WEC Development Project

Winter Semester 2025/2026

Final Design Report

Project Name: **Optimus Syria**

Sub Project: **Rotor Bearing System**

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Date: 28.01.2026

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1. Introduction

Renewable energy sources are inexhaustible and they are environmentally safe meaning that they do not produce much pollution as compared to the conventional energy systems. Among them, the wind energy has emerged to be one of the most exploited renewable energy in the world [1].

Wind turbine Optimus Syria development is a project under development by the Wind Engineering Master Students of Flensburg University of Applied Sciences in collaboration with Kiel University of Applied Sciences to design a giant size wind turbine (5 MW) to fit in Syria that is facing other challenges such as infrastructure constraint, land purchasing problem, ecological problem, logistical problems and so on [2].

A 5 MW turbine having a rotor diameter of 160 m is selected to fulfill the requirements and the work load of the project is distributed among different groups in which each group is specialized in various issues. The semi-integrated concept (3-point Suspension system) of drive train includes a cast iron shaft in order to pass the rotor torque from the hub into the gearbox. All the variations founded on this idea were analyzed and examined with the purpose to have a successful drive-train system that produces the necessary output.

This report focuses on calculations, design, and optimization of various variables of a Rotor bearing system of the mechanical driving system of the Wind Turbine and is more informative on how the rotor shaft, main bearing, and main bearing housing will be designed. The design of all the components is made according to the calculations according to the international standards of the wind turbines.

It also includes the results, conclusions and open issues that can be used in future development of the project.

2. Research Competition analysis /benchmark

The design process of the main shaft including main bearings is illustrated in figure 1. In our Optimus Syria project, the torque, rotor aerodynamics forces & moments, rotor & drivetrain weights are analyzed according equilibrium formulas from NREL sizing and dimensioning of shaft the next is calculating all the applied stress on the shaft.

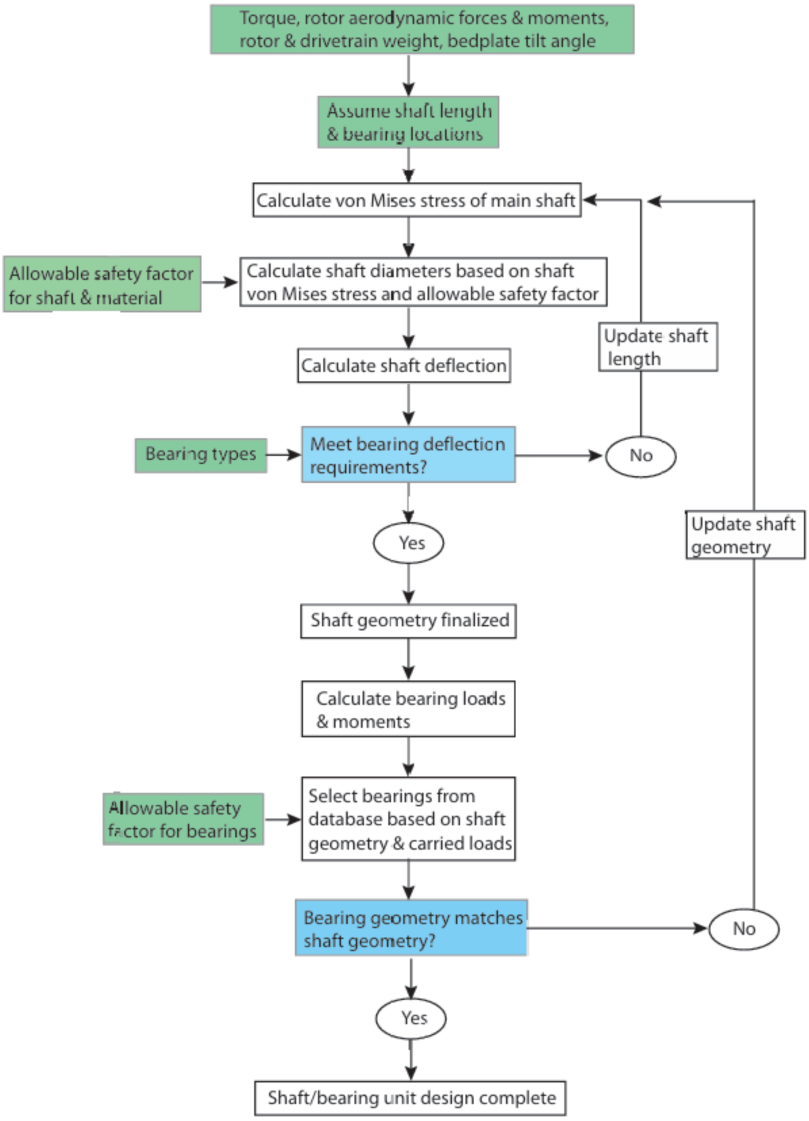


Figure 1: Flowchart of main shaft/bearing sizing model (source : NREL)

3. Relevant Standards and Guidelines

- Analytical Strength Assessment of Components: FKM Guideline 6th Edition 2012.
- DNVGL-ST-0361, Machinery for wind turbines.
- DNV GL-ST-0437, Loads and site conditions for wind turbines.
- DIN EN 10083-3:2007-01 Steels for quenching and tempering-Part3.
- DIN EN ISO 4014/4017 DIN 931/933, Hex head screws Grade 10.9.
- DIN EN ISO 4762 Grade 10.9.
- IEC 61400-1, Wind turbine- Design requirement.
- ISO 2982-2: 2013 Rolling Bearing accessories.
- ISO 4016; 4018, Bolt with nut.
- ISO 4032, 4034, Nuts.
- ISO 1896, Tight-fitting bolts.

4. Rotor Shaft

For the Optimus Syria project, the selected drive train is Semi-integrated (Three-Point Suspension) with one main bearing (fixed) attached to the machine bed and the other bearing (floating) attached to the gearbox. A bolted flange connection is selected to fix the hub with the shaft where the shaft will transfer the rotor torque from the hub to the gearbox and bolted flange connection is selected to connect the rear end of the shaft with the gearbox. Integrated rotor lock disc is part of the shaft

4.1 Design Work

The rotor shaft is chosen to be made from Cast Iron EN-GJS-400-18-LT since it is cheaper, less weight, suitable for low-temperature applications, and has excellent machinability. Below are the material properties as per the FKM Guideline [9]. Table 5.1.12

EN-GJS-400-18-LT			
Material Properties	**Symbol**	**Value**	**Unit**
Density	ρ	7250	Kg/m^3
Elongation at Fracture	A_5	18	%
Tensile Strength	$R_{m,N}$	400	MPa
Yield Strength	$R_{p0.2,N}$	250	MPa
Fatigue Limit, Tension/Compression	$\sigma_{w,Zd,N}$	135	MPa
Fatigue Limit, Shear Stress	$\tau_{W,s,N}$	90	MPa

Table 1 Material properties for EN-GJS-400-18-LT

4.2 Description of Process

4.2.1 Shaft Dimensioning

The project's initial load estimates were derived from Optimus Shakti data and subsequently scaled to Optimus Syria specifications under the guidance of Dr. Ing. Peter Quell. By maintaining a consistent calculation methodology, we achieved an optimal shaft design. Although final ultimate loads were provided by the Load Team later in the process, our current design remains based on the validated Optimus Syria scaled-up values.

The purpose of this chapter is to design the inner-diameter of the shaft, the outer-diameter at the bearing seat and the outer-diameter at the Gearbox connection to ensure that the shaft is designed to carry the loads which act on it. For the computation of Ultimate stress and Strength of the shaft, DLC 4.2 data given by the Loads team is taken into account.

4.2.2 Ultimate Limit State Calculation (ULS)

The following Figure 1 represents a static system, where the forces (FA and FB) act on the shaft and the moments distribution along its length in equilibrium. The bending moment is constant from the shaft's front end to the bearing seat and then it decreases until neglectable at the rear end where it is connected to the gearbox.

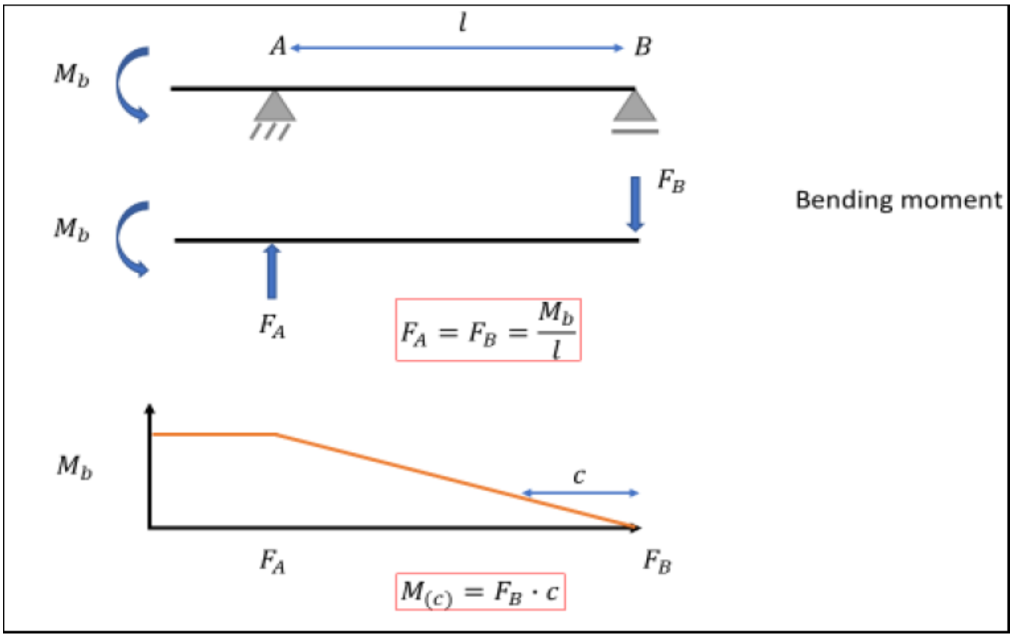


Figure 2: Forces & moments acting on the Rotor shaft in equilibrium

Once the Ultimate load data is secured, the static stress calculation at the bearing seat center is done by considering the highest resultant moment $Mres$ based on Mya & Mza values as shown in the table below.

Parameter	LSShftFxa (kN)	LSShftFya (kN)	LSShftFza (kN)	LSSTipMxa (kN-m)	LSSTipMya (kN-m)	LSSTipMza (kN-m)	LSShftFxy1 (kN)	LSShftMxy1 (kN-m)
LSShftFxa	**- 1.258E+003**	- 2.630E+003	- 1.396E+003	- 9.470E+002	4.572E+003	- 2.326E+003	2.915E+003	4.669E+003
LSShftFxa	**1.613E+003**	- 2.516E+003	- 2.309E+002	6.836E+003	2.677E+003	8.937E+002	2.989E+003	7.342E+003
LSShftFya	2.275E+002	**- 3.463E+003**	- 5.030E+002	- 2.726E+003	- 1.528E+004	- 3.389E+003	3.470E+003	1.552E+004
LSShftFya	1.252E+002	**3.627E+003**	- 4.389E+002	- 3.474E+003	8.030E+003	- 1.324E+004	3.630E+003	8.749E+003
LSShftFza	3.726E+002	- 4.437E+002	**- 3.632E+003**	4.682E+003	- 3.128E+004	- 1.624E+004	5.794E+002	3.163E+004
LSShftFza	2.527E+002	- 7.529E+002	**3.684E+003**	- 3.460E+003	- 6.815E+003	2.422E+004	7.942E+002	7.643E+003
LSSTipMxa	- 3.276E+002	3.514E+002	- 2.406E+003	**- 2.780E+004**	- 1.598E+004	5.503E+003	4.805E+002	3.206E+004
LSSTipMxa	- 2.372E+002	2.202E+003	- 1.292E+003	**2.514E+004**	1.001E+004	- 1.391E+003	2.215E+003	2.706E+004
LSSTipMya	3.872E+002	- 2.759E+002	- 3.561E+003	3.587E+003	**- 3.244E+004**	- 2.087E+004	4.754E+002	3.264E+004
LSSTipMya	- 1.538E+002	3.248E+003	5.994E+002	- 9.844E+003	**3.398E+004**	7.900E+003	3.252E+003	3.538E+004
LSSTipMza	4.932E+002	2.634E+003	- 3.005E+002	- 4.343E+003	- 1.368E+003	**- 3.254E+004**	2.680E+003	4.553E+003
LSSTipMza	2.589E+002	- 2.512E+003	2.255E+003	- 5.453E+003	1.135E+003	**3.390E+004**	2.526E+003	5.570E+003
LSShftFxy1	3.758E+000	- 2.665E+000	- 2.639E+003	- 4.729E+003	- 1.428E+004	2.738E+003	**4.607E+000**	1.505E+004
LSShftFxy1	1.252E+002	3.627E+003	- 4.389E+002	- 3.474E+003	8.030E+003	- 1.324E+004	**3.630E+003**	8.749E+003
LSShftMxy1	2.047E+002	- 2.651E+003	- 7.977E+001	- 1.530E+000	- 2.253E+000	2.734E+003	2.659E+003	**2.723E+000**
LSShftMxy1	- 1.538E+002	3.248E+003	5.994E+002	- 9.844E+003	3.398E+004	7.900E+003	3.252E+003	**3.538E+004**

Table 2 : DLC 4.2 Extreme Loads from Loads team

Parameter	DLC Name	Calculated Extreme	RotTorq (kN-m)	Time (sec)	Wind1VelX (m/sec)
RotTorq	DLC_4.2	- 2.059E+004	**- 2.059E+004**	2.134E+002	1.477E+001
RotTorq	DLC_4.2	1.862E+004	**1.862E+004**	2.126E+002	1.560E+001

Table 3 : DLC 4.2 Rotor torque from loads team

$$M_{Res} = \sqrt{M^2{}_{ya} + M^2{}_{za}}$$

$M_{Res} = 30098\ KN.m$ (Value taken from Optimus shakti), M_T =15242 KNm Value taken from Optimus shakti).

$$\frac{M_{Shakti}}{M_{Syria\ IIIA}} = \left(\frac{Dshakthi}{Dsyria}\right)^3$$

$$\mathbf{M}_{Syria,II\ A} = \frac{M_{Shakti+}M_{Syria}}{2}$$

$$\frac{L_{Shakti}}{L_{Syria\ IIIA}} = \left(\frac{D_{shakthi}}{D_{syria}}\right)^{1}$$

The above are the scaling formula used for our Optimus Syria calculation.

$Dshakti = 180\ m$

$Dsyria\quad = 160\ m$

$\mathbf{M}_{\text{Syria,II A } \mathbf{Bending}}$ = 25658 kNm (obtained after scaling)

$\mathbf{M}_{\text{Syria,II A } \mathbf{Torsional}}$ = 12993 kNm (obtained after scaling)

The above are the value we used in our calculation.

In order to determine the shaft outer diameter (D) using the maximum resulting moment (Mres) and the resulting torsional moment (Mt) on the shaft at the bearing seat, the following conditions should be met:

- Ultimate strength of shaft (ULS Strength) exceeds Ultimate stress (σULS).
- Shaft wall thickness ≥ 150 mm.
- Calculation procedure To do so, the following calculation procedure is carried out in light of the FKM Guidelines.

Intially the dimension of the shaft are assumed based on 10% of the rotor diameter(160m) which is equal to 1.6m = 1600mm. This value we have taken as our shaft outer diameter. The results are shown below:

Parameter	Description	Value	Unit	Source
D	The outer diameter of the shaft (at the bearing A seat)	1600	mm	Optimized values
d	The inner diameter of the shaft	1000	mm	
r	Fillet Radius	120	mm	
t	Notch Thickness	10	mm	

L	Length of the shaft	3760	mm	
R_Z	Surface Roughness	6.3	µm	

Table 4 : Shaft dimensions

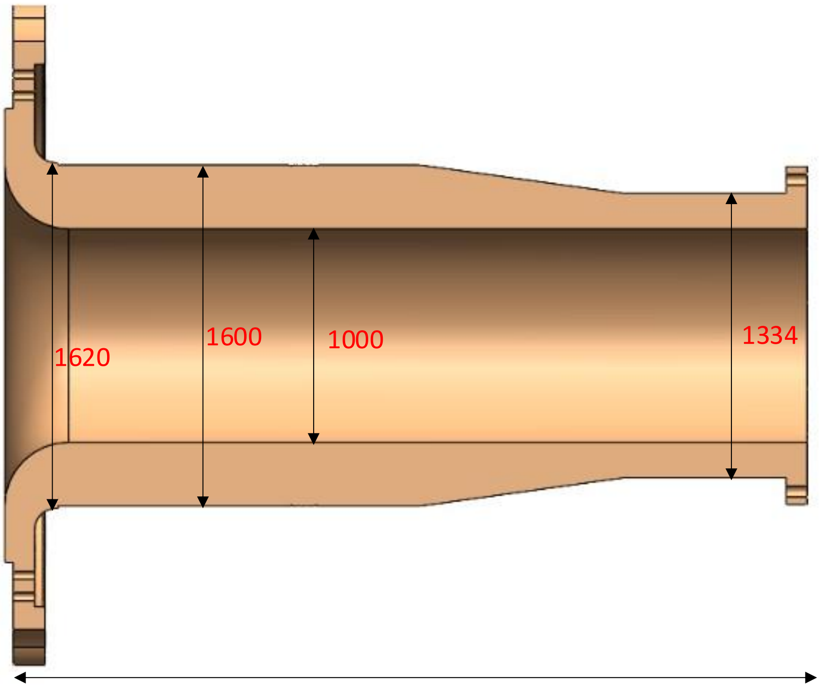


Figure 3 : Sectional view of the Rotor shaft

With the above values, the Ultimate Stress (σ_{ULS}) on the shaft is calculated.

Section modulus, $W = \frac{\pi}{32} * (\frac{D^4 - d^4}{D})$

Stress concentration factor, $K_{t,b} = 1 + \frac{1}{\sqrt{(0.62*\frac{r}{t}+20.6*\frac{r}{d}*(1+2*\frac{r}{D})^2+0.2*(\frac{r}{t})^3*\frac{D}{D'})}}$

Here, D'= D + (2×t)

Ultimate Stress, $\sigma_{ULS} = \frac{M_{res}}{W} * (K_{t,b})$

Parameter	Description	Value	Unit	Source
Mres (= Mb)	Bending Moment	25658	KNm	Scaled value from Optimus Shakti
Mt	Torsional Moment	12993	KNm	
W	Section Modulus	0.320	m^3	Fatigue Analysis course
Kt, b	Stress Concentration factor	3.09	-	FKM, equation 5.2.7
Output				
σULS	Ultimate Stress	158.75	MPa	Fatigue Analysis course

Table 5 : Calculation of Ultimate stress on the shaft

The next step is to compute the shaft's Ultimate Strength to determine whether it can support the ultimate loads for the selected dimensions. For this, we need the material's Static Yield strength and technological size factor.

As the component's dimensions increase, the material strength value decreases, which is represented by the technological size factor. The formula below is used to calculate it.

Technological Size Factor, $K_{d,m} = \frac{1-0.7686*a_{d,m}*\log\left(\frac{d_{eff}}{7.5}\right)}{1-0.7686*a_{d,m}*\log\left(\frac{d_{eff}}{7.5}\right)}$

The wall thickness of the shaft, $d_{eff} = D - d$

Static Yield strength, $R_e = K_{d,m} * K_A * R_{e,N}$

Strength of the material = $\frac{R_e}{\Upsilon_m}$

Parameter	Description	Value	Unit	Source
ad,m	Material constant	0.15	-	FKM, Table 1.2.1
deff,N,m	Diameter of test specimen	60	mm	
KA	Anisotropic factor	1	-	FKM, Table 1.2.4
deff	Wall thickness of the shaft	300	mm	FKM, Table 1.2.3
Re,N	Yield strength of the material	400	MPa	FKM, Table 5.1.4

Kd,m	Technological size factor	0.894	-	FKM, equation 1.2.8
Re	Static yield strength	357.59	MPa	Fatigue Analysis course
γM	Material partial safety factor for ULS	1.2	-	DNVGL ST 0361 Section 4.7.1
Output				
ULS Strength	Ultimate Strength of the shaft	297.99	KNm	Fatigue Analysis course

Table 6 : Calculation of the Ultimate Strength of the shaft

The calculation results show that the designed shaft is safe and can withstand the provided loads in the Ultimate limit state (ULS) as the mentioned condition is achieved.

i.e., ULS Strength ≥ σ_{ULS}

4.2.3 Rear end diameter of the shaft

The torsional moment is constant over shaft's length, as seen in Figure 4 below. The rear-end diameter of the shaft (dr) is thus determined using the following formula.

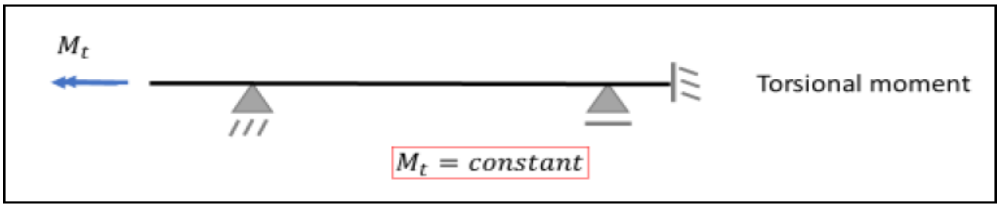


Figure 4 : Torsional moment distribution over the shaft length

The diameter of the shaft is optimized such that the Strength of the material is higher than the stress acting on the shaft as done in section 4.2.2.

Section modulus, $W = \frac{\pi}{32} * \left(\frac{D^4 - d^4}{D}\right)$

Ultimate Stress, $\sigma_{ULS} = \frac{M_t}{W}$

ULS Strength $= \frac{R_e * 0.577}{\gamma_M}$

Parameter	Description	Value	Unit	Source
D_r	Rear end outer diameter of the shaft	1334	mm	Optimized values
d	Inner diameter of the shaft	1000	mm	
M_t	Torsional Moment	12993	KNm	Loads team
W	Section Modulus	0.320	m^3	Fatigue Analysis course
γM	Material partial safety factor for ULS	1.2	-	DNVGL ST 0361 Section 4.7.1
$R_{e,N}$	Yield strength of the material	400	MPa	FKM, Table 5.1.4
Output				
σ_{ULS}	Ultimate Stress	158.75	MPa	Fatigue Analysis course
ULS Strength	Strength of the material	192.33	MPa	Fatigue Analysis course

Table 7 : Calculation of Ultimate Strength of the shaft at the rear end

The results of the calculations indicate that the dimension of the designed shaft is safe and it can withstand the specified torsional moment because the specified condition is met.

i.e., ULS Strength $\geq \sigma_{ULS}$

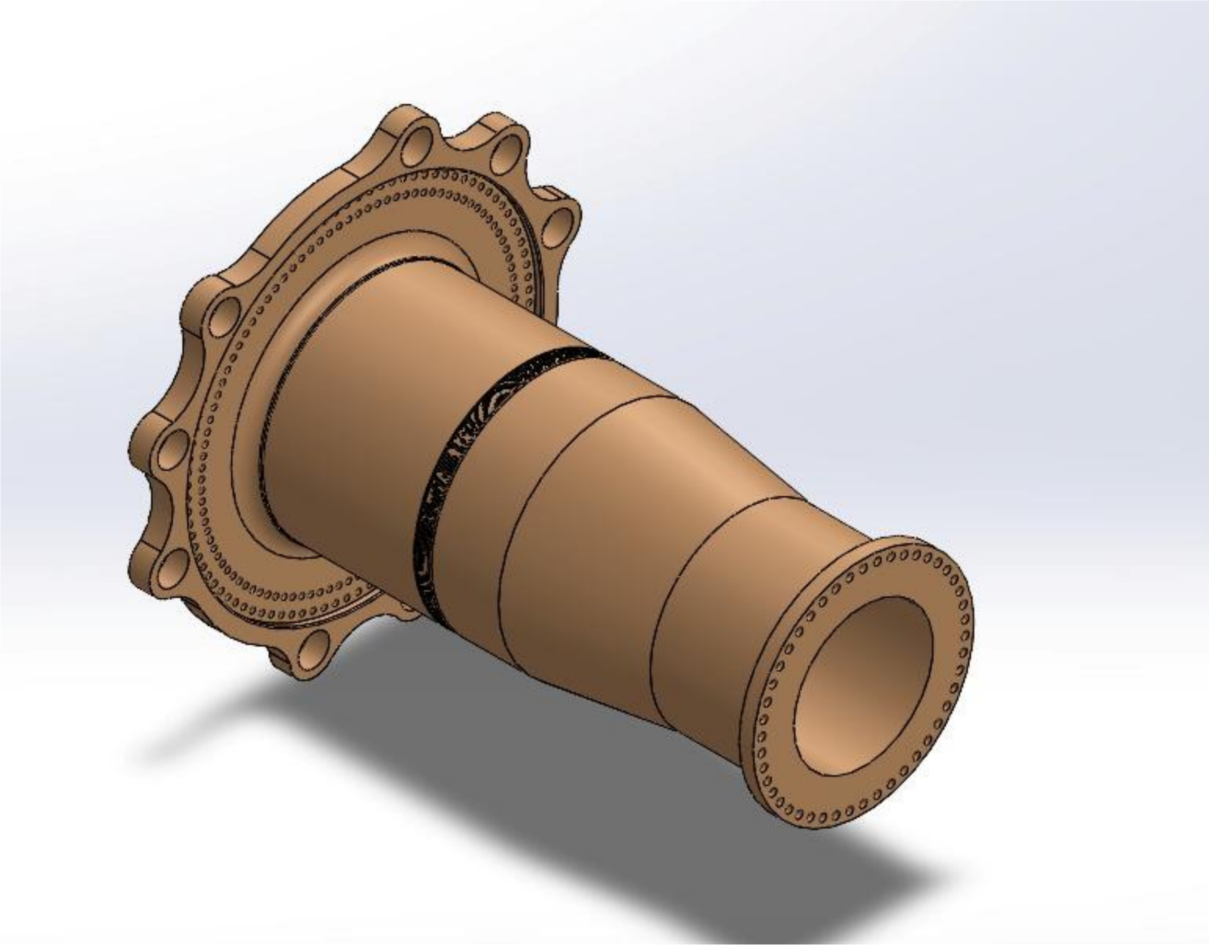


Figure 5 : Isometric view of the Rotor shaft design

4.2.4 Fatigue Calculation

Lifetime, this chapter conducts fatigue calculations of the shaft geometry based on fatigue load data as per In order to determine that the shaft is capable of carrying Damage-equivalent loads throughout its FKM Guidelines [9].

- Mean Stress factor, KAK

The mean Stress factor is calculated as a function of Mean stress sensitivity (Mσ), Mean Stress (Sm), and Mean amplitude (Sa) using the formula chosen from Figure 5, region ll.

$$K_{AK} = \frac{1}{1 + M_\sigma * \frac{S_m}{S_a}}$$

$$\text{here, } \frac{S_m}{S_a} = \frac{1+R}{1-R}$$

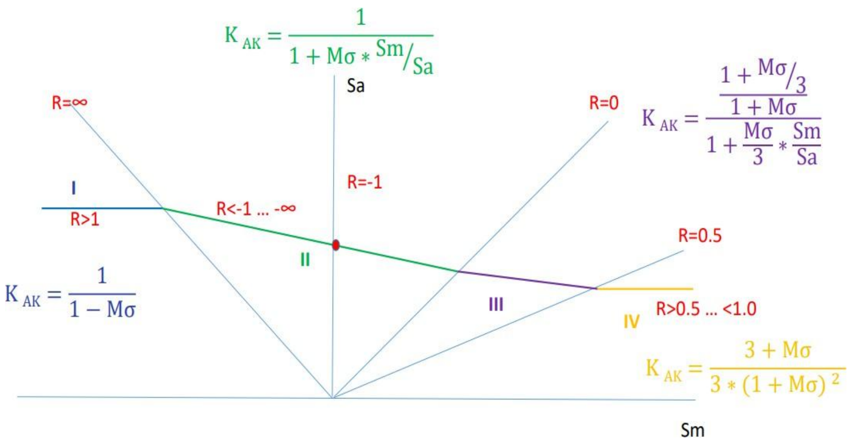


Figure 6 : : Fatigue limit for normal stresses (Haigh diagram) [9]

As stated in FKM Guidelines, taking into consideration the case of completely reversed stress leads to a stress ratio R = -1.

Therefore, **KAK** = 1

- Static Strength of the component, Rm

The static strength of the component is calculated based on the formula 1.2.1 taken from FKM Guidelines.

Rm = Kd,m × KA × Rm,N

Parameter	Description	Value	Unit	Source
Kd,m	Technological Size Factor	0.894	-	FKM, equation 1.2.8
KA	Anisotropic Factor	1	-	FKM, Table 1.2.4
Rm,N	Static Strength of Material	400	MPa	FKM, Table 5.1.4
Output				
Rm	Static Strength of component	357.60	MPa	FKM, equation 1.2.1

Table 8 : Calculation of Static strength of the component, Rm

- Mean stress sensitivity, M_σ

The Mean stress sensitivity in combination with Mean stress factor describes to what extent the mean stress affects the amplitude of the component fatigue limit.

$$M_\sigma = (a_M \times 10^{-3} \times R_m) + b_M$$

Parameter	Description	Value	Unit	Source
aM	Material constant	0.35	-	FKM, Table 2.4.1
bM	Material constant	-.05	-	
Rm	Static Strength of component	357.60	MPa	FKM, equation 1.2.1
Output				
Mσ	Mean Stress sensitivity	0.075	MPa	FKM, equation 2.4.5

Table 9 : Calculation of Mean stress sensitivity, Mσ

- Roughness factor, K_R

The roughness factor K_R accounts for the influence of the surface roughness on the fatigue strength of the component.

$$K_R = 1 - a_{R\sigma} * \log(Rz) * \log\left(\frac{2 * R_m}{R_{m,N,min}}\right)$$

Parameter	Description	Value	Unit	Source
aR,σ	Material Constant	0.22	-	FKM, Table 2.3.6
Rz	Surface Roughness of Component	1	µm	FKM Figure 2.3-2, finished surface
Rm	Static Strength of Component	357.60	MPa	FKM Eq. 1.2.1
Rm,N,min	Minimum Tensile Strength	400	MPa	FKM, Table 2.3.6
Output				
KR	Roughness Factor	1	-	FKM, equation 2.3.20

Table 10 : Calculation of Roughness factor, KR

- Fatigue notch factor, Kf,b

The bearings seat is considered as a press fitted member, as per FKM Guidelines section 5.3.3.5, "The fatigue notch factors apply to the section of the shaft where the press-fitting of the member ends". It is determined as the following.

Fatigue notch factor Kf,b is a function of Notch stress gradient nσ (r) and Nominal stress gradient nσ (d) and to calculate these values local stress gradient nσ (r) and global stress gradient Gσ (d) are required.

Notch stress gradient, nσ (r)= $1 + \sqrt{G_\sigma(r)} * 10^{-(aG+\frac{Rm}{b_G})}$

Nominal stress gradient, nσ (d) $= 1 + G_\sigma(d) * 10^{-(aG-0.5+\frac{Rm}{b_G})}$

Local stress gradient, $G_\sigma(d) = \frac{2}{d}$

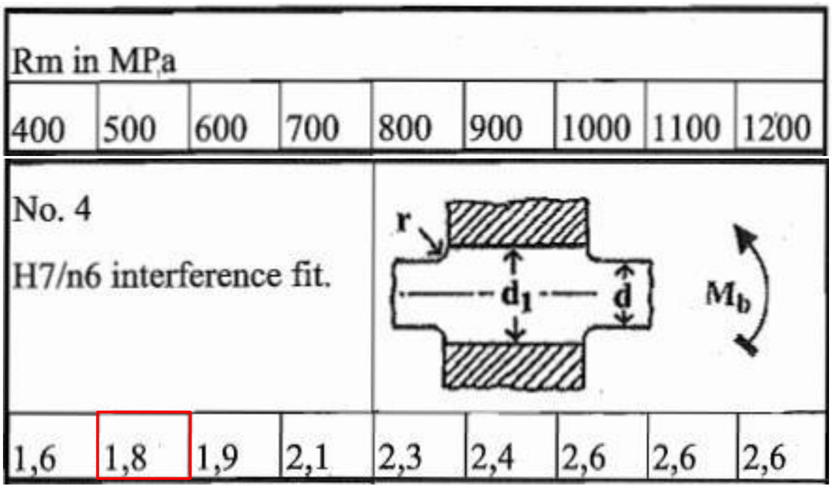
Global Stress gradient, $G_\sigma(r) = \frac{2.3}{r} * (1+\varphi)$,here , $\phi = \frac{1}{(4*\sqrt{\frac{t}{r}}+2)}$

Fatigue notch factor,$K_{f,b} = \frac{K_{t,b}}{n\sigma\,(d) * n\sigma\,(r)}$

Parameter	Description	Value	Unit	Source
aG	Material Constant	0.40	-	FKM, Table 2.3.1
bG	Material Constant	2	-	
ϕ	Stress gradient ratio	2	-	FKM, Table 2.3.5
nσ(r)	Notch stress gradient	.0167		FKM, equation 2.3.7
nσ(d)	Nominal stress gradient	1.002		FKM, equation 2.3.6
Gσ(r)	Local Stress gradient	.0167	mm^{-1}	FKM, Table 2.3.5
Gσ(d)	Global Stress gradient	0.00125	mm^{-1}	FKM, equation 2.3.16
Kt,b	Stress Concentration factor	1.98	-	FKM, equation 5.2.7
Output				
Kf,b	Fatigue notch Factor	1.75	-	Fatigue Analysis course

Table 11 : Calculation of Fatigue notch factor, Kf,b

The above-calculated value defines the factor at the notch. But the value from Figure 7, takes into consideration the small distance to the bearing seat. Hence, the Fatigue notch factor (Kf,b) value is taken as 2.1, based on the Static Strength of the component (Rm = 357.60 MPa) from FKM, Table 5.3.1.



Rm in MPa								
400	500	600	700	800	900	1000	1100	1200
No. 4 H7/n6 interference fit.								
1,6	1,8	1,9	2,1	2,3	2,4	2,6	2,6	2,6

Figure 7 : Fatigue notch factor of the shaft with a press-fitted member as per FKM [9

Therefore, **Kf,b** = 1.75

- Stress fatigue limit, SAK

After calculating all the required factors, the Stress fatigue limit SAK is calculated using the formula.

$$S_{AK} = K_{AK} * K_{d,m} * \frac{\sigma_{W,zd,N}}{K_{f,b} + \frac{1}{K_R} - 1}$$

Parameter	Description	Value	Unit	Source
$\sigma_{w,Zd,N}$	Fatigue Limit, Tension/Compression	135	MPa	FKM, Table 5.1.4
KAK	Mean Stress factor	1.08	-	FKM, equation 2.4.9
Kf,b	Fatigue notch Factor	1.75	-	FKM, Table 5.3.1
KR	Roughness Factor	1	-	FKM, equation 2.3.20
Kd,m	Technological size factor	0.894	-	FKM, equation 1.2.8
Output				
SAK	Stress fatigue limit	74.37	MPa	Fatigue Analysis course

Table 12 : Calculation of Stress fatigue limit, SAK

- Stress Range, Δσ

Stress range is calculated and compared with the allowable bending stress in order to decide which slope to use for the damage calculation of the shaft.

Allowable bending stress,$\Delta\sigma_{allowed} = \frac{\Delta S_{AK}}{j_d} = 2 * \frac{S_{AK}}{j_d}$

Stress range,$\Delta\sigma = \frac{\Delta M}{W}$

Parameter	Description	Value	Unit	Source
SAK	Stress fatigue limit	74.37	MPa	Fatigue Analysis course
jD	General safety factor	1.56	-	FKM, Table 4.5.1
$\Delta\sigma_{allowed}$	Allowable bending stress	95.34	MPa	Fatigue Analysis course
ΔM	DEL Bending moment	25658	KNm	Loads team
W	Section Modulus	0.320	m3	Fatigue Analysis course
Output				
Δσ	Stress range	80.18	MPa	Fatigue Analysis course

Table 13 : Calculation of Stress range, Δσ

Since Stress range ($\Delta\sigma$) is less than allowable bending stress ($\Delta\sigma_{allowed}$), we use the slope (m = 9) as shown in the S-N curve (Figure 8).

- Damage sum, D

To calculate the expected shaft lifetime, the damage sum is calculated which is the ratio between the reference number of cycles (N_i) and the allowed number of cycles ($N_{allowed}$), using the following equations:

$$\text{Mean amplitude, } S_a = \frac{\Delta\sigma}{2}$$

$$\text{Allowed number of cycles,} N_{allowed} = \left(\frac{\frac{S_{AK}}{jD}}{S_a}\right)^k * N_d$$

$$\text{Damage Sum, } D = \frac{N_i}{N_{allowed}}$$

Parameter	Description	Value	Unit	Source
Δσ	Stress range	80.18	MPa	Fatigue Analysis course
Sa	Mean amplitude	40.09	MPa	Fatigue Analysis course
Ni	Number of cycles	1.00E+06	cycles	Assumption
k	Slope exponent	9	-	Assumption
Nd	Number of design load cycles	1.00E+06	cycles	FKM, Table 4.2.1.1
Nallowed	Allowed number of cycles	4.75E+06	cycles	Fatigue Analysis course
Output				
D	Damage sum	0.2105	-	Fatigue Analysis course

Table 14 : Calculation of Damage sum, D

The SN curve is an efficient instrument that is used to demonstrate the fatigue behavior of the material, considering the associations between the number of cycles to failure and cyclic stress range. The two axes are represented using a logarithmic scale. In Figure 8, the vertical axis is used to display the range of stress and the horizontal axis is used to display the number of cycles.

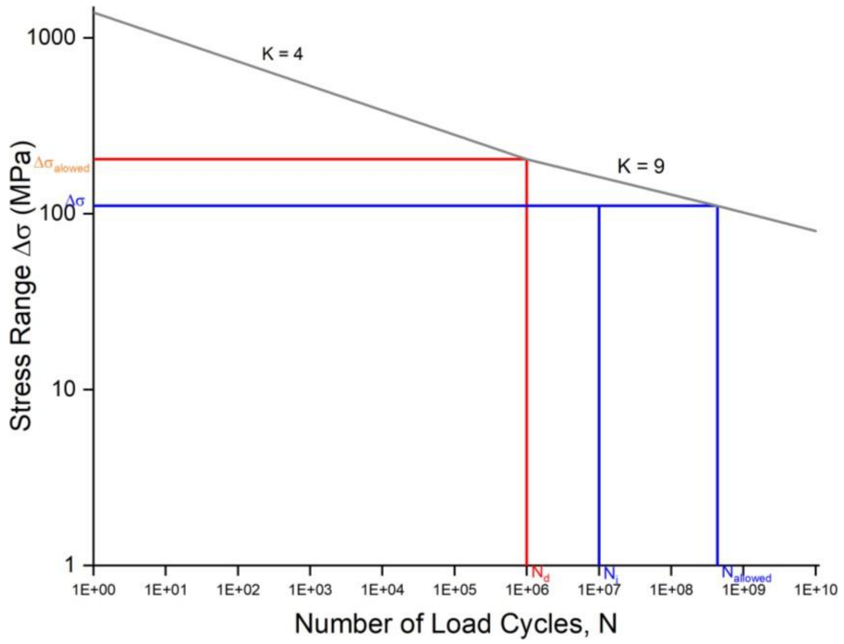


Figure 8 : S-N curve

- Stress Reserve Factor, SRF

Stress Reserve Factor is the capability of a structure to survive the loads that are above the actual loads. A significant number of structures have been deliberately constructed beyond what is required to accommodate cases of emergency, unexpected loads and so forth.

Below calculation refers to stress reserve factor for fatigue:

$$\text{Stress Reserve Factor, SRF}=D^{\frac{1}{m}}$$

Parameter	Description	Value	Unit	Source
D	Damage sum	0.2105	-	Fatigue Analysis course
m	Slope	9	-	S-N curve
Output				
SRF	Stress Reserve Factor	1.189	MPa	Fatigue Analysis course

Table 15 : Calculation of Stress Reserve Factor, SRF

- Lifetime margin factor

The lifetime margin indicates the shaft's life expectancy, which is calculated as follows.

Lifetime margin $=\frac{1}{D}$

Parameter	Description	Value	Unit	Source
D	Damage sum	0.2105	-	Fatigue Analysis course
Output				
-	Lifetime Margin factor	4.75	-	Fatigue Analysis course

Table 16 : Calculation of Lifetime margin factor

4.2.5 Conclusion

The fatigue calculation is important to ascertain the ability of the shaft to support the loads during the 20 years planned life of the shaft.

The shaft is safe based on Ultimate strength check (ULS), Fatigue Endurance check, Safety margin (Stress reserve factor). The shaft will be not damaged during the design lifetime and fatigue lifespan appears to be sufficient according to the computation. Failure will occur when 1.0 (100 percent) of the total amount of damage will occur.

While the current design is safe using the baseline material, upgrading to **EN-GJS-600-3** (Ductile Iron) would provide an even greater level of structural integrity and risk mitigation.

5. Main Bearing

A bearing is a machine part that minimizes the friction between two moving components, usually by holding a rotating or vibrating shaft. Bearings are an essential feature of machinery in terms of functionality, performance, and longevity, providing an opportunity to sustain heavy loads, minimize the wear, and provide easy movement in many different applications.

5.1 Description of Process

Axial and radial loads should be taken by the main bearing of the rotor shaft and it should be able to accomodate misalignments. So Spherica; roller bearing Type (double row, cylindrical bore) is planned as a SKF brand.

We have taken SKF 240/1120/CAF/W33 Spherical Roller Bearing as a reference and we have scaled that to our required bore size 1600mm SKF 240/1600/CAF/W33. We have also verified our scaled up dimension with the SKF company(contact person : Michael Reugels) via email and our dimesnions are similar to the given dimensions from SKF company.

5.2 Design and Calculation of Bearing

The rotor shaft should be considered in equilibrium in order to determine the forces and moments on the shaft. The forces, moments and distances were determined as illustrated in the figure below with respect to the center of the rotor hub.

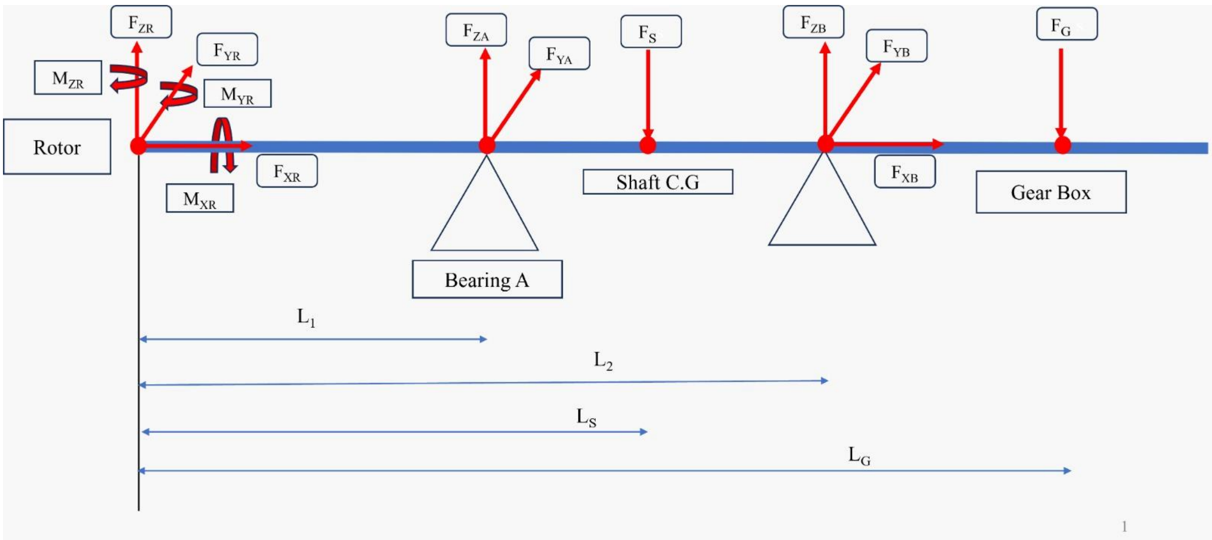


Figure 9 : Sketch for the loads equilibrium of the system.

5.2.1 Calculation of Reaction forces and moments at bearing A and B

$\boldsymbol{F_s}$: Shaft gravitational force. kN

$\boldsymbol{F_g}$: Gearbox gravitational force. kN

$\boldsymbol{F_A}$: Radial Forces in (Y, Z) direction for Bearing **A**. kN

$\boldsymbol{F_B}$: Radial Forces in (Y, Z) & Axial Force in X directions for Bearing **B**. kN

$\boldsymbol{M_{ZR}}$: Yawing Moment. kN.m

$\boldsymbol{M_{YR}}$: Tilting Moment. kN.m

$\boldsymbol{M_{ZR}}$:: Rotor Torque. kN.m

$\mathbf{l_1}$: Distance of bearing A center from Hub center. mm

$\mathbf{l_S}$: Distance of shaft center of gravity from center from Hub center. mm

$\mathbf{l_2}$: Distance of Bearing B center from Hub center. mm

$\mathbf{l_g}$: Distance of gearbox center of gravity from center from Hub center. mm

Calculation of Bearing Reaction Forces

Equilibrium of Forces at Bearing (B):

$$\sum \boldsymbol{F_X}(\boldsymbol{B}) = F_{XR} + F_{BX} = 0$$

$$\longrightarrow \boldsymbol{F_{BX}} = -F_{XR}$$

$$\sum \boldsymbol{F_Y}(B) = F_{YR} + F_{AY} + F_{BY} = 0$$

$$\longrightarrow \boldsymbol{F_{BY}} = -F_{YR} - F_{AY}$$

$$\sum \boldsymbol{F_Z}(\boldsymbol{B}) = F_{ZR} + F_{AZ} + F_{BZ} - F_s - F_g = 0$$

$$\longrightarrow \boldsymbol{F_{BZ}} = -F_{ZR} - F_{AZ} + F_s + F_g$$

Equilibrium of Moments at Bearing (B):

$$\sum \mathbf{M_Z}(\boldsymbol{B}) = \mathrm{M_{ZR}} - F_{YR} * \mathrm{l_2} - F_{AY} * (\mathrm{l_2} - \mathrm{l_1}) = 0$$

$\longrightarrow \boldsymbol{F_{AY}} = \frac{\mathrm{M_{ZR}} - \mathrm{F_{YR}} * \mathrm{l_2}}{(\mathrm{l_2} - \mathrm{l_1})}$

$\sum \mathbf{M_Y}(\boldsymbol{B}) = \mathrm{M_{YR}} + F_{ZR} * \mathrm{l_2} + F_{AZ} * (\mathrm{l_2} - \mathrm{l_1}) - F_S * (\mathrm{l_2} - \mathrm{l_S}) + F_g * (\mathrm{l_g} - \mathrm{l_2}) = 0$

$\longrightarrow \boldsymbol{F_{AZ}} = \frac{-\mathrm{M_{YR}} - \mathrm{F_{ZR}} * \mathrm{l_2} + F_S * (\mathrm{l_2} - \mathrm{l_S}) - F_g * (\mathrm{l_g} - \mathrm{l_2})}{(\mathrm{l_2} - \mathrm{l_1})}$

Weight Forces

Rotor Shaft: $\boldsymbol{F_s} = m_s * g$

Gear Box : $\boldsymbol{F_g} = mg * g$

Calculation

Weight Forces

$\boldsymbol{F_s} = m_s * g = 31183.7 * \frac{9.81}{1000} = \mathbf{305.91}$ kN

$\boldsymbol{F_g} = mg * g = 11832.58 * \frac{9.81}{1000} = \mathbf{116}$ kN

Bearing Reaction Forces at Position A (Non-Locating)

$\boldsymbol{F_{AY}} = \frac{\mathrm{M_{ZR}} - \mathrm{F_{YR}} * \mathrm{l_2}}{\mathrm{l_2} - \mathrm{l_1}} = \frac{-1861.05 - 1570.59 * 5570}{5.570 - 2.570} = \mathbf{-3536.41}$ kN

$\boldsymbol{F_{AZ}} =$

$\frac{-M_{YR} - F_{ZR} * l_2 + F_S * (l_2 - l_S) - Fg(Lg - L2)}{l_2 - l_1} = \frac{-19001.85 - 1566.97 * 5.570 + 305.91 * (5.570 - 2.63469) - 116.08 * (6.380 - 5.570)}{5.570 - 2.570} =$

$\boldsymbol{F_{AZ}} = \mathbf{-8888.56}$ kN

Bearing Reaction Forces at Position B (Locating)

$\boldsymbol{F_{BX}} = -F_{XR} =$ **-1404.5** kN

$\boldsymbol{F_{BY}} = -\mathrm{F_{YR}} - F_{AY} = -1570.59 + 3536.41 = \mathbf{1965.82}$ kN

$\boldsymbol{F_{BZ}} = -\mathrm{F_{ZR}} - F_{AZ} + F_s + F_g = -1566.97 + 8888.59 + 305.91 + 116.08 = \mathbf{7743.58}$ kN

Resultant Bearing Forces

$\boldsymbol{F_{A,Resultant}} = \sqrt{F_{AY}{}^2 + F_{AZ}{}^2} = \sqrt{-3536.41^2 + -8888.56^2} =$ **9566.23 kN**

$\boldsymbol{F_{B,Resultant}} = \sqrt{F_{BY}{}^2 + F_{BZ}{}^2} = \sqrt{1965.82^2 + 7743.58^2} =$ **7989.21 kN**

The following tables summarize the inputs and the outputs for the forces required for the bearing selection process [20]:

Inputs		
	Value	Unit
Weights		
Shaft Weight	31183.70	Kg
Gearbox Weight	11832.58	Kg
Moments		
M_{xR}	14308.27	kN.m
M_{yR}	19001.85	kN.m
M_{zR}	-1861.05	kN.m
Forces		
F_{xR}	1404.50	kN
F_{yR}	1570.59	kN
F_{zR}	1566.97	kN
Distances	Value	Unit
Hub-1st Bearing L_1	2.57	m
Hub-2nd Bearing L_2	5.57	m
Hub-CoG Shaft L_s	2.634	m
Hub- CoG Gearbox L_g	6.38	m

Outputs		
	Value	Unit
Forces		
F_S	305.91	kN
F_G	116.08	kN
First Bearing		
F_{AY}	-3536.41	kN
F_{AZ}	-8888.56	kN
Second Bearing		
F_{BX}	-1404.50	kN
F_{BY}	1965.82	kN
F_{BZ}	7743.58	kN
Resultant Forces		
$F_{A,resultant}$	9566.23	kN
$F_{B,resultant}$	7989.21	kN

Table 17 : Calculation of the forces required for bearing selection

5.2.2 Safety factor

Further calculation is needed to ensure the design safety using the following equations from SKF Bearing Catalogue.

Calculation of static safety factors

S0: $S_o = C_o / P_o$

Where:

- S_o: The static safety factor S_o is an empirical factor. According to SKF-Catalogue 4003T. When non-orbiting operation, normal load conditions, usage of roller bearings:

S_o = 1.0 (SKF). $S_o \geq 2$ according to $\underline{IEC61400-4}$

- C_o: is the radial load which corresponds to a calculated contact stress at the most heavily loaded roller element.

C_o: Basic static load rating is 50000 kN(assumption from Spherical Roller Bearing SKF catalogue 240 series).

-P_o: Equivalent static loading [N] by the relevant load case

The equivalent static loading P_o consists of radial and axial loads by:

$-P_o = X_o F_{Resultant} + (Y_o * F_a)$

With X_o, Y_o: position-dependent factor from bearing catalogue

- For Bearing A (Main Bearing).

$-P_o = X_o F_{A,Resultant}$

$-X_o \approx 1$

$\boldsymbol{P_o} = F_{A,Resultant} = \mathbf{9566.23\ kN}$

$-S_o = \frac{C_o}{P_o} > 2$ Barley safe according to $\underline{IEC61400-4}$

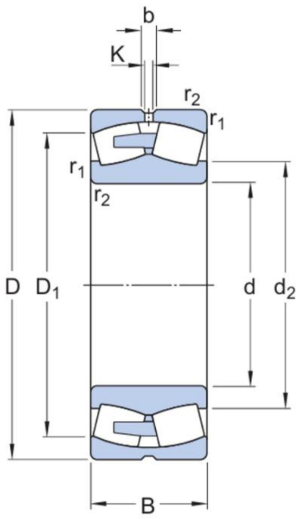
-$\boldsymbol{C_o} \approx$ 50000 kN at $S_o \approx 5.23$

Bearing Selection		
	Value	Unit
Bearing A		
Y_0	1	-
C_0	45000	kN
P_0	20219	kN
S_0	2.23	-

Table 18 : Bearing safety factor calculation

5.3 The design and dimensions of main bearing

We have taken SKF 240/1120/CAF/W33 Spherical Roller Bearing as a reference and we have scaled that to our required bore size 1600mm SKF 240/1600/CAF/W33.



d	1 120 mm	Bore diameter
D	1 580 mm	Outside diameter
B	462 mm	Width
d_2	≈ 1 268 mm	Shoulder diameter of inner ring
D_1	≈ 1 423 mm	Shoulder/recess diameter of outer ring
b	22.3 mm	Width of lubrication groove
K	12 mm	Diameter of lubrication hole
$r_{1,2}$	min. 9.5 mm	Chamfer dimension
	Normal	ISO tolerance class for dimensions

Figure 10: The dimensions of bearing 240/1120/CAF/W33 from SKF catalog.

Calculation data

SKF performance class		SKF Explorer
Basic dynamic load rating	C	22 364 kN
Basic static load rating	C_0	50 000 kN
Fatigue load limit	P_u	2 750 kN
Reference speed		130 r/min
Limiting speed		240 r/min
Limiting value	e	0.26
Calculation factor	Y_1	2.6
Calculation factor	Y_2	3.9
Calculation factor	Y_0	2.5

Figure 11: The calculation data of bearing 240/1120/CAF/W33 from SKF catalog.

The dimensions and geometrics of main bearing:

Dimensions of Main Bearing			
Items	**Symbols**	**Value**	**Unit**
Bore diameter of bearing	**d**	1600	mm
Outside diameter of bearing	**D**	2240	mm
Width of bearing	**T**	630	mm
Basic static load rating	C_0	50000	kN
Number of roller elements	**N**	28	
Diameter of roller element	**Dr**	131	mm
Length of roller element	**L**	249	mm
Materials of Bearing	**Steel (100Cr6)**		

Table 19: The dimensions of Main Bearing.

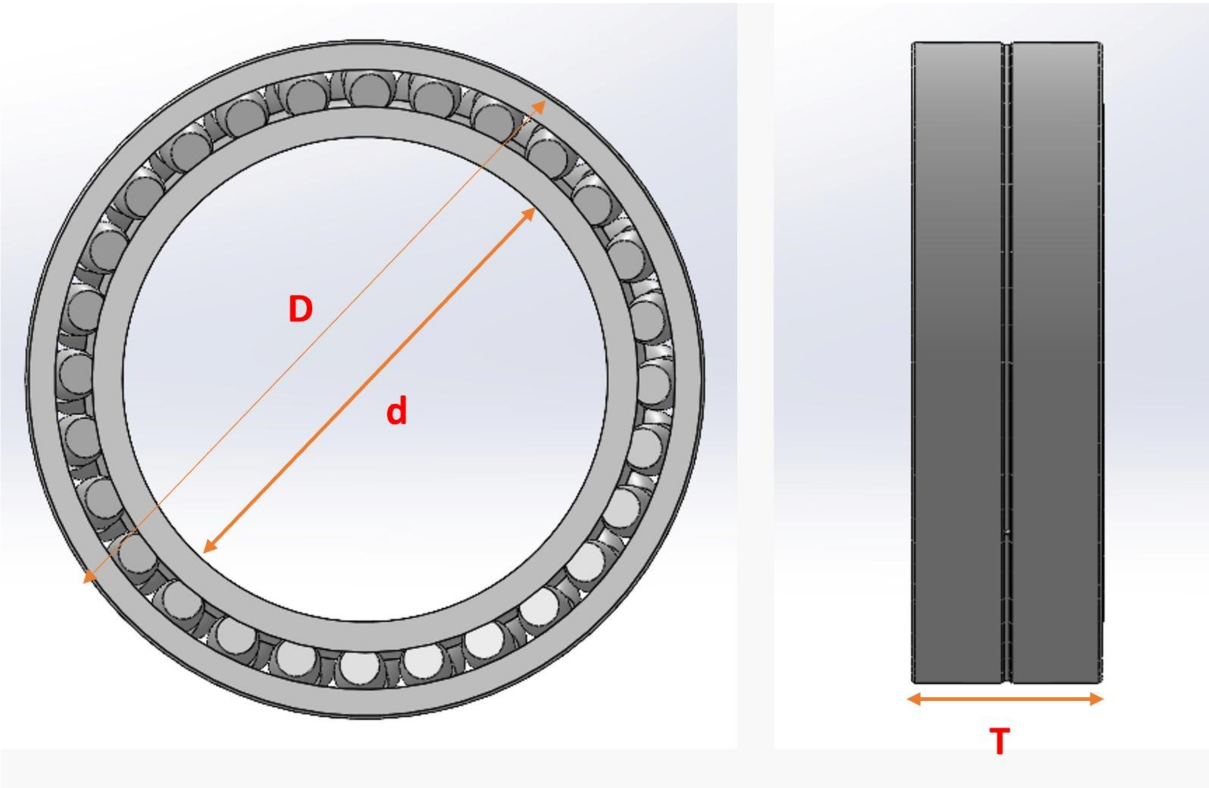


Figure 12: Our drawing was created by SolidWorks for main bearing

6. Flange Connections.

The flange is designed and cast as part of the rotor shaft, with the bolt holes machined afterward. It is a raised ridge on the hub side of the shaft that adds structural strength, allows for easier attachment and force transfer, and helps stabilize and guide the movement of the machine.

6.1 Flange Specifications (Hub side)

S.No	Description	Value	Unit
1	Outer diameter (D)	2450	mm
2	Inner Diameter	2250	mm
3	Flange thickness (t)	150	mm

Table 20 : Flange specification

6.2 Bolted Connection

The rotor shaft and rotor hub are connected by bolts using a bolted flange connection. The primary purpose of a bolted connection is to transfer loads from hub to shaft and maintain the two matching parts' connection. As seen in the figure below, the flange and rotor shaft of the Optimus Syria wind turbine will be cast on one end of the shaft where the rotor shaft will be attached to the rotor hub.

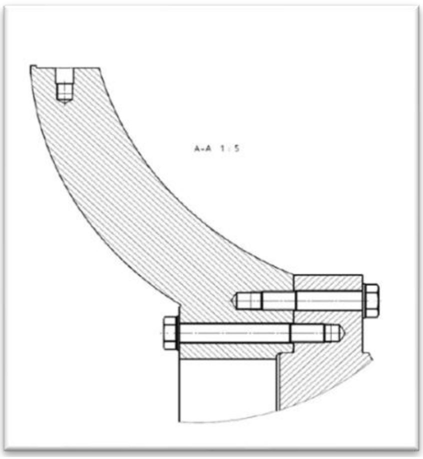


Figure 13: Selected Bolted Connection at hub side

This Bolted connection has the following advantages,

- Increased capacity to take torque and bending moments.
- A small distance between both bolt circles is a better load distribution.
- Optimal flow of forces from hub to shaft.

6.2.1 Bolts

Bolts are extremely standardized parts of machines. Following a set of guidelines is crucial when choosing or creating a bolt. We choose to use the "ISO 4014 DIN 931" standard for the Optimus Syria flange connection. M36 bolts, which are part of the 10.9 grade high tensile bolt class, are utilized in the bolted connection of the flange.

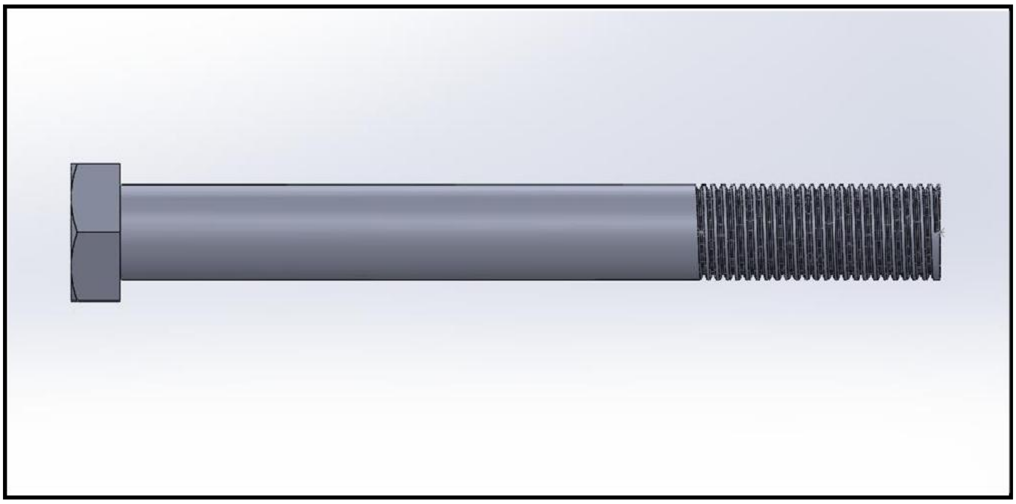


Figure14: Bolt for Flange Connection

6.2.2 Specifications of the bolts (Hub Side)

S.No	Description	Value	Unit
1	Grade	10.9	_
2	Bolt diameter (d)	39	mm
3	Bolt length (L)	350	mm
4	Length of unthreaded part (L1)	200	mm
5	Length of threaded part (L2)	150	mm

6	Length of the head (LH)	24.36	mm
7	Pitch	4	mm
8	Hexagonal Head Length	Max=60.79 Min=55	mm
9	Nominal Stress Area, As, mm^2	817	mm^2

Table 21: Bolt Specifications

6.2.3 Hydraulic Wrench.

The wrench converts high-pressure hydraulic energy from an external pump into a controlled rotational movement on the nut via a hydraulic cylinder and a ratchet mechanism. Each pressure setting of the pump corresponds to a specific torque output, so by calibrating pressure–torque charts the applied torque (and thus the approximate bolt preload) can be set reproducibly.

Various detachable sockets that can be adjusted based on the size of bolts and nuts make up impact wrenches.

We can determine the maximum number of bolts on the bolt circle diameter based on the socket's measurements.

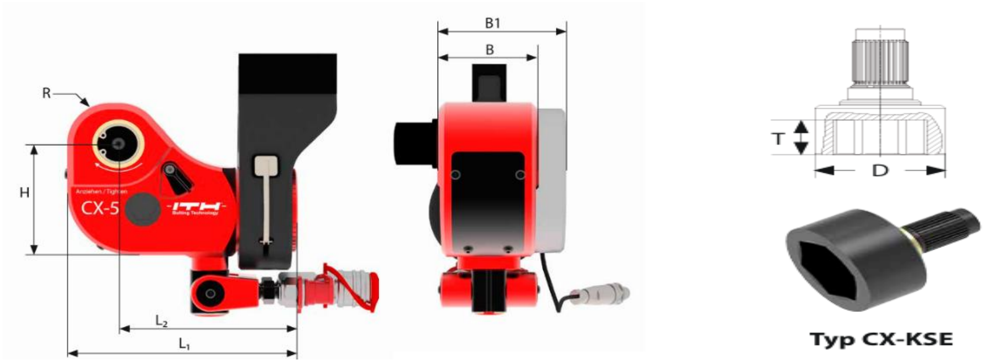


Figure 14: Hydraulic torque wrench (source : ITH)

6.3 Flange calculations.

In this section we are doing flange calculation of bolts on Hub side, Gearbox side and at housing legs.

6.3.1 Specifications of wrench for M36 Bolt.

Description	Value	Units
Maximum torque capacity	5000	Nm
L1	175	mm
L2	136	mm
B	73	mm

B1	93	mm
R	33	mm
H	91	mm
Weight	4	mm

6.3.2 Calculation of M36 bolts on Hub Side.

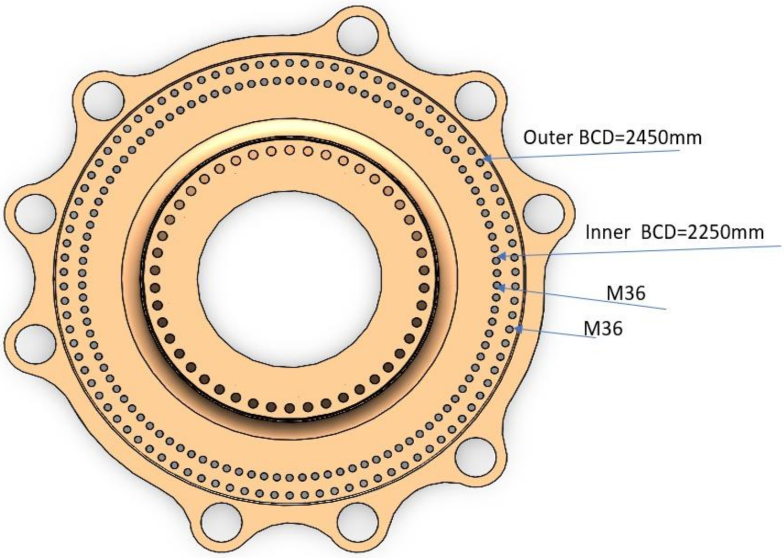


Figure 16: Hub side flange

- Applied Torsional Moment: M_t = 12800kN·m
- Applied Bending Moment (Used for Axial Load): M_B = 21573.8 kN.m

$$X = \frac{e_{nut}}{2} + Clearance + \frac{D_{tool}}{2}$$

- Bolt Counts: $n_1 = \frac{\pi \times 2450}{75.895} \approx 102$

$$n_2 = \frac{\pi \times 2250}{75.895} \approx 94$$

- Outer BCD: 2450mm =⇒ r_{max} = 1225mm
- Inner BCD: 2250mm =⇒ r_{min} = 1125mm

- Polar Second Moment of Area

$$\sum r^2 = (n_1 \cdot r_1^2) + (n_2 \cdot r_2^2)$$

$$\sum r^2 = (102 \cdot 1225^2) + (94 \cdot 1125^2) = 272{,}032{,}500\ mm^2$$

- Load due to torque moment

$$F_{q,Mt} = \frac{M_t \cdot r_{outer}}{\sum r^2}$$

$$F_{q,Mt} = \frac{12800 \cdot 1225}{272{,}032{,}500\ mm^2} = 57.64\ kN$$

- Load due to lateral force

$$F_{q,F} = \frac{F_{lat}}{n_{total}}$$

$$F_{q,F} = \frac{F_{lat}}{n_{total}} = 9.24\ kN$$

- Summation of axial components = 97.15 + 3.23=100.38 kN
- Force required to prevent slip under shear load.
- Summation of shear components = $F_{q,Mt} + F_{q,F} = 57.64 + 9.24 = 66.88$ kN

$$F_{KL} = \frac{F_Q}{\mu_T} = \frac{87}{0.4} = 217.5\ kN$$

- Load due to bending moment

$$F_{a,Mb} = \frac{M_B \cdot r_{outer}}{\sum r^2}$$

- Sum of Axial Design Load and Clamping Force

$$F_{a,Mb} = \frac{M_B \cdot r_{outer}}{\sum r^2} = 97.15\ kN$$

$$F_{erf} = F_B + F_{KL}$$
$$F_{erf} = 87 + 217.5 = 304.5\ kN$$

- Load due to static thrust

$$F_{a,zd} = \frac{F_{stat}}{n_{total}} = 32.3\ \text{kN}$$

- Stress area calculation

$$A_S = \frac{F_{erf}}{\left(\frac{R_{p0.2}}{\chi \cdot \gamma_F}\right) - \left(\frac{\beta \cdot E \cdot f_z}{l_k}\right)}$$

$$A_S \approx 578.79\ mm^2$$

6.3.3 Calculation of M48 bolts on Gearbox Side.

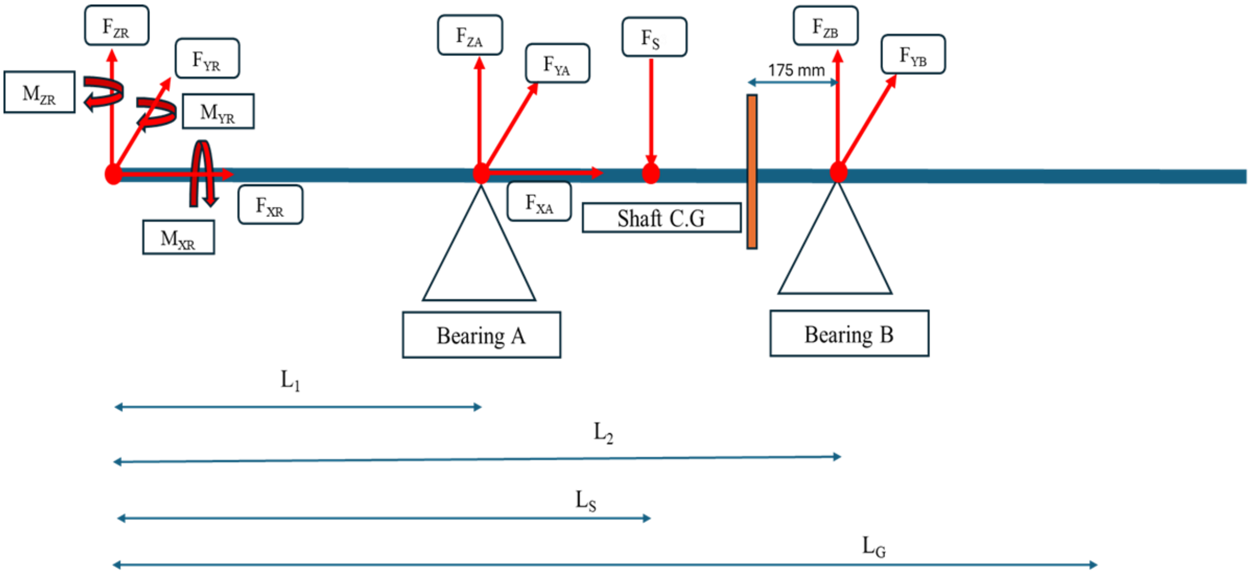


Figure 17: Force and Momentum line diagram

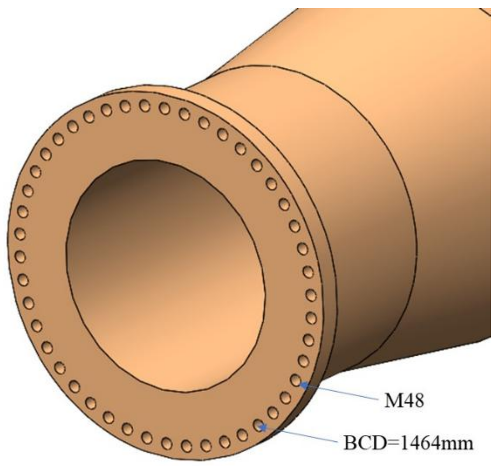


Figure 18: Force and Momentum line diagram

- Clamping length of M48 Bolts = 5*48= 240mm

$$F_{A\ result} = \sqrt{F_{AZ}^{2} + F_{AY}^{2}}$$

$$F_{By} = -F_{yR} - F_{Ay} = 1965.82\ KN$$

$$F_{BZ} = -F_{ZR} - F_{AZ} + F_S + F_G = 7839.17\ \text{KN}$$

- Resultant force on Bearing B $= F_{BResulatnt} = \sqrt{F_{By}^2 + F_{Bz}^2} =$
- Bending moment on Gear box side flange = $F_{BResulatnt} * 175mm = 1414KN.m$
- 175 mm (the sum of center gear box flange(100mm) + ½ of Torque arm (75mm))
- $T_{rated} = \frac{P*60}{2*\pi*n} = \frac{5263\ KW*60}{2*\pi*10.7} = 4697$ kN.m
- Taking Coefficient of friction = 0.6
- Tightening Factor (Ka) = 1.2
- Shear Force per bolt = $F_{q,Mt} = \frac{2 \cdot M_t}{n \cdot d_t}$ = 139.5 kN
- Required clamping force $F_{KL} = \frac{F_{q,Mt}}{\mu_T} = \frac{139.5}{0.6} = 232.5\ kN$
- $F_{erf} = F_{a,ges} + F_{KL}$ = 48.2 +232.5 = 280.7 kN
- $A_S = \frac{F_{erf}}{\left(\frac{R_{p0.2}}{\chi \cdot \gamma_F}\right) - \left(\frac{\beta \cdot E \cdot f_Z}{l_k}\right)}$ = 454.5 mm^2

6.3.4 Calculation of M42 bolts at housing legs.

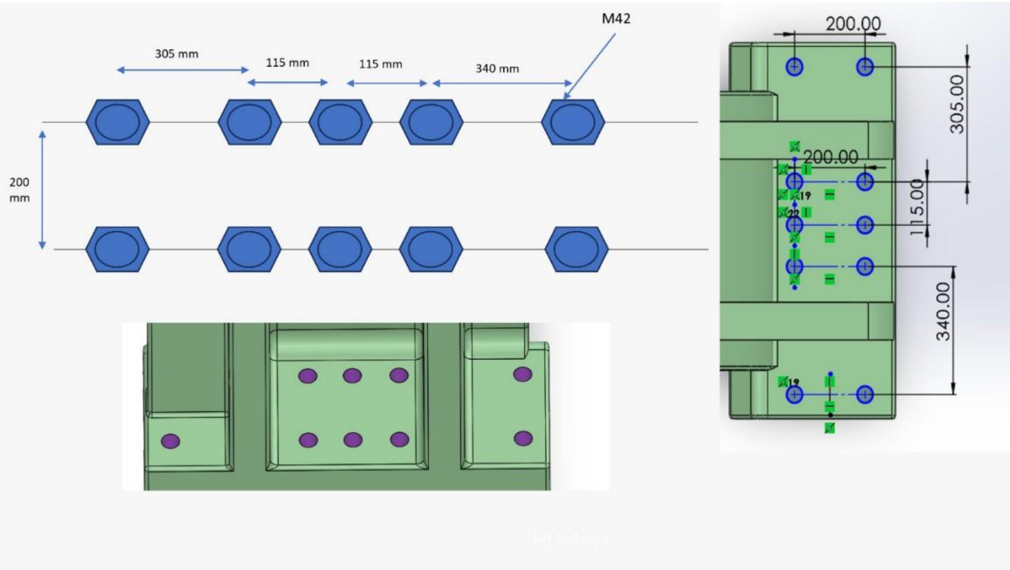


Figure 19: Arrangements of bolts at housing legs

Force Component	Value (kN)	Unit
F_{AY}	-3536.41	kN
F_{AZ}	-8888.56	kN
$F_{A,result}$	Force components 9566.23	kN

Table 23: Force components

Shear force on screw by external lateral force $F_{q,F} = \frac{F_{q,F,ges}}{n_s} = 353\ kN$

1. Distances of column Centroid :

$$\bar{x} = \frac{0 + 305 + 420 + 535 + 875}{5} = 427\ mm$$

$x_1 = 0$
$x_2 = 305$
$x_3 = 420$
$x_4 = 535$
$x_5 = 875$

2. calculating deviation from the centroid

Formula for deviation of each point:

$$d_i = x_i - \bar{x}$$

1. $d_1 = x_1 - \bar{x} = 0 - 427 = -427$
2. $d_2 = x_2 - \bar{x} = 305 - 427 = -122$
3. $d_3 = x_3 - \bar{x} = 420 - 427 = -7$
4. $d_4 = x_4 - \bar{x} = 535 - 427 = 108$

$$d_5 = x_5 - \bar{x} = 875 - 427 = 448$$

3. Square each deviation

Formula for squared deviation:

$$d_i^2 = (x_i - \bar{x})^2$$

Step-by-step:

1. $d_1^2 = (-427)^2 = 182{,}329$
2. $d_2^2 = (-122)^2 = 14{,}884$
3. $d_3^2 = (-7)^2 = 49$
4. $d_4^2 = (108)^2 = 11{,}664$
5. $d_5^2 = (448)^2 = 200{,}704$

4. Sum of squared deviations for one row

Formula:

$$\sum d_i^2 = d_1^2 + d_2^2 + d_3^2 + d_4^2 + d_5^2$$

Step:

$$\sum d_i^2 = 182{,}329 + 14{,}884 + 49 + 11{,}664 + 200{,}704 = 409{,}630 \text{ mm}^2$$

5. Account for 2 identical rows

If there are 2 identical rows of these points, the total sum of squared deviations is:

$$\text{Total} \sum x^2 = 2 \times \sum d_i^2$$

$$\text{Total} \sum x^2 = 2 \times 409{,}630 = 819{,}260 \text{ mm}^2$$

So, step-wise with formulas:

$$d_i = x_i - \bar{x}, d_i^2 = (x_i - \bar{x})^2, \sum d_i^2 = 409{,}630, \text{Total} \sum x^2 = 2 \times 409{,}630 = 819{,}260 \text{ mm}^2.$$

Total $\sum\ y^2 = 10 \times 100^2 = 100{,}000$ mm^2

6. **Total Polar Moment**

$\sum\ r^2 = \sum\ x^2 + \sum\ y^2\ =\ 819{,}260\ \ \text{mm}^2$

7. **Shear force on by torsional moment Mt**

General Formula
The maximum shear force due to torsion ($F_{q,mt,max}$) is calculated using the following formula:

$$F_{q,mt,max} = \frac{M_t \cdot r_{tmax}}{\sum\ r^2}$$

Variable Definitions
M_t: Torsional moment (12,993,000,000 Nmm)
r_{tmax}: Maximum distance from the centroid (459 mm)
$\sum\ r^2$: Sum of the squares of the distances (919,260 mm^2)

Step-by-Step Calculation
Step A: Substitute the values into the formula

$$F_{q,mt,max} = \frac{12{,}993{,}000{,}000\ \text{Nmm} \cdot 459\ \text{mm}}{819{,}260\ \text{mm}^2}$$

Step B: Multiply the numerator

$$F_{q,mt,max} = \frac{5{,}963{,}787{,}000{,}000}{819{,}260}$$

Step C: Final Result

$$F_{q,mt,max} \approx 6{,}487{,}600\ \text{N}$$

8. **Total shear force on screw by torsional and lateral forces**

$F_{q,ges}\ =\ F_{q,F}\ +\ F_{q,Mt}$ = 353 + 6487 = 6841 kN

9. **Axial force on screw by thrust force**

$F_{a,zd}\ = \frac{F_{a,zd,ges}}{n_s} = \frac{8888}{10}$ = 888 kN

$F_{a,Mbmax} = \frac{(M_B \times x_{max})}{\sum x_i^2}\ =\ \frac{25{,}658{,}000{,}000\ Nmm\ X\ \ 488mm}{819{,}260\ \text{mm}^2}$ = 14,030,693 N

10. Total axial forces on screw

$F_{a,ges} = F_{a,zd} + F_{a,Mbmax}$ = 888 + 14030 = 14919 kN

- Yield strength: $R_{p0.2} = 900$ N/mm²
- Young's modulus: $E = 210{,}000$ N/mm²
- Tightening factor: $K_A = 1.4$
- Reduction factor: $\chi = 1.19$ (from thread friction $\mu_G = 0.12$)
- Bolt compliance: $\beta = 1.1$ (for shank screw)
- Settlement: $f_z = 0.011$ mm
- Grip length: $l_k = 230$ mm
- Required preload: $F_{erf} = 4{,}425{,}300$ N

Step 1: Tightening Loss Term

Calculate the effective stress reduction due to tightening friction and factor:

$$\sigma_{tight} = \frac{\chi \cdot K_A \cdot R_{p0.2}}{1} = 1.19 \cdot 1.4 \cdot 900$$
$$\approx 1{,}498 \text{ N/mm² (raw), normalized to } \approx 540.2 \text{ N/mm²}$$

This caps the usable preload stress below yield.

Step 2: Settlement Loss Term

Compute additional stress loss from clamped parts settling under load:

$$\sigma_{settle} = \beta \cdot E \cdot \left(\frac{f_z}{l_k}\right) = 1.1 \cdot 210{,}000 \cdot \left(\frac{0.011}{230}\right) \approx 11.0 \text{ N/mm²}$$

Step 3: Net Allowable Stress

Subtract losses to get the effective stress capacity:

$$\sigma_{net} = 540.2 - 11.0 = 529.2 \text{ N/mm²}$$

Step 4: Minimum Tensile Stress Area

Divide total preload by net stress for required area:

$$A_{s,min} = \frac{F_{erf}}{\sigma_{net}} = \frac{4{,}425{,}300}{529.2} \approx 8{,}362 \text{ mm²}$$

7. Bearing Housing

One of the parts of wind turbines that support the bearing and shield it from contaminants is the bearing housing. The installation of the shaft and bearing is made possible by the bearing housing. Cast iron (EN-GJS 400-18-LT) is the material of choice for casting.

Additionally, the bearing housing is attached to the machine bed in order to transmit loads from the rotor to the machine bed.

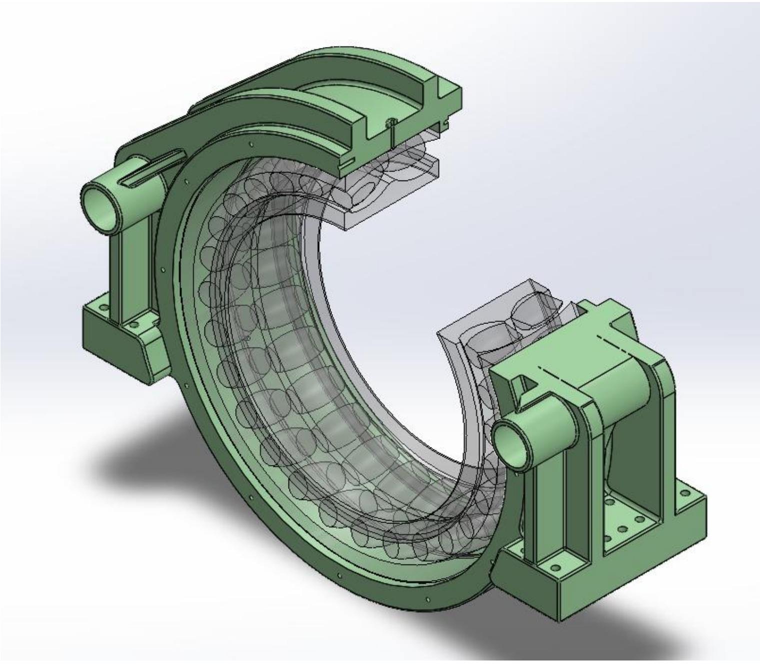


Figure 20: Section view of Bearing Housing

7.1 Design Work

The Bearing housing design is based on:

- The Bearing dimensions
- The vertical distance from the Rotor axis to the housing legs, to mount on a machine bed.
- The lock pin holes in the housing are designed at 15 degree as per rotor lock requirements.

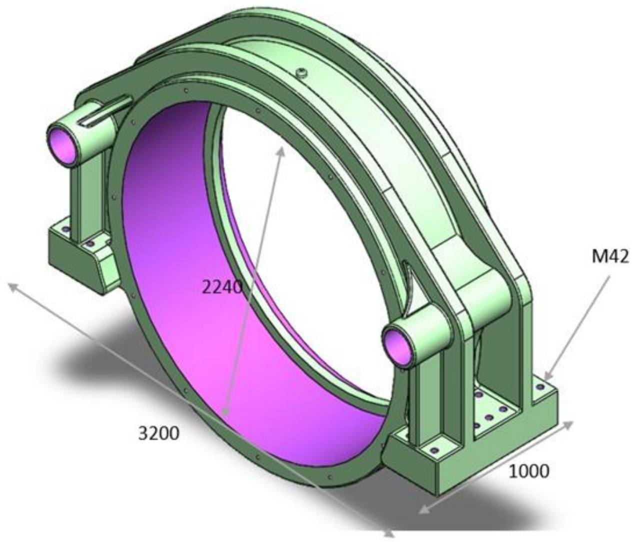


Figure 21: Details of Bearing Housing

parameter	value	unit
OD	2500	mm
ID/Bore diameter	2240	mm
Width	810	mm
Total length	3200	mm
Rib Radius	1400	mm
Height of Seat	230	mm
Length of the Legs	1000	mm

Table 24: The dimensions and geometrics of Housing

.

7.2 Design rules for casting

- Use equal wall thickness if possible.
- Strengthen and stiffen light sections with ribs.
- Provide generous fillets where two surfaces come together, eliminate sharp corners.
- Use inclined surfaces for easy demolding.
- Avoid material accumulation.
- Provide bosses around holes or openings to reinforce the part and compensate for the metal lost due to the hold.
- Use simple curves which are easy for the patternmaker to produce.

7.3 Bearing lubrication system

The bearing is lubricated with oil, which is added through the hole at the top of the bearing housing and directed to the outer ring of the bearing. From there, the oill flows across the bearing rollers and exits through the gap between the inner and outer labyrinth rings.

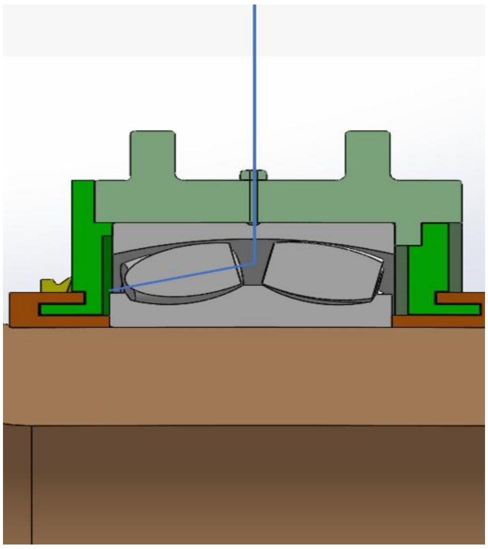


Figure 22: Bearing Housing lubrication System

8. Sealing

The oil seal plays a crucial role in preventing lubrication oil from leaking out of the system. The labyrinth rings consist of 4 parts, 2 parts each side of bearing one inner which is fixed with the shaft by lock nut and rotate with it, and one outer which fixed with bearing housing by M20 bolts.

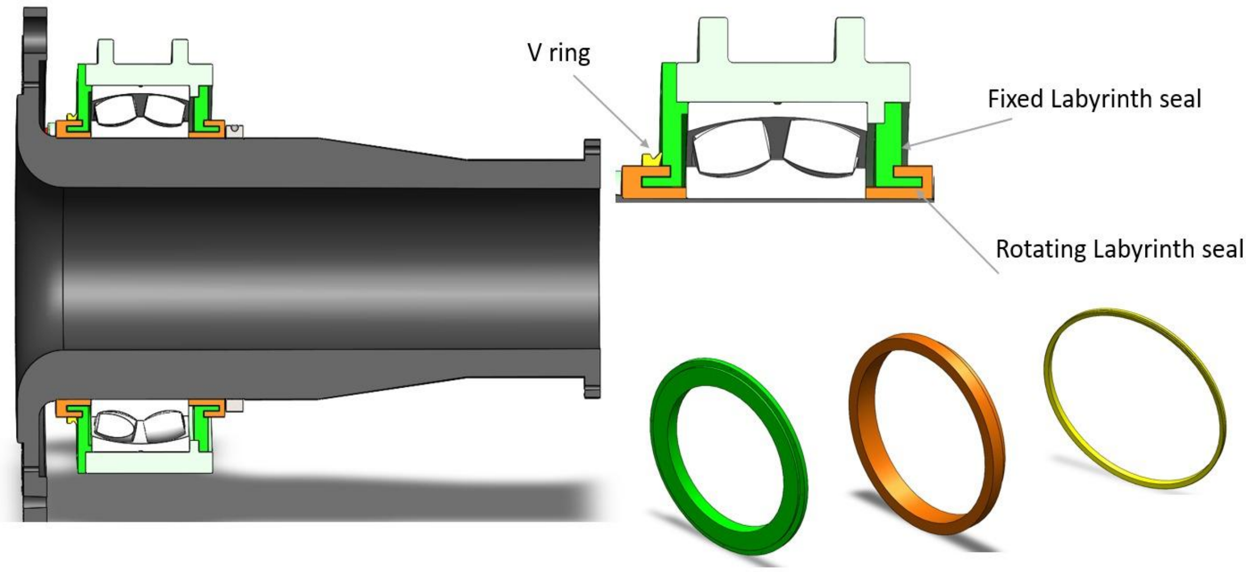


Figure 23: Labyrinth Sealing

9. Lock Nut

One of the mechanical tools that is used to hold the inner raceway bearing on the shaft. It functions as a stopper, preventing the inner ring bearing from moving with the help of the shoulder shaft on the other side

9.1 Design Work

The lock nut Tr 1600x 8 is designed and scaled based on the similar series lock nut available in market.

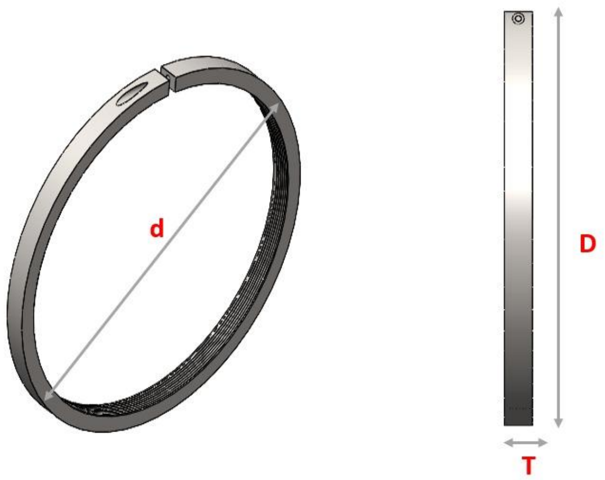


Figure 24: Lock Nut.

Thread	Outer diameter	Width	Gap width	Thread for mounting lever	Material
G	d3	B	b	G1	Plain Steel (1.0503(C45))
Tr 1600x 8	1750	110	40	M24	

Table 25: Locknut Specifications

10. Rotor Lock

A rotor lock is a mechanical component used to hold the rotor shaft in place during high wind conditions, allowing the rotor to be safely stopped during installation, maintenance, or servicing. The rotor lock disc is mounted on the low-speed side of the shaft, next to the rotor shaft flange.

The integrated lock disc is a critical structural component in a 5 MW wind turbine, acting as a high-strength interface between the main shaft and the gearbox to transmit immense torque through a flanged connection. Manufactured from cast iron to withstand extreme dynamic loads over a 20-year lifespan.

10.1 Rotor Lock Disk

A rotor lock is a mechanical component used to hold the rotor shaft in place during high wind conditions, allowing the rotor to be safely stopped during installation, maintenance, or servicing. The rotor lock disc is mounted on the low-speed side of the shaft, next to the rotor shaft flange.

10.1.1 Design of Rotor Lock Disc

In light of earlier reports, the rotor lock disc's size has been adjusted. Taking into account the correct horizontal and vertical blade locations, as well as the fact that we are employing two lock pins, we have chosen to use a lock disc with twelve holes. Below are specific measurements and the 3D design.

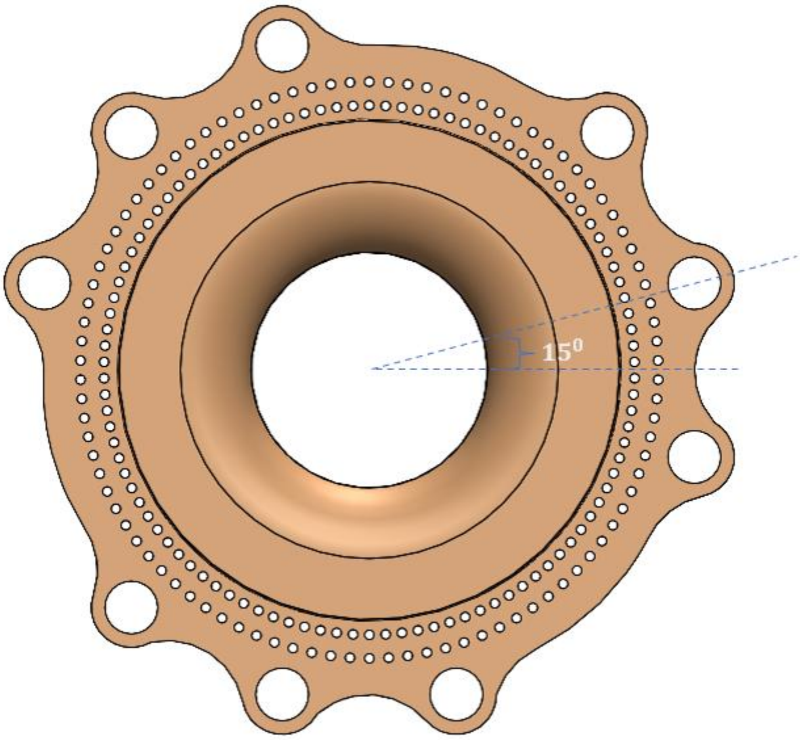


Figure 25: Integrated Rotor Lock Disc

10.1.2 Specifications of Rotor Lock Disc

S.No	Description	Value	Unit
1	Outer diameter of lock disc (D)	3190	mm
2	Width of lock disc (b)	190	mm
3	Diameter of outer pin holes (d_1)	223	mm
4	No. of pin holes in outer diameter	9	-
5	Material	Cast iron (EN-GJS-400-18-LT)	-

Table 26: Rotor Lock Disc Specifications.

The following table outlines the specific azimuthal positions for the rotor locking pins across all three blades at both 0^0 and 90^0 reference angles, ensuring precise locking.

Blade & Reference Angle	1-pin Angle	2-pin Angle
1st Blade (0∘)	15	165
2nd Blade (0∘)	255	45
3rd Blade (0∘)	135	285
1st Blade (90∘)	285	75
2nd Blade (90∘)	165	315
3rd Blade (90∘)	45	195

Table 27: Rotor Locking position.

10.2 Lock pin

Two number of bolts called as 'Lock Pin' is pushed against the hole of the lock disc in order to lock the rotation of the shaft. The lock pins are mounted on the legs of the bearing housing. They can be actuated hydraulically, electrically, and also manually. But here we have considered the Hydraulic mechanism for the actuation of the pins.

10.2.1 Design of Lock Pin

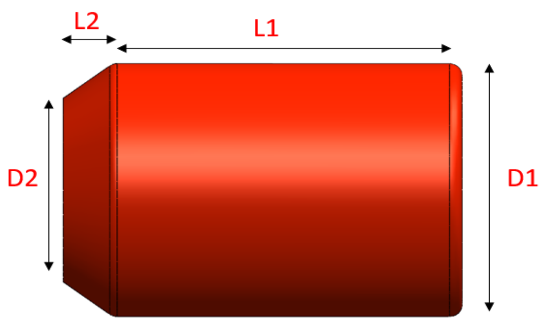


Figure 26: Lock Disc Pin

10.2.2 Specifications of lock pin

S.No	Description	Value	Unit
1	Diameter of lock pin (D1)	220	mm
2	Front-end diameter of lock pin (D2)	160	mm
3	Length of lock pin (L1)	280	mm
4	Front-end length of lock pin(L2)	40	mm
5	Material	42CrMO$_4$	-

Table 28: Details of Lock pin.

10.2.3 Rotor Lock Calculations.

This calculation is regarding the dimensioning of the Lock Pin. Considering the loads acting on the Lock disc, we have calculated the required diameter for the Lock pin in order to sustain the loads in concern to the Lock pin material. The material used for the Lock pin is 42CrMO4 **.**

Considering the Torsional moment given by the Loads & Dynamics team,

- Mechanical Power $P_{mech} = \frac{P}{\eta} = \frac{5}{0.95} = 5.263$ MW.
- $T_{rated} = \frac{P*60}{2*\pi*n} = \frac{5263 \text{ kW}*60}{2*\pi*10.7} = 4697$kNm
- Torsional moment Mt = 150% of rated torque = 7047 kNm

Force acting on the Lock disc,

$$F = \frac{M_t}{2r} = \frac{7047\ kNm}{2 * 1.4} = 2472\ kN$$

where, Lock pin hole circle radius r = 1425 mm

Bending Moment acting on the Lock pin, (Refer to Figure 32 below)

$M_{b,pin} = F \times L$

$M_{b,pin}$ = 2472 kN × 85 mm = 210196 kN.mm

In the above Calculation, L = $\frac{Thickness}{2} + 10\ mm$ clearance

The thickness of the disk = 150 mm

The disk hole = 189 mm

Pressure acting area of the hole, A=189*150=28350 mm^2

pressure acting on the hole of the disk

$$P = \frac{F}{A} = \frac{2472\ kN}{28350\ \mathrm{m}m^2}$$

P = 86 N/ mm^2

Allowable bending stress of the material 42CrMO4 = 0.66fy

Where Yield strength of the material 42CrMO4 , fy = 500 MPa or 500 N/mm^2 Allowable bending stress of the material 42CrMO4 = 0.66 × 500 N/mm^2 Allowable bending stress, σ = 330 N/mm^2

The allowable bending stress of 42CrMO4 is more than the Pressure acting on the hole of the disc.

Allowable bending stress, σ=$\frac{M_{b,pin}}{W}$

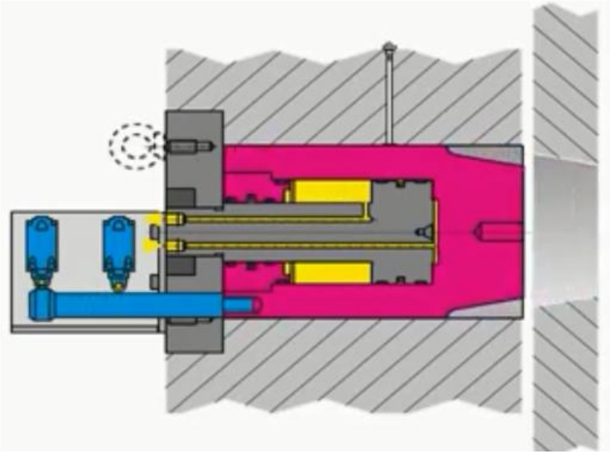
Where , section Modulus W =$\frac{\pi * d^3}{32}$

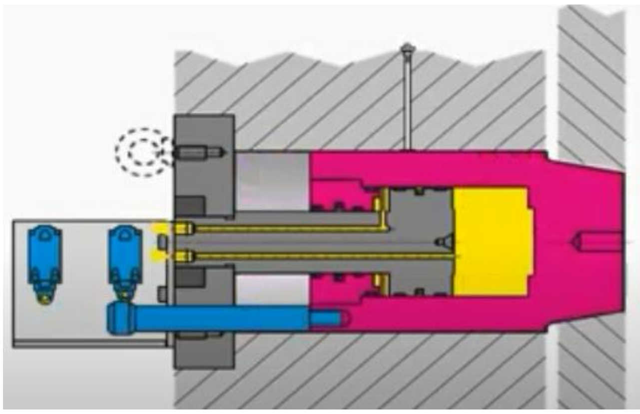
330 N/mm^2=$\frac{32*210196\ kN.mm}{\pi * d^3}$

The diameter of the lock pin ,d=186 mm

10.3 Lock Pin Hydraulic Mechanism

Given the safety precautions, a hydraulic design with position monitoring may be better. The locking mechanism of the rotor lock is hydraulic. Within a temperature range of -30°C to +70°C, the double-acting hydraulic cylinder reliably secures the rotor blade of wind power plants up to 6.5 MW by producing the bolt's retracting and expanding action.





10.4 Rotor Lock Sensors

The desired position of the blade or rotor shaft is achieved by alternately activating the start and stop movements of the rotor shaft in short, controlled stages after the rotor speed has been slowed down with the aid of rotor brakes. Once the desired position has been achieved, the lock pins are engaged. The sensors determine the desired position for the rotor locks to be activated. As seen in the examples below, this lock sensor can be installed on a frame or the machine bed to detect the holes in the lock disc going through it [15].

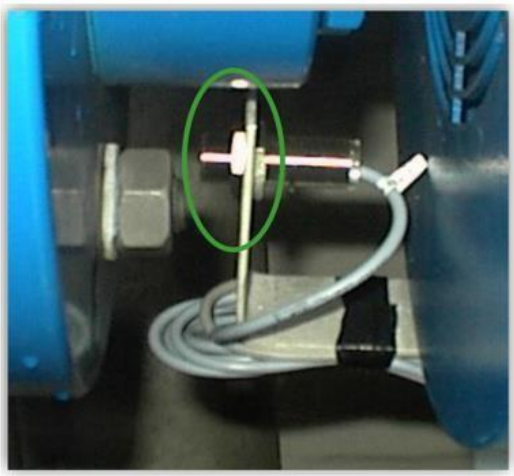


Figure 15: Rotor Lock Sensors (Examples)

The rotor lock's lock position can be detected with inductive sensors like the IIS246. In order to actuate the lock pin from an unlocked to a locked position hydraulically, this sensor determines whether the lock position has been reached [16].

Lock Position sensor IIS246

- Inductive sensor M39 x 1.5 / L = 60 mm.
- Sensing range 15 mm flush mountable.

- Ambient temperature -40 to 85 °C.
- Switching frequency 100 Hz.

11. Final Assembly

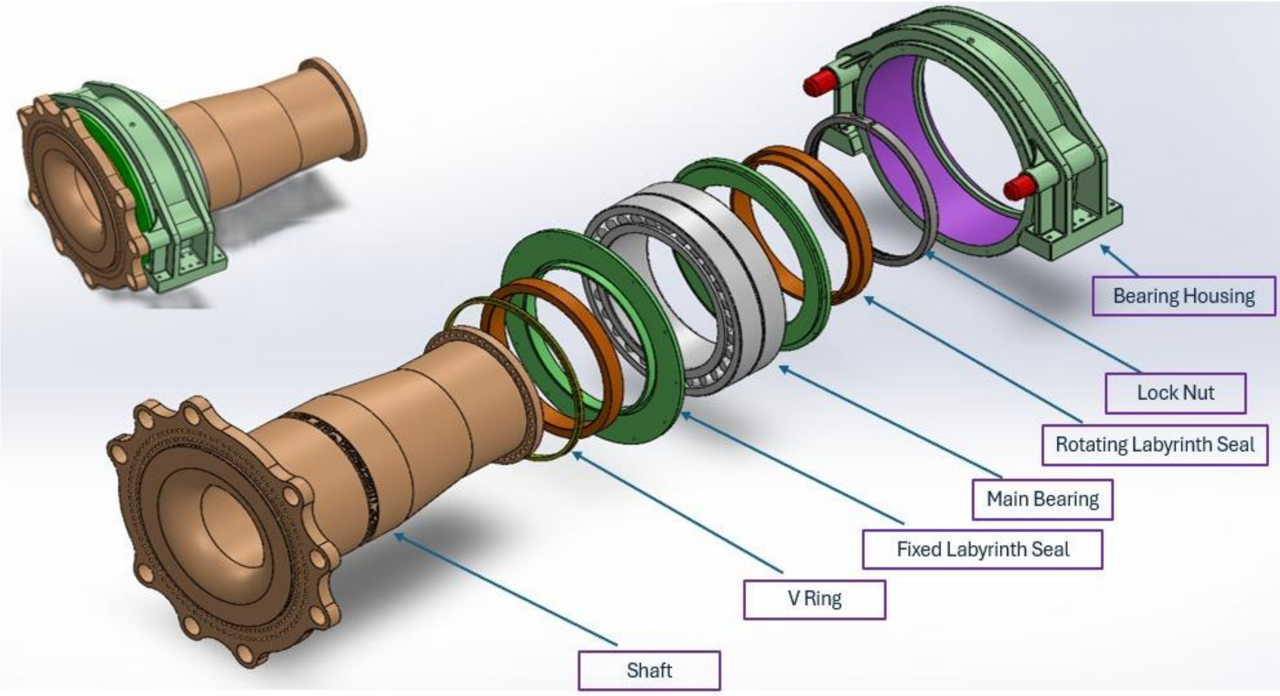


Figure 16: Assembly CAD Model

13. Used Software

Each team member is responsible for individual component calculations, design, and CAD drawings.

- Microsoft Word
- Microsoft Excel
- Microsoft PowerPoint
- Microsoft Teams
- SolidWorks
- Github

14. Open Issues

- Due to the short time, FEM analysis couldn't be done using ANSYS for fatigue simulations because we didn't have any Fatigue info from the loads team

14.1 Shaft Dimensioning

- The shaft dimensions can be more optimized as the shaft Ultimate strength is way higher than the Ultimate stress caused due to the Loop4 loads provided by the Loads team.

14.3 Suppliers

- Tried to contact the suppliers in Syria for casti ron shaft & bearing housing manufacture but we didn't receive any response from them.
- Suppliers for Lock sensors are limited.

15. Weight and Costs

The following table represents the Weights & Costs (as per 2026) of the main components used in the Rotor Bearing System.

Component	Material	Mass (Kg)	Cost per Kg(Euro)	Total cost(Euro)
Main Bearing Housing	Cast iron (EN-GJS-400-18-LT	9815	2	19630
Shaft	EN-GJS-600-3 (Cast Iron)	31184	2,5	77960
Bearing	Steel (100Cr6)	6701	6	40224
Lock nut	Plain steel(1.0503(C45))	333	2	666
Lock Pin(*2)	30 NiC rMo16-6	92	4	736
labyrinth seal fixed upside	PTFE(general)	408	12	4896

labyrinth seal fixed downside	PTFE(general)	335	12	4020
Labyrinth seal rotating upside	PTFE(general)	199	12	2388
Labyrinth seal rotating down	PTFE(general)	178	12	2136
V ring	Rubber(NBR)	153	4	612
TOTAL(from Assembly)		49398		

Table 29: Details of weight & cost of the main components

Bolt Specification	Quantity	Single-Piece Retail Price (Average Estimate)	Total Retail Cost (Estimated)
M36 × 250(DIN 933)	196	€27,29	5348,84
M48 × 150	46	€51,59	2373,14
M20*100	24	8	192
M45*300	20	55	1100

Table 30: Details of Bolts & cost of the main components

Washer Specification	Quantity	Single-Piece Retail Price (Average Estimate)	Total Retail Cost (Estimated)
M36 (37*66*5)	196	1	196
M48 (50*92*8)	46	2.50	115
M20(21*37*3)	24	.50	12
M45(46*85*8)	20	2	40

Table 31: Details of Washers & cost of the main components

16. CO_2 emission and recyclability

The details in the following table represent the carbon footprint of the components.

S No.	Components	CO2 in Kg
1	Cast Iron Shaft	55477
2	Rotor Lock Disc	12774
3	Bearing Housing	52062
4	Lock Nut	6494
5	Bearings	23800
6	Rotor Lock Pin	48830
7	Sealing	9916

Table 32: Details of Carbon footprint

17. External Supporters

1. Michael Reugels
   PD&E EMEA, Roller Bearings Development and Engineering department
   Mail : michael.reugels@skf.com

2. Andreas Reich
   Produktbereichsleiter Werkzeugspann- und Wechseltechnik
   Head of QDC/QMC
   Römheld Gmb
   Mail : a.reich@hilma.de

18. Teamwork

Chapter	Responsibility
Introduction	Jill Sadariya
Competition Analysis	Team
Relevant Standard & Guidelines	Team
Shaft Design & Fatigue Calculation	Divyesh Mistry and Venkata Sreekanth .K
Main Rotor Bearings	Sreehari Padachery and Jill Sadariya
Bearing Housing	Sreehari Padachery
Lock Nut	Jill Sadariya
Rotor Lock Disc & lock pin	Divyesh Mistry and Jill Sadariya
Sealings	Sreehari Padachery
Flange design & calculation	Team
Calculations	Team
Weights & Costs	Venkata Sreekanth .K
Carbon footprint	Venkata Sreekanth .K
Open Issues	Team
Teamwork	Team
Lessons Learned	Team
Conclusion	Divyesh Mistry
Annex	Divyesh Mistry and Sreehari Padachery

Table 33: Team work

19. Lessons Learned

The use of theoretical frameworks in this assignment really helped the team understand the topic better. It was that the success of the project depended not only on technical expertise but also noted on the use of effective communication protocols. The team was able to organize complicated data and finish the investigation through ongoing collaboration. Despite initial limitations in practical expertise, all deliverables were completed, establishing a foundation of technical competencies applicable to future professional environments.

20. Conclusion

Wind turbines operating in extreme environmental conditions require robust structural integrity. The Optimus Syria rotor bearing assembly was designed to facilitate critical torque transmission to the gearbox while effectively distributing mechanical loads from the hub to the machine bedplate. Given the complexity of the mechanical sub-systems, strict adherence to safety design protocols was maintained throughout the component selection process. Although procurement challenges arose due to limited supplier data, these were mitigated through technical consultation with academic advisors. Ultimately, the project successfully met its sustainability objectives, optimizing both the Levelized Cost of Energy (LCOE) and carbon dioxide CO_2 emission metrics.

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23 . Annex:

Annex A: Project Contract

Annex B: Technical Drawings

Annex A

SUB-TEAM PROJECT ORDER CONTRACT

Project Name: Optimus Syria
Sub-Team: Rotor Bearing System
Client: Bakhtyar Karim (Head of Project)
Date: 13.07.2025
Project Number: 01.2025
Team Leader: Divyesh Sathishkumar Mistry

Problem Description

The Rotor Bearing System sub-team is responsible for selecting and dimensioning the main bearing(s) that support the wind turbine rotor. This includes choosing between single or double bearing configurations, determining appropriate bearing types (e.g., spherical, tapered, cylindrical), and analyzing bearing loads, life expectancy, and lubrication. Close interaction with the Rotor Hub, Shaft, and Loads teams is required for accurate design.

Goals

- Define rotor bearing layout and select bearing configuration
- Choose suitable bearing types and justify the selection
- Calculate bearing loads based on rotor and shaft system
- Estimate lifetime and lubrication requirements
- Ensure mechanical compatibility and proper integration

Deliverables

- Rotor bearing configuration concept (single/double bearing)
- Bearing type selection with reasoning
- Load analysis and life estimation calculations
- Integration drawing or layout sketch
- Contributions to the final report and final presentation

Milestones & Deadlines

Milestone: Initial preparations
Deadline: 24th Aug – 7th Sep 2025
Comments: Research bearing types and reference turbines

Milestone: Concept freeze

Deadline: 13th Sep 2025
Comments: Decide layout and number of bearings

Milestone: Load principle measurements
Deadline: 7th Oct 2025
Comments: Collect loading data from Loads and Structure teams

Milestone: Interface freeze
Deadline: 28th Oct 2025
Comments: Confirm interface with shaft, hub, and nacelle

Milestone: Input freeze for simulation
Deadline: 18th Nov 2025
Comments: Provide bearing force/moment characteristics

Milestone: Design freeze
Deadline: 9th Dec 2025
Comments: Final bearing model with dimensions and materials

Milestone: Final Report
Deadline: Week 3/2026 (27–28 Jan)
Comments: Submit bearing design and integration concept

Milestone: Final Presentation
Deadline: Week 4/2026 (30–Jan)
Comments: Present bearing design choices and analysis

Team Leader Duties

• Delegate research and calculation tasks to team members
• Align interfaces with Shaft, Hub, and Nacelle design teams
• Perform or verify bearing load and lifetime calculations
• Maintain transparent documentation and technical notes
• Communicate regularly with Head of Project
• Ensure all deliverables meet deadlines and design expectations

---

Restrictions

[To be completed by the team leader: List any specific technical, academic, or organizational restrictions that apply to your team.]

Risk Management

Risk: Inappropriate bearing type or configuration selected
Action: Benchmark similar turbines and discuss with tutors

Risk: Overestimated bearing life or strength
Action: Use safety factors and perform sensitivity analysis

Risk: Integration issues with rotor shaft or nacelle
Action: Check interface dimensions and assembly feasibility early

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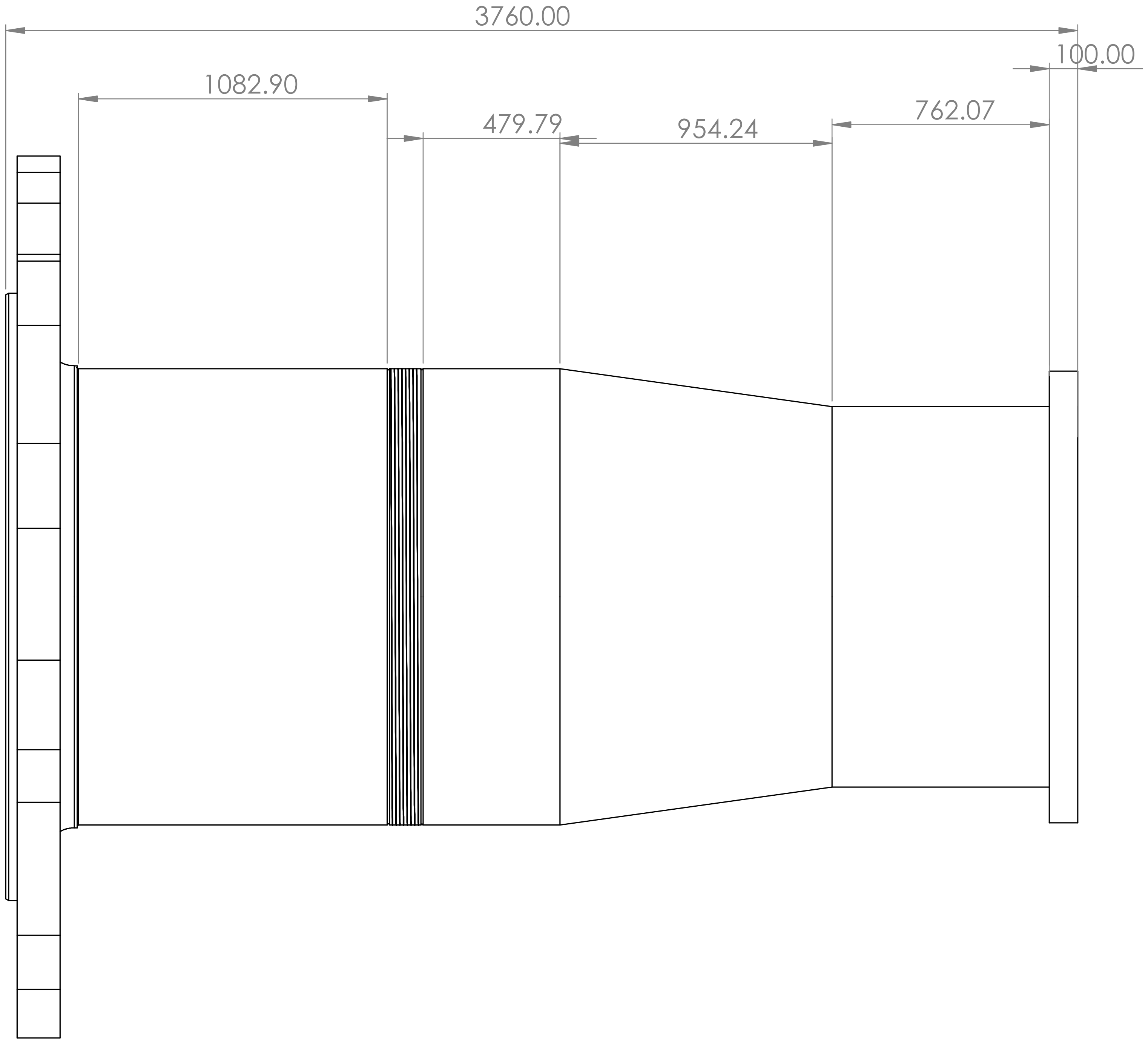
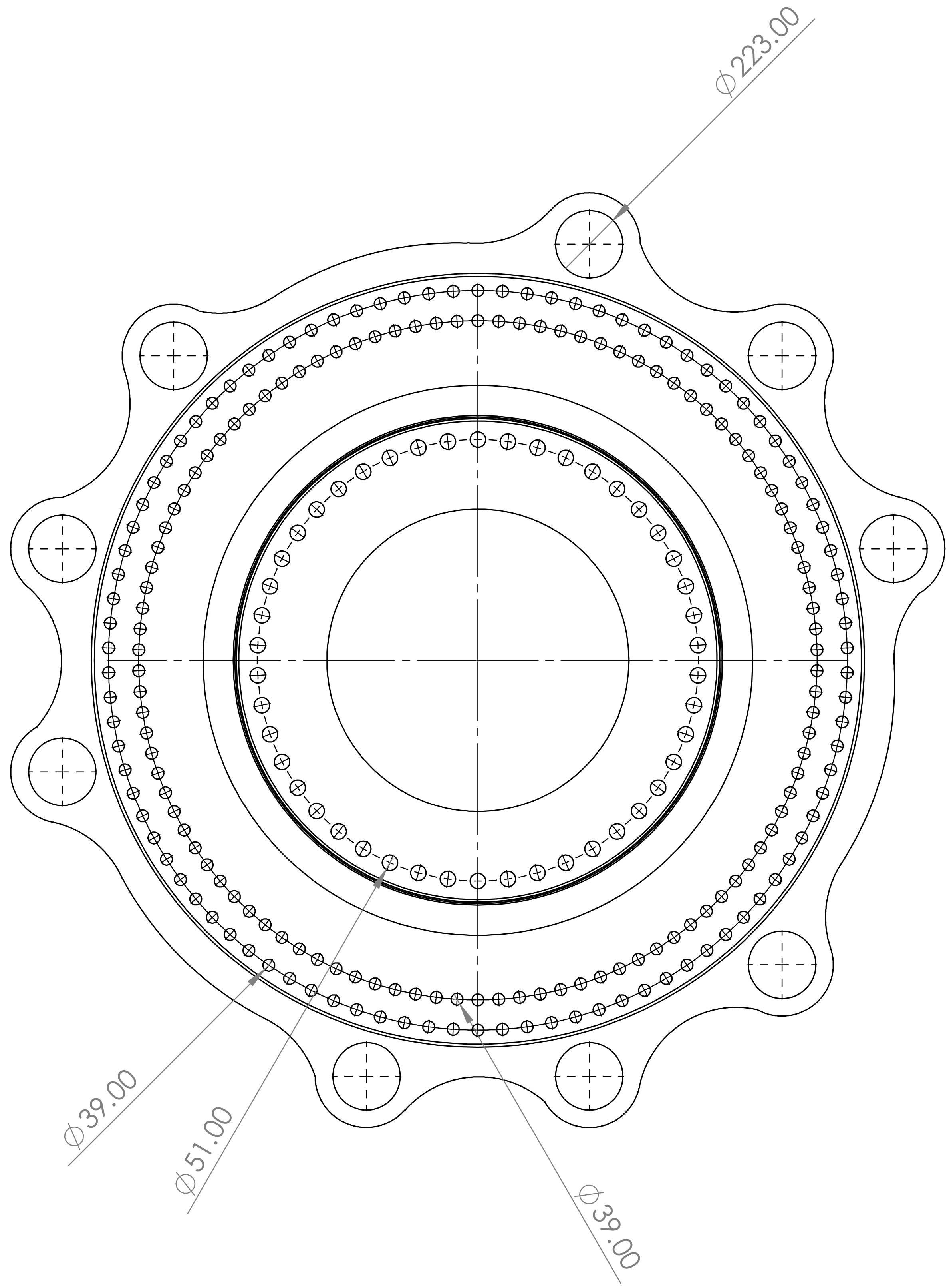
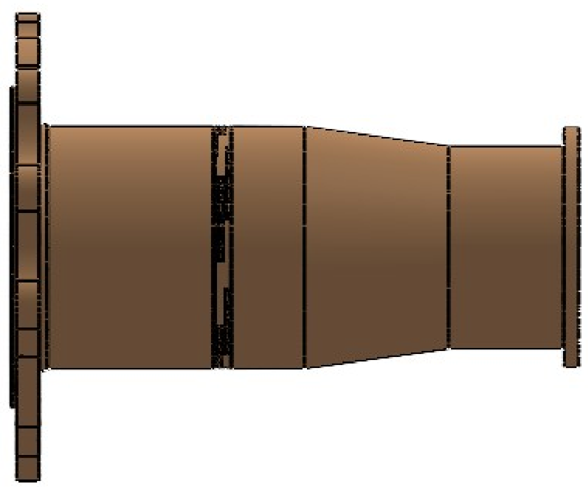
Signatures

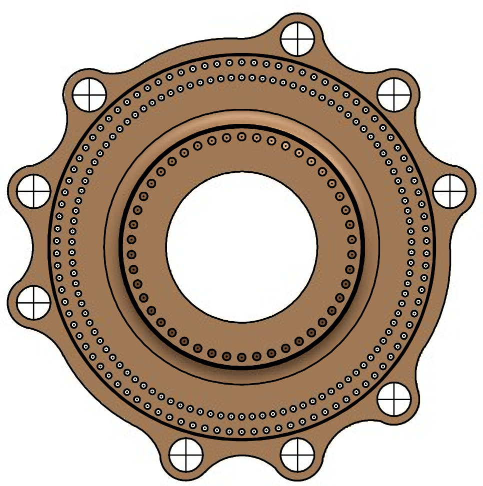
Role: Head of Project
Name: Bakhtyar Karim
Signature: ____________________

Role: Sub-Team Leader
Name: Divyesh Sathishkumar Mistry
Signature: ____________________









UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN MILLIMETERS
SURFACE FINISH:
TOLERANCES:
LINEAR:
ANGULAR:

FINISH:

DEBURR AND BREAK SHARP EDGES

DO NOT SCALE DRAWING

REVISION

	NAME	SIGNATURE	DATE
DRAWN			
CHK'D			
APPV'D			
MFG			
Q.A			

TITLE:

Optimus Syria

MATERIAL:
EN-GJS-400-18-LT

DWG NO.

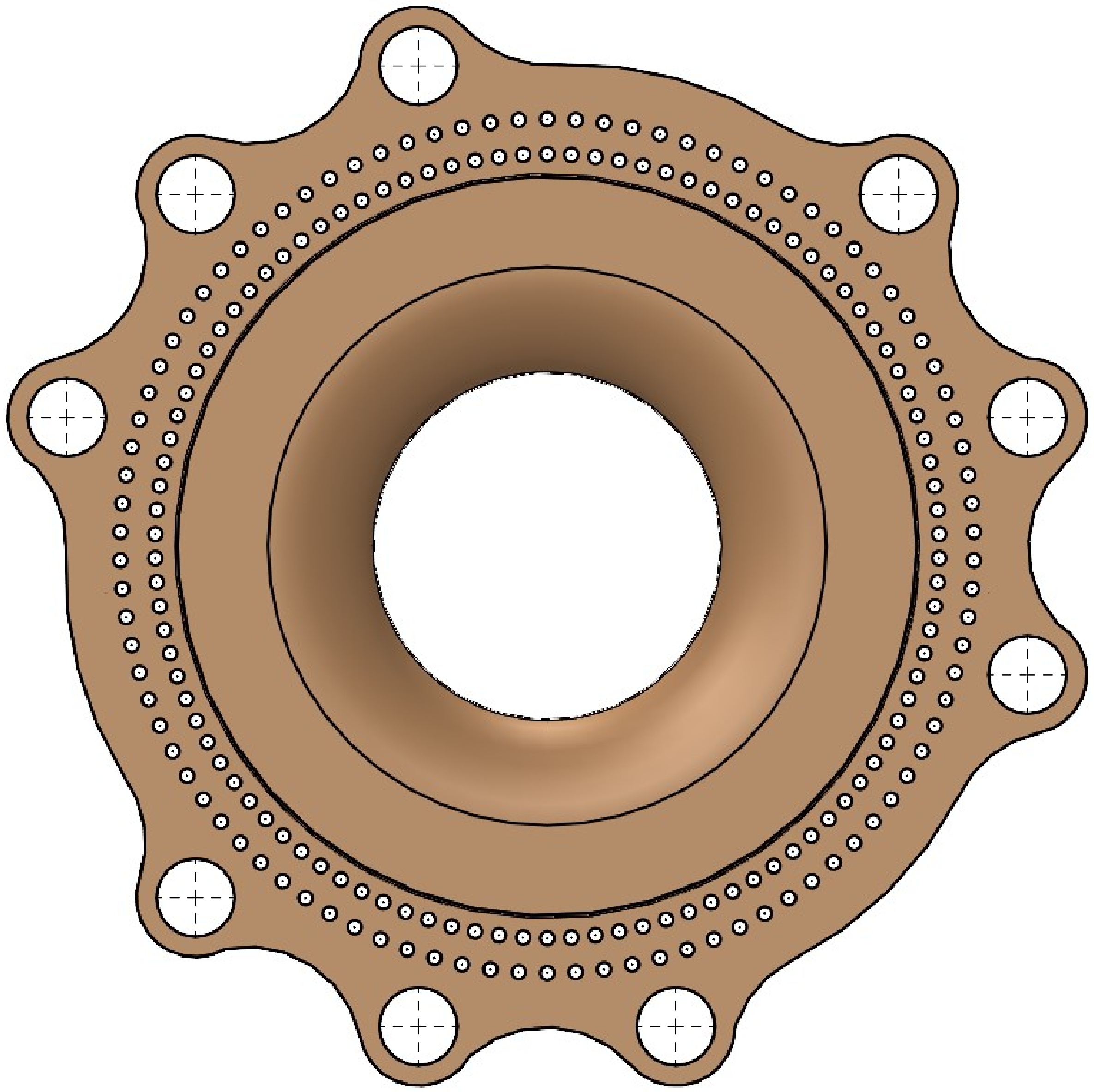
Rotor Shaft

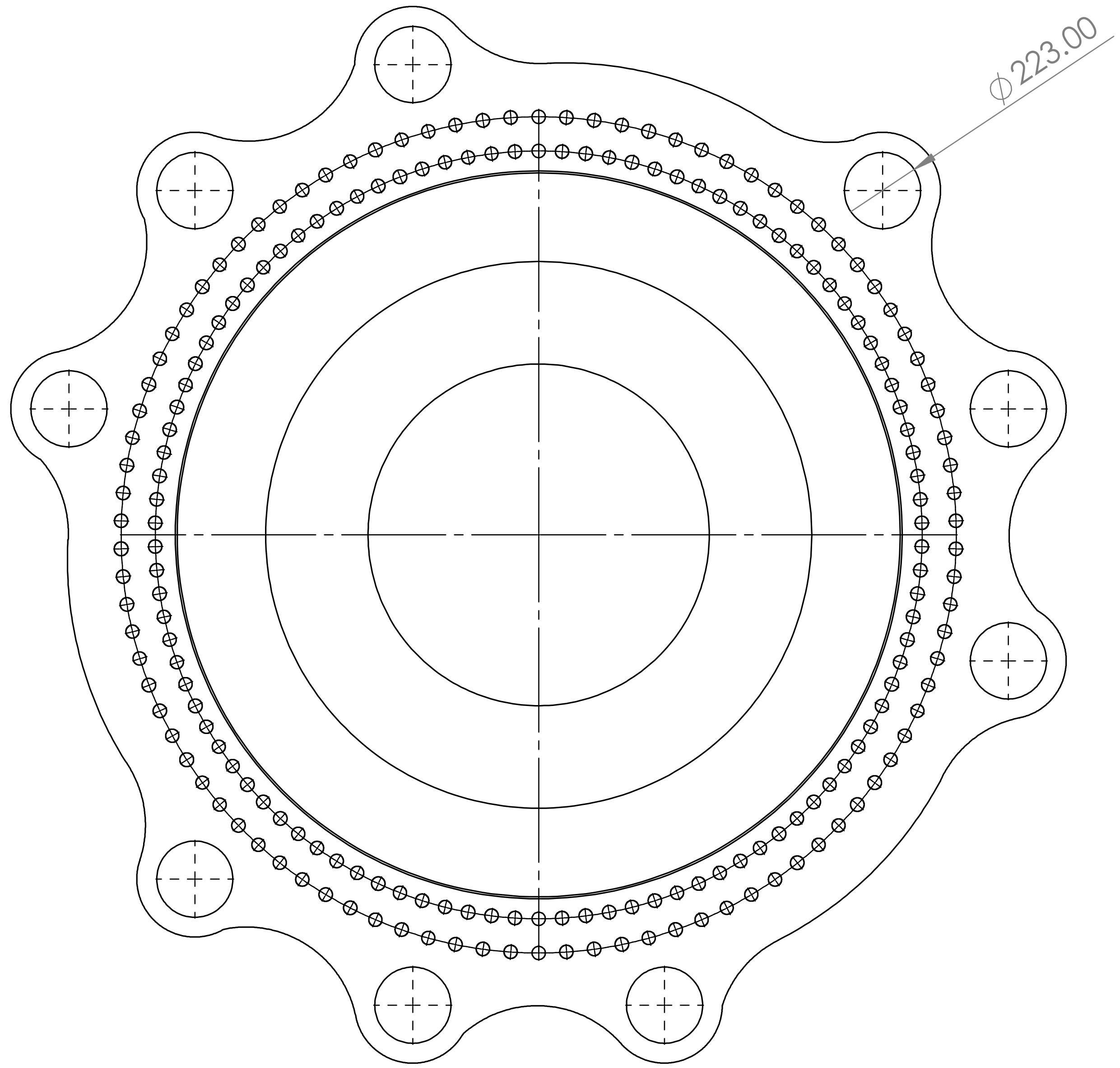
A3

WEIGHT: 31184 Kg

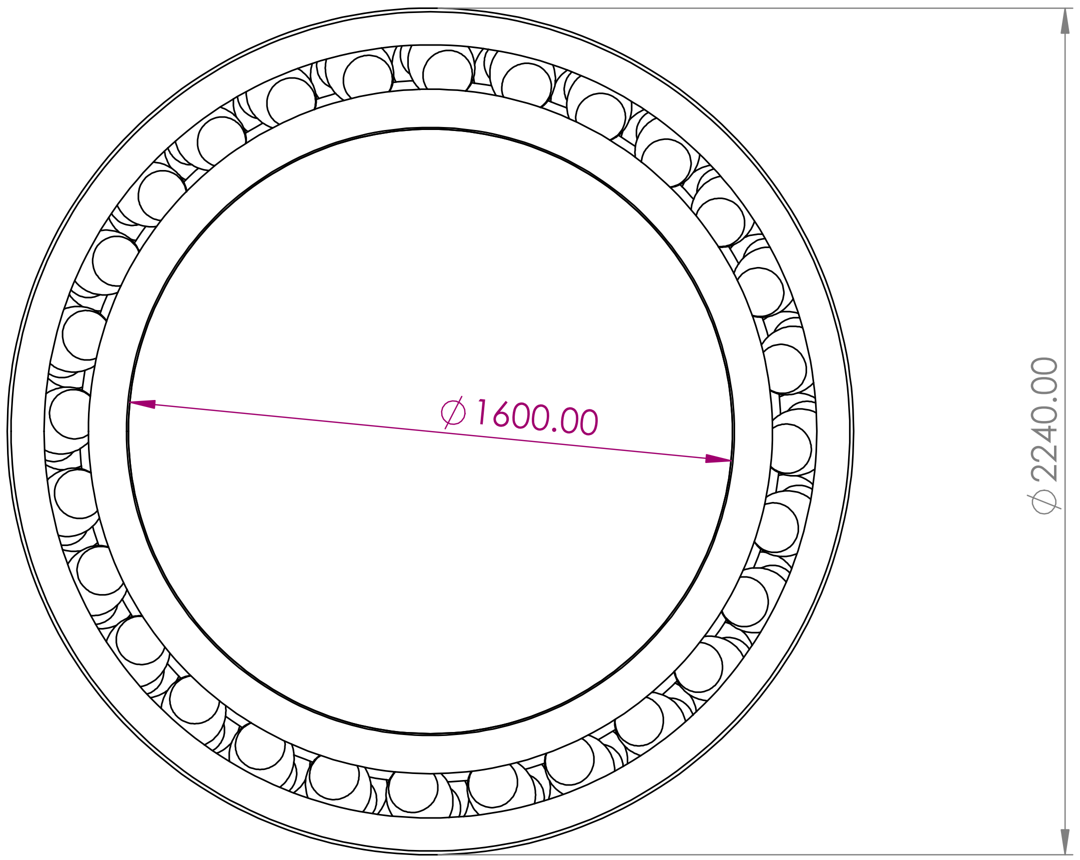
SCALE:1:100

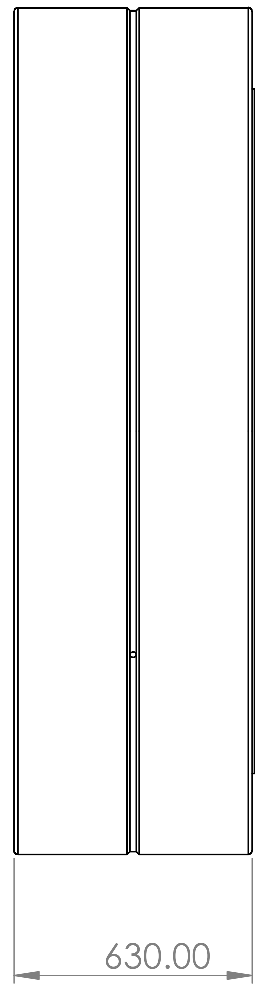
SHEET 1 OF 1



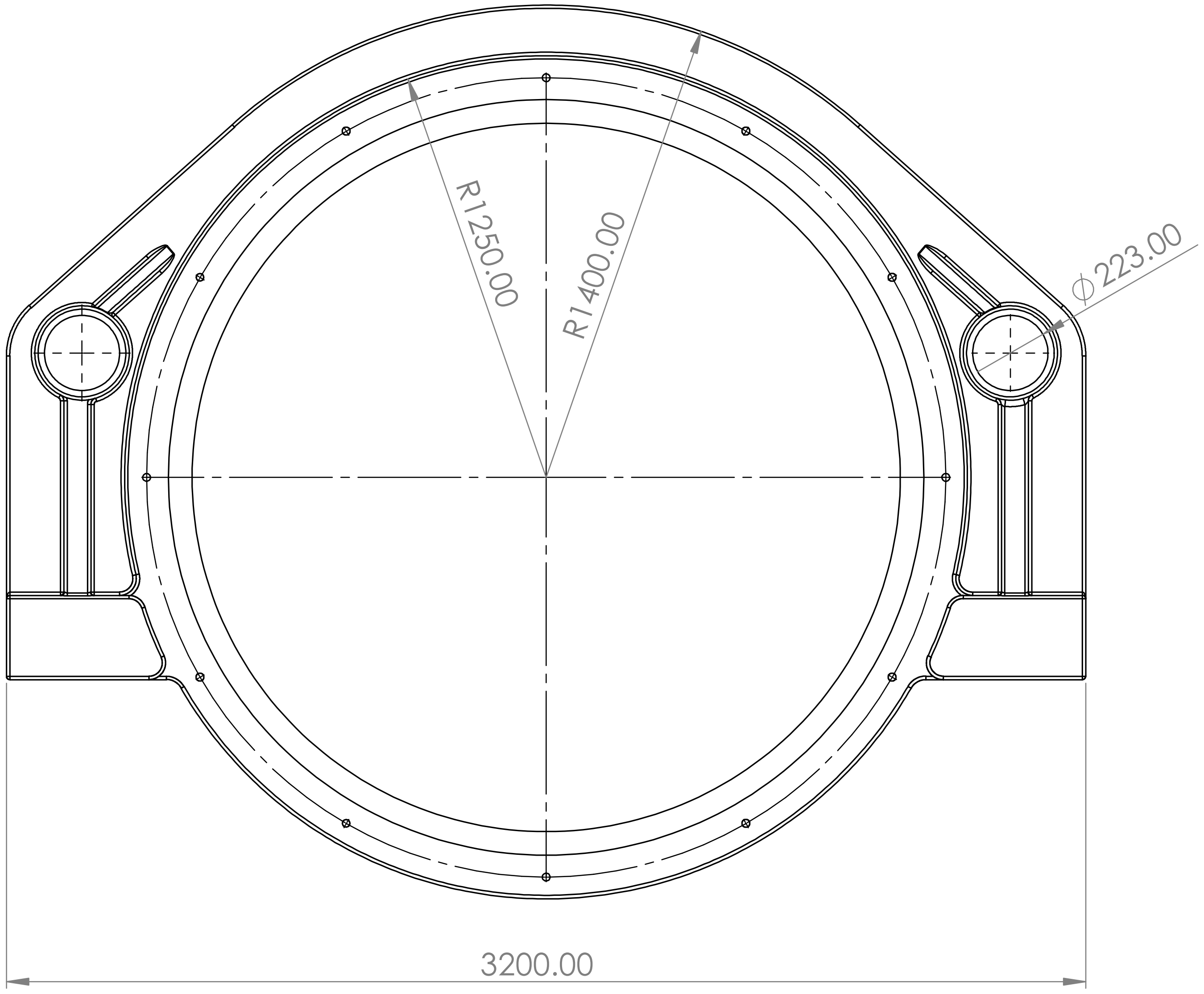


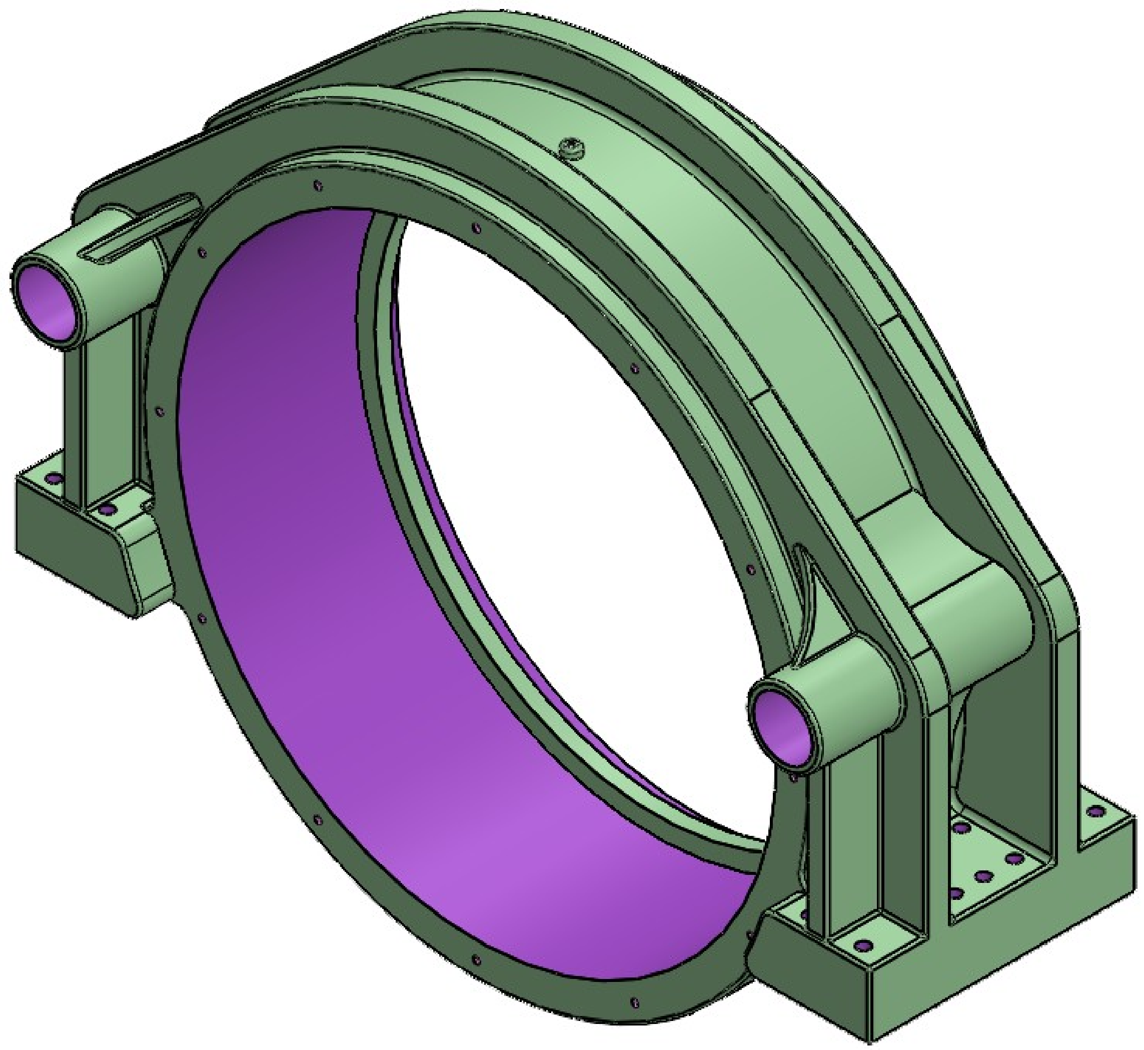
UNLESS OTHERWISE SPECIFIED: DIMENSIONS ARE IN MILLIMETERS SURFACE FINISH: TOLERANCES: LINEAR: ANGULAR:	FINISH:	DEBURR AND BREAK SHARP EDGES	DO NOT SCALE DRAWING	REVISION
	NAME	SIGNATURE	DATE	TITLE: Optimus Syria
DRAWN				
CHK'D				
APPV'D				
MFG				
Q.A			MATERIAL: EN-GJS-400-18-LT	DWG NO. Rotor Lock A3
			WEIGHT:	SCALE:1:50 SHEET 1 OF 1

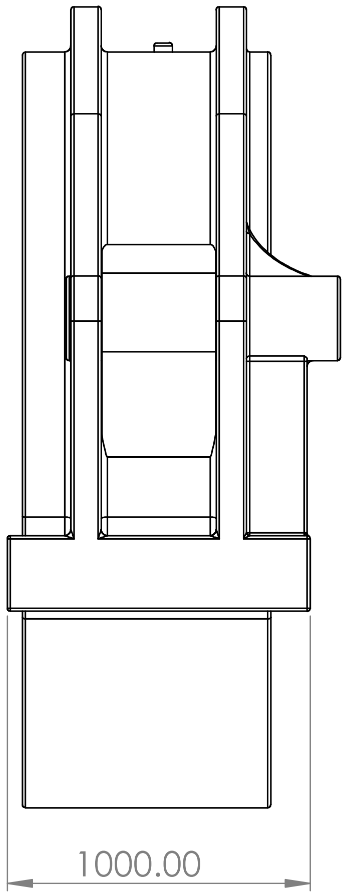




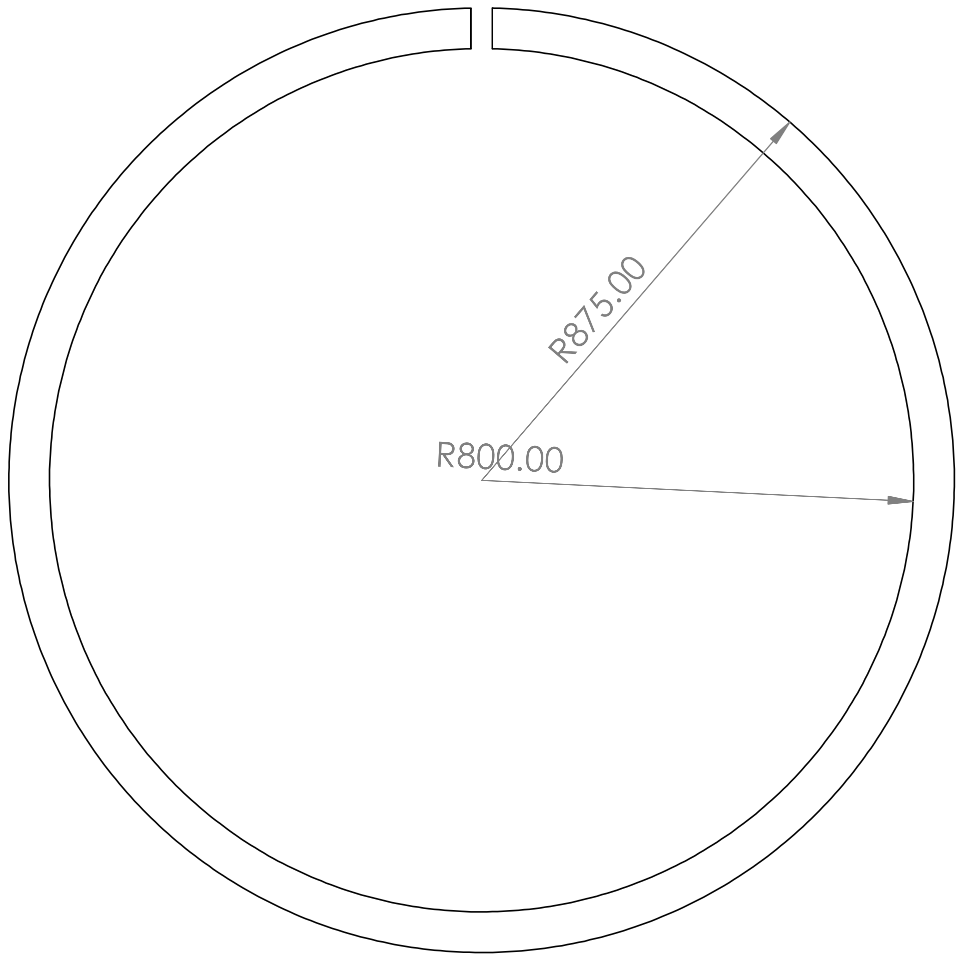
UNLESS OTHERWISE SPECIFIED: DIMENSIONS ARE IN MILLIMETERS SURFACE FINISH: TOLERANCES: LINEAR: ANGULAR:	FINISH:	DEBURR AND BREAK SHARP EDGES	DO NOT SCALE DRAWING	REVISION
	NAME	SIGNATURE	DATE	TITLE: Optimus Syria
DRAWN				
CHK'D				
APPV'D				
MFG				
Q.A			MATERIAL: Steel (100Cr6)	DWG NO. Spherical roller bearing — A3
			WEIGHT: 6702 Kg	SCALE:1:50 — SHEET 1 OF 1

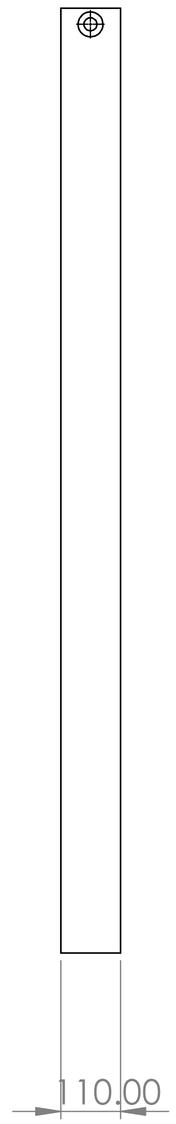




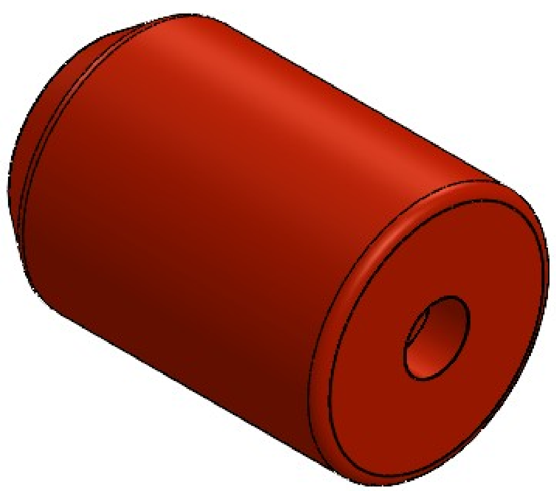


UNLESS OTHERWISE SPECIFIED: DIMENSIONS ARE IN MILLIMETERS SURFACE FINISH: TOLERANCES: LINEAR: ANGULAR:	FINISH:			DEBURR AND BREAK SHARP EDGES	DO NOT SCALE DRAWING	REVISION	
	NAME	SIGNATURE	DATE		TITLE: Optimus Syria		
DRAWN							
CHK'D							
APPV'D							
MFG							
Q.A				MATERIAL: EN-GJS-400-18-LT	DWG NO. Bearing Housing		A3
				WEIGHT: 9815 Kg	SCALE:1:50	SHEET 1 OF 1	

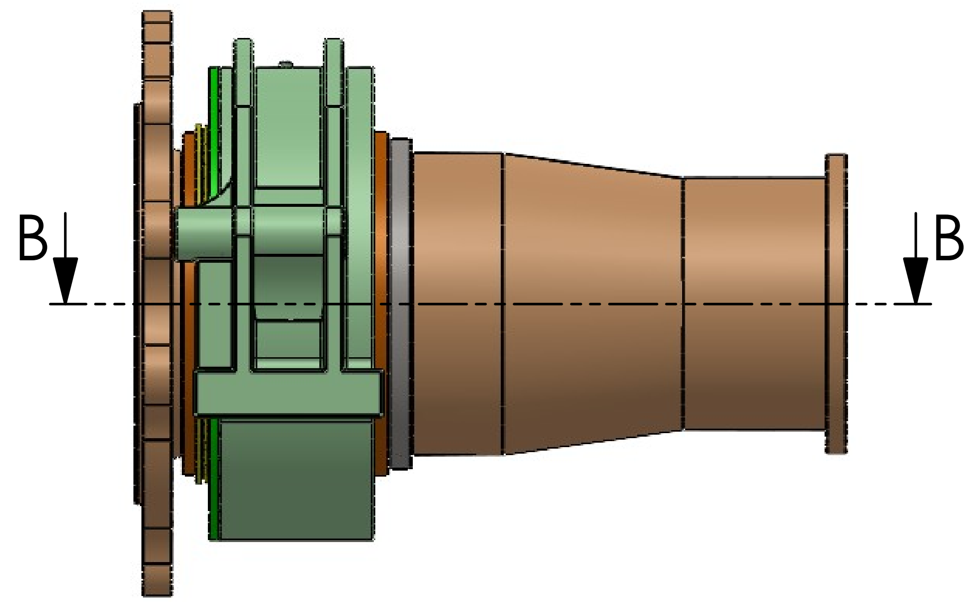


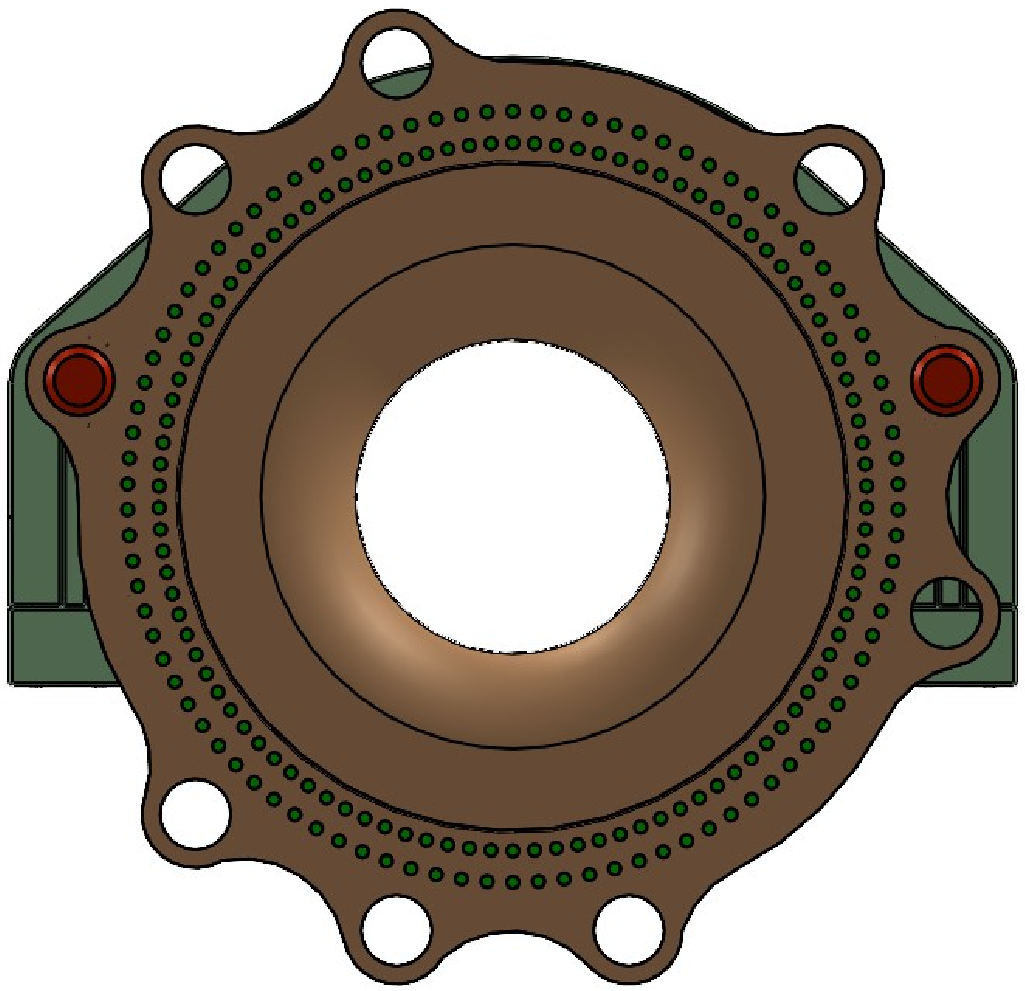


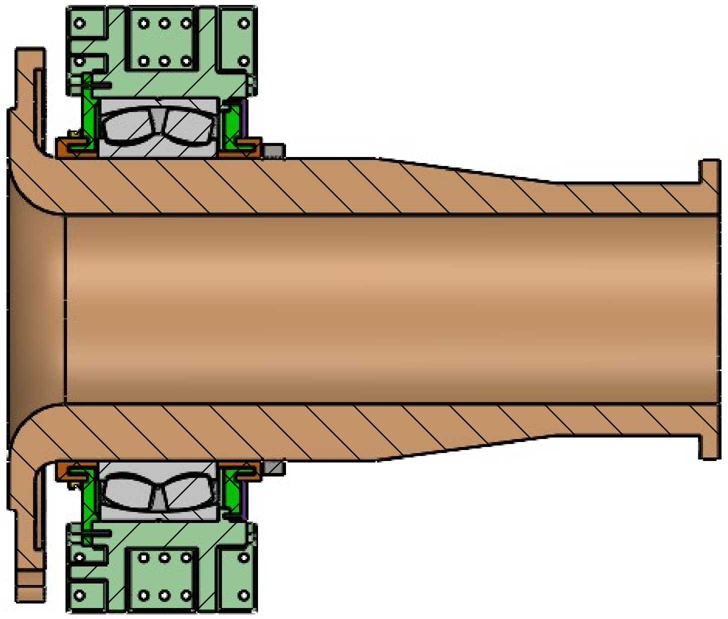
UNLESS OTHERWISE SPECIFIED: DIMENSIONS ARE IN MILLIMETERS SURFACE FINISH: TOLERANCES: LINEAR: ANGULAR:	FINISH:		DEBURR AND BREAK SHARP EDGES	DO NOT SCALE DRAWING	REVISION
	NAME	SIGNATURE	DATE	TITLE:	
DRAWN				Optimus Syria	
CHK'D					
APPV'D					
MFG					
Q.A			MATERIAL: Plain Steel (1.0503(C45))	DWG NO. Lock Nut	A3
			WEIGHT: 333 Kg	SCALE:1:20	SHEET 1 OF 1



UNLESS OTHERWISE SPECIFIED: DIMENSIONS ARE IN MILLIMETERS SURFACE FINISH: TOLERANCES: LINEAR: ANGULAR:	FINISH:		DEBURR AND BREAK SHARP EDGES	DO NOT SCALE DRAWING	REVISION
	NAME	SIGNATURE	DATE	TITLE:	
DRAWN				Optimus Syria	
CHK'D					
APPV'D					
MFG					
Q.A			MATERIAL: 30NiCrMo16-6	DWG NO. Lock Pin	A3
			WEIGHT:	SCALE:1:5	SHEET 1 OF 1







SECTION B-B

SCALE 1 : 40

UNLESS OTHERWISE SPECIFIED: DIMENSIONS ARE IN MILLIMETERS SURFACE FINISH: TOLERANCES: LINEAR: ANGULAR:	FINISH:	DEBURR AND BREAK SHARP EDGES	DO NOT SCALE DRAWING	REVISION

	NAME	SIGNATURE	DATE
DRAWN			
CHK'D			
APPV'D			
MFG			
Q.A			

TITLE: Optimus Syria

MATERIAL:

WEIGHT: 49353 Kg

DWG NO. Rotor Bearing System Assembly

A3

SCALE:1:50

SHEET 1 OF 1

Annex C: Brochures, data sheets

Table data from FKM Guidelines

Table 1.2.2 Constants $d_{eff,N,m}$, …, and a_{dm}, …, for cast iron materials

Values in the upper row refer to the tensile strength R_m,
Values in the lower row refer to the yield strength R_p.

Kinds of material	$d_{eff,N,m}$ $d_{eff,N,p}$ in mm	$a_{d,m}$ $a_{d,p}$
Cast steel DIN 1681	100 100	0,15 0,3
Heat treatable steel casting, DIN 17 205	300 ✧1 300	0,15 0,3
Heat treatable steel casting, q&t, DIN 17 205, types ✧2 No. 1, 3, 4	100 100	0,3 0,3
as above types ✧3 No. 2	200 200	0,15 0,3
as above types No. 5, 6, 8	200 200	0,15 0,3
as above types No. 7, 9	500 500	0,15 0,3
GGG DIN EN 1563	60 60	0,15 0,15
GT ✧4 DIN EN 1562	15 15	0,15 0,15

q&t= quenched and tempered

✧1 For GS-30 Mn 5 or GS-25 CrMo 4 there is $d_{eff,N,m}$ = 800 mm or 500 mm respectively, values $a_{d,m}$ and $a_{d,p}$ as given above.

✧2 Material types see Table 5.1.11.

✧3 Valid for strength level V I, for level V II deff,N,m = deff,N,p = 100 mm with values $a_{d,m}$ and $a_{d,p}$ as above.

✧4 The values for GT are needed for the assessment of the fatigue strength only.

Table 1.2.3 Effective diameter d_{eff}

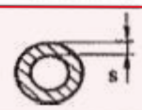
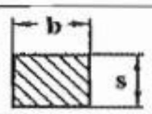
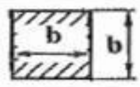
No.	Cross-sectional shape	d_{eff} Case 1	d_{eff} Case 2
1		d	d
2		2s	s
3		2s	s
4		$\frac{2b \cdot s}{b+s}$	s
5		b	b

Table 1.2.4 Anisotropy factor K_A

Steel:

R_m in MPa	up to 600	above 600 and up to 900	above 900 and up to 1200	above 1200
K_A	0,90	0,86	0,83	0,80

Table 2.2.1 Fatigue strength factor for completely reversed normal stress, $f_{W,\sigma}$, and shear stress, $f_{W,\tau}$ ✧1

Material group	fW,σ	fW,τ
Case hardening steel	0,40 ✧2	0,577 ✧2 ✧3
Stainless steel	0,40 ✧4	0,577
Forging steel	0,40 ✧4	0,577
Steel other than these	0,45	0,577
GS	0,34	0,577
GJS	0,34	0,65
GJM	0,30	0,75
GJL	0,34	1,0 ✧5
Wrought aluminum alloys	0,30 ✧6	0,577
Cast aluminum alloys	0,30 ✧6	0,75

✧1 $f_{W,\sigma}$ and $f_{W,\tau}$ are valid for a number of cycles of $N = 10^6$. $f_{W,\tau}$ equal to f_τ in Table 1.2.5.

✧2 Blank hardened. The influence of carburization on the component fatigue limit for completely reversed stress shall be taken into consideration based on the surface treatment factor K_V, Chapter 2.3.3.

✧3 $0{,}577 = 1/\sqrt{3}$, according to the v. Mises criterion.

✧4 Preliminary values.

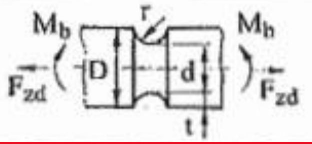
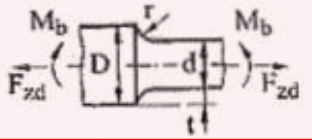
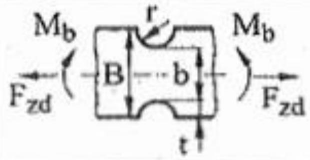
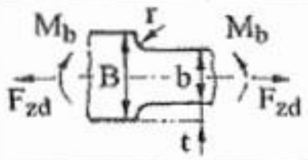
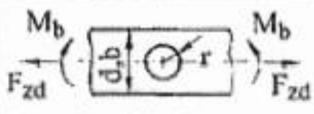
✧5 According to the normal stress criterion.

✧6 $f_{W,\sigma}$ does not correspond with the endurance limit for $N = \infty$ here.

Table 2.3.1 Constants a_G and b_G

Material group	Stainless steel	Other kinds of steel	GS	GJS	GJM	GJL
a_G	0,40	0,50	0,25	0,05	–0,05	–0,05
b_G	2400	2700	2000	3200	3200	3200

Table 2.3.5 Related stress gradient $G_\sigma(r)$ and $G_\tau(r)$ for simple component designs ✧1

Component design	$\overline{G}_\sigma(r)$ ✧2 ✧3	$\overline{G}_\tau(r)$ ✧4
	$\frac{2}{r}\cdot(1+\varphi)$	$\frac{1}{r}$
	$\frac{2,3}{r}\cdot(1+\varphi)$	$\frac{1,15}{r}$
	$\frac{2}{r}\cdot(1+\varphi)$	-
	$\frac{2,3}{r}\cdot(1+\varphi)$	-
 Round or flat bar	$\frac{2,3}{r}$	-

✧1 $r > 0$. For round bars, the equations are approximately valid in the case of a longitudinal hole as well.

✧2 $\varphi = 0$ for $t/d > 0,25$ or $t/b > 0,25$,
$\varphi = 1/\left(4\cdot\sqrt{t/r}+2\right)$ for $t/d \leq 0,25$ or $t/b \leq 0,25$.

✧3 The related stress gradient $G_\sigma(r)$ applies to axial and bending stresses; the difference is taken into consideration by the K_t-K_f ratio $n_\sigma(d)$ in Eqs. (2.3.2) and (2.3.3).

✧4 The related stress gradient $G_{\tau}(r)$ applies to shear and torsional stresses; the difference is taken into consideration by the K_t-K_f ratio $n_\tau(d)$ in Eqs. (2.3.2) and (2.3.3).

Table 2.3.6 Constants $a_{R,\sigma}$ and minimum tensile strength, $R_{m,N,min}$, for the material group considered

Material group	Steel	GS	GJS	GJM	GJL
$a_{R,\sigma}$	0,22	0,20	0,16	0,12	0,06
$R_{m,N,min}$ in MPa	400	400	400	350	100

Table 2.4.1 Constants a_M and b_M

Material group	Steel ✧1	GS	GJS	GJM	GJL
a_M	0,35	0,35	0,35	0,35	0
b_M	-0,1	0,05	0,08	0,13	0,5

Table 4.4.3 Number of cycles at knee point, slope exponents and values $f_{II,\sigma}$, $f_{II,\tau}$ ✧1 of the component constant amplitude S-N curves

Normal stress

Component	$N_{D,\sigma}$	$N_{D,II\sigma}$	k_σ	$k_{II,\sigma}$	$f_{II,\sigma}$
Steel and cast iron materials (*model I S-N curve*)					
non-welded	10^6	-	5	-	-
welded	$5 \cdot 10^6$	-	3	-	-
Aluminum materials and austenitic steel (*model II S-N curve*)					
non-welded	10^6	10^8	5	15	0,74
welded	$5 \cdot 10^6$	-	3	-	-

✧1 $f_{II,\sigma}$, $f_{II,\tau}$ from N_D, $N_{D,II}$ and k_{II} in accordance with Eq. (4.4.49)

Table 4.5.1 Material safety factors j_F for non-welded steel and for wrought aluminum alloys

j_F		Consequences of failure ✧1		
		severe	mean	moderate
Regular inspections ✧2	no	1,5	1,4	1,3
	yes	1,35	1,25	1,2

✧1 Severe consequences of failure: loss of human life.
Mean consequences of failure: loss of the entire structure, reduction by about 7,5 %.
Moderate consequences of failure: loss of secondary components; possibilities for load redistribution in statically indeterminate system, reduction by about 15 %.

✧2 Regular inspection in the sense of monitoring for early detection of damage: reduction by about 10 %.

Table 5.1.12 Mechanical properties in MPa for spheroidal graphit cast irons, after DIN EN 1563 (1997-08-00) or after DIN 1693 / 01 (1973-10-00) (namings given in brackets) ✧1.

Type of material	Material No.	$R_{m,N}$	$R_{p0,2,N}$ ✧2	A_5 ✧3	$\sigma_{W,zd,N}$	$\sigma_{Sch,zd,N}$	$\sigma_{W,b,N}$	$\tau_{W,s,N}$	$\tau_{W,t,N}$
EN-GJS-350-22-LT (GGG-35.3) EN-GJS-350-22-RT EN-GJS-350-22	EN-JS1015 (0.7033) EN-JS1014 EN-JS1010	350	220	22	120	100	160	75	110
EN-GJS-400-18-LT (GGG-40.3) EN-GJS-400-18-RT EN-GJS-400-18	EN-JS1025 (0.7043) EN-JS1024 EN-JS1020	400	240 250 250	18	135	110	185	90	120
EN-GJS-400-15 (GGG-40)	EN-JS1030 (0.7040)	400	250	15	135	110	185	90	120
EN-GJS-450-10	EN-JS1040	450	310	10	155	125	205	100	135
EN-GJS-500-7 (GGG-50)	EN-JS1050 (0.7050)	500	320	7	170	135	225	110	150
EN-GJS-600-3 (GGG-60)	EN-JS1060 (0.7060)	600	370	3	205	160	265	135	180
EN-GJS-700-2 (GGG-70)	EN-JS1070 (0.7070)	700	420	2	240	180	305	155	205
EN-GJS-800-2 (GGG-80)	EN-JS1080 (0.7080)	800	480	2	270	200	340	175	235
EN-GJS-900-2	EN-JS1090	900	600	2	305	220	380	200	260

✧ 1 Effective diameter $d_{eff,N}$ = 60 mm.

✧ 2 $R_{p0,2,N} / R_{m,N} < 0,75$ for all types of material listed.

✧ 3 Elongation in %. For non-ductile materials, $A_5 < 12,5\%$, the assessment of the static strength is to be carried out by using local stresses, Chapter 1.0, and all safety factors are to be increased by adding a value Δj , Eq. (2.5.2), … , see Chapters 2.5, 3.5 or 4.5 , respectively.

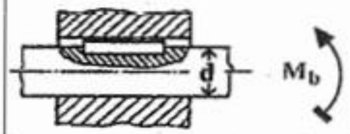
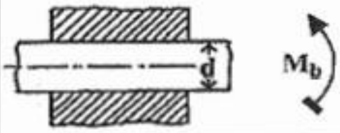
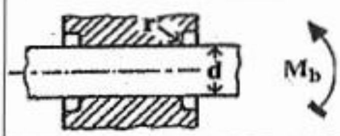
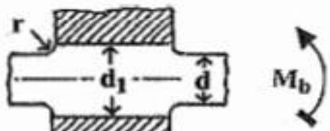
5.3.3.5 Shafts with press-fitted members

The fatigue notch factors for shafts with press-fitted (or shrink-fitted) members shall be determined based on Table 5.3.1 or Figure 5.3-11.

Fatigue notch factor based on Table 5.3.1

Table 5.3.1 Fatigue notch factor, $K_{f,b}(d_P)$, of shaft with press-fitted member for bending, according to *Tauscher* (Ref.: *Lehr*)

Notch radius of the test specimen $r_P = 0{,}06\ d_P$, test specimen diameter $d_P = 40$ mm. The bending moment is transmitted via the hub. The same fatigue notch factors are valid for fits with closer seat.

Rm in MPa								
400	500	600	700	800	900	1000	1100	1200
No. 1 H7/n6 interference fit.								
2,1	2,3	2,5	2,6	2,8	2,9	3,0	3,1	3,2
No. 2 H8/u8 interference fit.								
1,8	2,0	2,1	2,3	2,5	2,7	2,9	2,9	2,9
No. 3 H8/u8 interference fit.								
1,5	1,7	1,8	1,9	2,0	2,2	2,3	2,3	2,3
No. 4 H7/n6 interference fit.								
1,6	1,8	1,9	2,1	2,3	2,4	2,6	2,6	2,6
No. 5 H8/u8 interference fit. This design is not recommended!