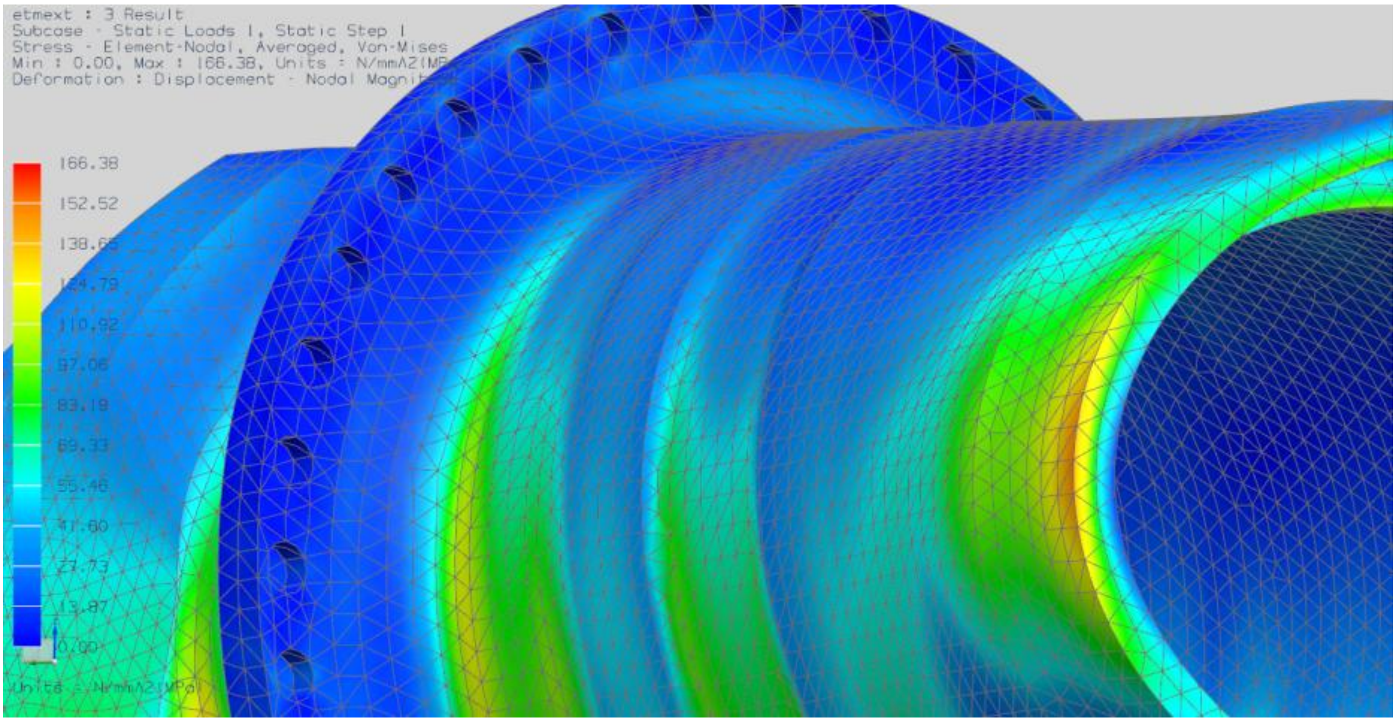
F_a [kN]	1958			
T_gen [kNm]	15782			
F_r1_z [kN]	-9303	-9303	4449	4449
F_r2_z [kN]	-6712	-6712	6531	6531
F_r1_y [kN]	7277	-7277	-7277	7277
F_r2_y [kN]	6818	-6818	-6818	6818

DLC1.3 - ETMEXT	1: V-H+	2: V-H-	3: V+H-	4: V+H+
Model Confidence Level	97.416%		97.613%	
Von Mises Elemental [MPa]	164,42	157,58	140,26	145,53
Von Mises Elemental (avg) [MPa]	201,71	195,77	166,41	167,74
Von Mises Elemental Nodal [MPa]	**208,1**	200,78	**203,83**	172,41






Stress - Element-Nodal, Unaveraged, Von-Mises
Min : 0.00, Max : 203.78, Units : N/mmΛ2(MPa)
Deformation : Displacement - Nodal Magnitude
169.83
152.84
135.86
118.88
101.90
84.91
67.93
50.95
33.97
16.98
0.00
Units : N/mmΛ2(MPa)


etmext : 3 Result
Subcase - Static Loads 1, Static Step 1
Stress - Element-Nodal, Averaged, Von-Mises
Min : 0.00, Max : 166.38, Units : N/mmΛ2(M
Deformation : Displacement - Nodal Magni
166.38
152.52
110.92
97.06
83.19
69.33
55.46
41.60
27.73
13.87
0.00

7.4 Summary

The design load effects slightly overshoots the criteria of the maximum von Mises stress of 200MPa with the ultimate loads for DLC 1.3. This effect is located on one of the support legs and is tensile stress. More stiffness can be obtained by increasing the length of the flange that extends from the base of the bedplate.

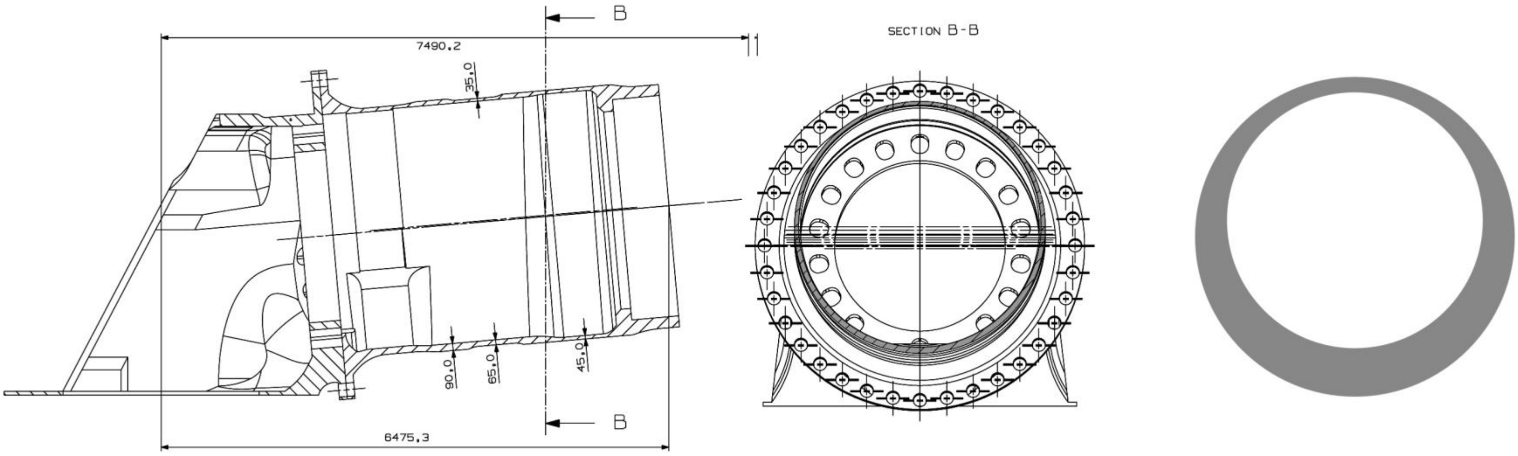


Figure 8.1 - Nose profile for the optimized design

The uneven nose profile (see fig 8.1) causes the high horizontal bending moments to induce high stress on the nose, as shown in ETM13-3 and 4. This can be reduced by adding a larger chamfer / blend from the front bearing facing backwards.

The load effect located on the bottom of the turret in the transition to flange is compressive and is considered to dominate the structure for most of its lifetime based on the weibull distribution shown in figure 2.1.

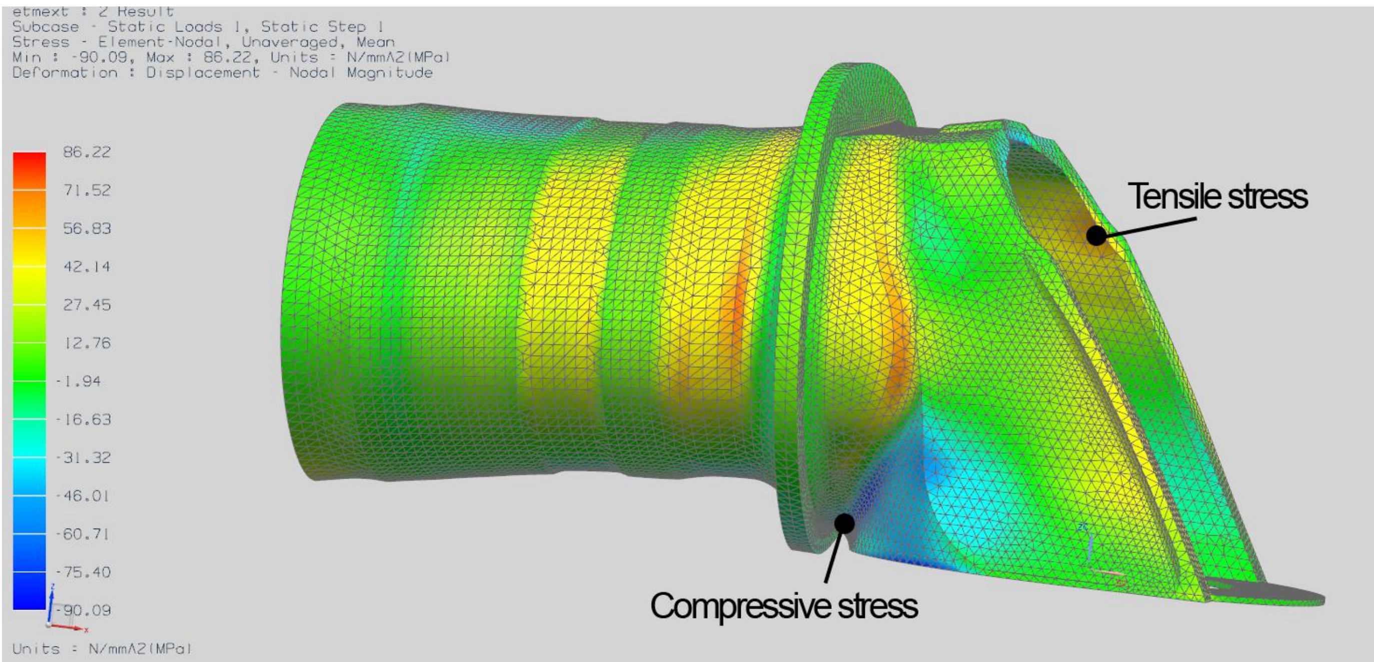


Figure 8.1 - Mean stress for ETMEXT2

8 Conclusion

The assignment was open for mechanical design and dimensioning of several important rotor nacelle components for the NOWITECH 10MW reference turbine. The work was concentrated on mechanical design of the bedplate with the intention of developing a weight efficient solution. In addition a brief assessment of possible solution for the main shaft bearings was conducted.

The main purpose for the bedplate is to provide support for the rotor nacelle assembly components and transmit the aerodynamic loads to the tower. Active loads on the rotor nacelle assembly for very large wind turbines are dominated by bending moments caused by extreme wind conditions. The ultimate loads have been estimated as those caused by extreme turbulence conditions with a 50 year recurrence period. The obtained ultimate loads have been combined to represent multi-axial loading under these conditions, as suggested by the IEC61400-1 standard. These combined loads have been used as dimensioning cases. The highest acceptable equivalent stress in the component was set to 200MPa under the extreme aerodynamic loads. The bedplate will be made out of cast steel, this required the component to be divided into two components with a max mass limit of 50tons per component.

The presented bedplate is a weight reduction of more than 100tons compared to the naïve design, having a total weight of 82,4tons. This weight estimate requires the base to have infinite stiffness and is therefore expected to increase as a yaw system is developed.

Ultimate strength analysis based on the worst-case loads encountered in DLC 1.3 results with a maximum experienced stress factor of 208.3MPa in the flange transition of the turret. This stress is of a compressive nature. These results was obtained under the assumption that the flange is able to supply enough pretension to transmit the loads efficiently and that the base have infinite stiffness. The bedplate is expected to distribute the gravitational and aerodynamic loads in an evenly distributed manner, based on the geometry. This has not been verified and a closer assessment of the load transfer to the tower through a yaw system is required.

Based on a preliminary basic lifetime estimating using the ultimate loads, a main shaft bearing solution consisting of a double tapered and a spherical bearing has been suggested. A TDO configuration for the double tapered is preferred, but has not been located in any online product catalog. As the loads distributed on the bearings are significantly higher for the front fixed bearing, a smaller bearing is suggested for the floating bearing to reduce weight for the bedplate and main shaft, overall having a significant impact on the tower head weight.

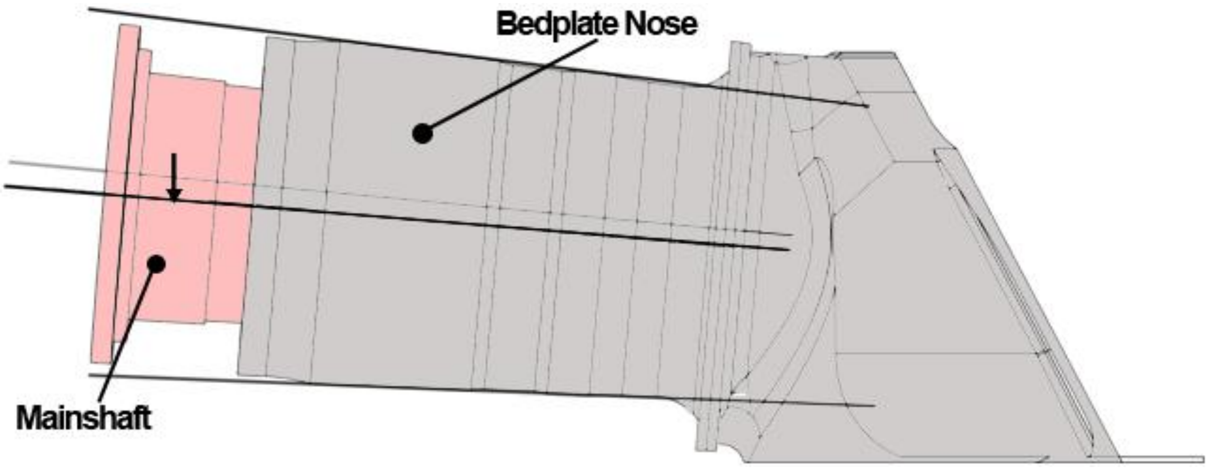
8.1 Future work

Suggestions for future work for the bedplate are summarized below

- Future mechanical design on the bedplate should include dimensioning of the bolt connections for the flange.
- A yaw system must be developed and implemented for the bedplate. A double row tapered roller bearings is suggested for the yaw bearing, as this is commonly used for large wind turbines.
- The bedplate design has not been evaluated in regard to fatigue. Fatigue for steel constructions should not be under estimated, especially for high cycle load conditions

subjected to large wind turbines. A fatigue analysis should include areas subjected to high stress such as the nose transition on the turret and outer edge of the support legs.

- Selection of steel quality for the bedplate.
- A assembly order for the rotor nacelle assembly should be developed.
- Further weight reductions can achieved if the generators inner dimensions can be modified easily. A conical shape is proposed in the figure below.



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Appendix A - Results from FAST analysis

Design Load Case	Force conditions	Sensor	max	min
1.1 - Normal Turbulence Model	Wind speed = 13m/s	Fx [kN]	1632	
		Fy [kN]	110	
		Fz [kN]	-1461	-1714
		Mx [kNm]	8559	
		My [kNm]	10390	-6837
		Mz [kNm]	8500	
	Ultimate loads	Fx [kN]	1882	
		Fy [kN]	300	
		Fz [kN]	-1285	-1866
		Mx [kNm]	10790	
		My [kNm]	23430	-14430
		Mz [kNm]	19210	
1.3- Extreme Turbulence Model	Wind speed = 13m/s	Fx [kN]	1746	
		Fy [kN]	132	
		Fz [kN]	-1410	-1806
		Mx [kNm]	9129	
		My [kNm]	14550	-10440
		Mz [kNm]	15330	
	Ultimate loads	Fx [kN]	2176	
		Fy [kN]	340	
		Fz [kN]	-1542	-1919
		Mx [kNm]	11690	
		My [kNm]	28660	-16050
		Mz [kNm]	22480	

Appendix B - Complete Simulation cases

	Component	1: V-H+	2: V-H-	3: V+H-	4: V+H+
DLC 1.1 - NTM13	F_a [kN]	1469			
	T_gen [kNm]	11555			
	F_r1_z [kN]	-6162	-6162	-603	-603
	F_r2_z [kN]	-3848	-3848	1369	1369
	F_r1_y [kN]	2707	-2707	-2707	2707
	F_r2_y [kN]	2558	-2558	-2558	2558
DLC 1.1 - NTMEXT	F_a [kN]	1694			
	T_gen [kNm]	14567			
	F_r1_z [kN]	-8710	-8710	3567	3567
	F_r2_z [kN]	-6191	-6191	5302	5302
	F_r1_y [kN]	6242	-6242	-6242	6242
	F_r2_y [kN]	5837	-5837	-5837	5837
DLC 1.3 - ETM13	F_a [kN]	1571			
	T_gen [kNm]	12324			
	F_r1_z [kN]	-7419	-7419	715	715
	F_r2_z [kN]	-4981	-4981	2618	2618
	F_r1_y [kN]	4722	-4722	-4722	4722
	F_r2_y [kN]	4544	-4544	-4544	4544
DLC 1.3 - ETMEXT	F_a [kN]	1958			
	T_gen [kNm]	15782			
	F_r1_z [kN]	-9303	-9303	4449	4449
	F_r2_z [kN]	-6712	-6712	6531	6531
	F_r1_y [kN]	7277	-7277	-7277	7277
	F_r2_y [kN]	6818	-6818	-6818	6818

Appendix C - Simulation Calculations and Cases

The ultimate loads for extreme turbulence model (DLC 1.3 ETMEXT) are calculated as:

$$M_{1\_V+} = 28660kNm - 1542kN*3,7m - 294,3kN*1,2m + 450kN*0,3 = 22736kNm$$

$$M_{1\_V-} = -16050kNm - 1919kN*3,7m - 294,3kN*1,2m + 450kN*0,3 = -23368kNm$$

$$M_{2\_\max} = 22480kNm + 340kN*3,7m = 23738kNm$$

gives for the DLC 1.3 ETM EXT (partly taken from bearing loads)

$$F_{r2\_y} = \frac{23738kNm}{4,7m}*1.35 = 6815kN$$

$$F_{r1\_y} = 6815kN + 340kN*1.35 = 7274kN$$

$$F_{r2\_z\_\max} = \frac{22736kNm}{4,7m}*1,35 = 6530kN \text{ (downwards)}$$

$$F_{r1\_z\_\max} = 6530kN - 1542*1.35 = 4448kN \text{ (upwards)}$$

$$F_{r2\_z\_\min} = \frac{-23368kNm}{4,7m}*1,35 = -6717kN \text{ (upwards)}$$

$$F_{r1\_z\_\min} = -6717kN - 1919*1.35 = -9307kN \text{ (downward)}$$

F_a = 2176*0.9 = 1958kN

T_gen = 11690kNm*1.35 = 15782kNm

For no-wind conditions $F_a = 0kN$ pure gravitational loading of the hub and rotor has to be calculated.

$$F_{r2\_nowind} = \frac{F_{zR}*y_r + F_{gr}*y_{gr} + F_s*y_s}{y_g}$$

with $F_{zR} = (27,5t*3 + 83,6t)*9,81\frac{m}{s^2} = 1630kN$

$$F_{r2\_nowind} = \frac{1630kN*3,7 + 294,3kN*1,2m - 450kN*0,3m}{4,7m} = 1330kN \text{ (upwards)}$$

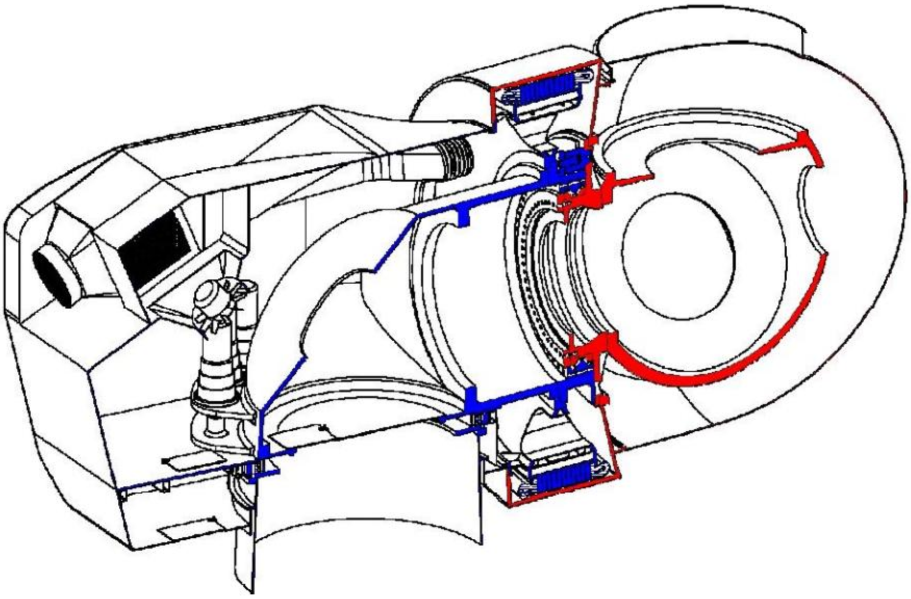
$$F_{r1\_nowind} = 1330kN + 1630kN + 294,3kN + 450kN = 3713kN \text{ (downwards)}$$

With a load safety factor γ_f of 1,35 (all these forces are considered unfavourble) the complete loading under no wind conditions are presented in the figure below

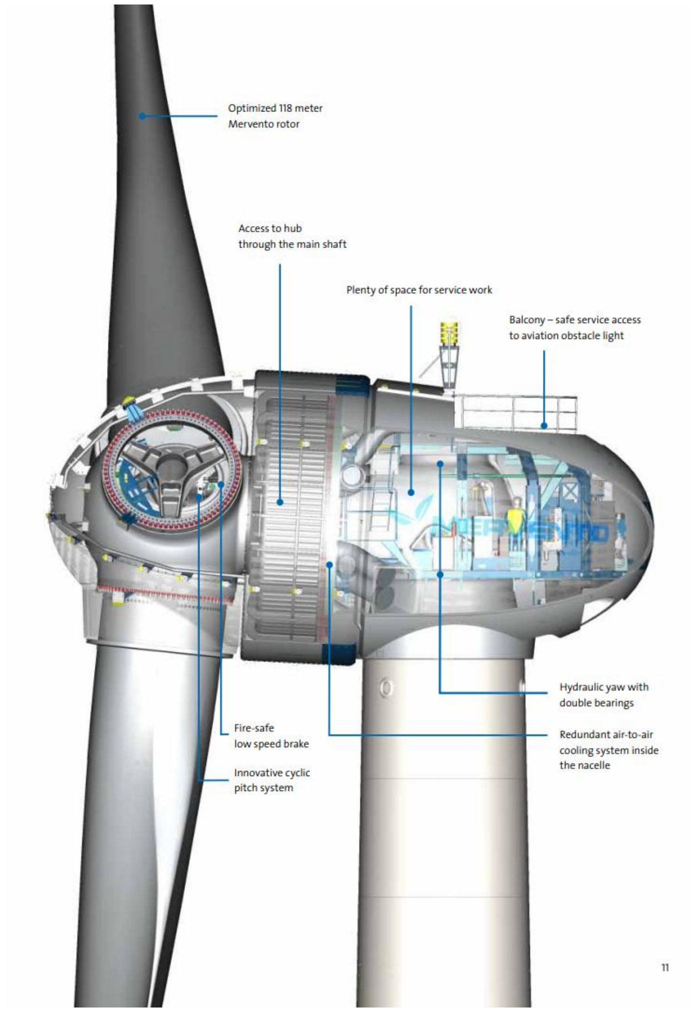
$$F_{gs} = 1,35 * 1962 kN = 2649 kN$$
$$F_{gr} = 1,35 * 294,3 kN = 397 kN$$
$$F_{bedplate} = 1,35 * 981 kN = 1324 kN$$
$$F_{r2} = 1,35 * 1330 kN = 1800 kN$$
$$F_{r1} = 1,35 * 3713 kN = 5013 kN$$

Appendix D - Rotor nacelle assemblies

Vensys 2,5MW drivetrain



Mervento 3.6-118



Appendix E - Bearing calculations

Bearing	BT2B 332497/HA4		
C [kN]	34700		
C_0 [kN]	108000		
F_r [kN]	12000		
F_a [kN]	2200		
	Calculate		
e	0.94		
X	1	X_0	1
Y	0.72	Y_0	0.7
P [kN]	13584	P_0 [kN]	13540
L_{10} [Mrev]	23		

Bearing	248/1800 CAK30FA/W20		
C [kN]	20000		
C_0 [kN]	63000		
F_r [kN]	8000		
F_a [kN]	0		
	Calculate		
e	0.15		
X	1	X_0	1
Y	4.5	Y_0	4.5
P [kN]	8000	P_0 [kN]	8000
L_{10} [Mrev]	21		

Appendix F - Overview of digital attachments

The rotor nacelle assembly structure is provided as step file. The components in the step file have been obtained as follows:

Parts developed in this project:
Tower_2_3_2.prt - Tower part of the bedplate
Nose_2.part.prt - Nose part of the bedplate
MAINSHAFT

Parts obtained from the SKF:
Double tapered roller bearing
Spherical roller bearing

Parts obtained from the NOWITECH:
The remaining components.