HIghly advanced Probabilistic design and Enhanced Reliability methods for high-value, cost-efficient offshore WIND

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Definitions:	

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List of Abbreviations

SWT	Siemens Wind Turbine
EDF	Électricité de France
ISO	International Organization for Standardization
O&M	Operation and Maintenance
LDD	Load Duration Distribution
IEC	International Electrotechnical Commission
DEFL	Damage Equiavlent Fatigue Load
HSS	High-Speed Shaft
AC	Alternating Current
DC	Direct Current
ASTM	American Society for Testing and Materials

1. Executive Summary

The deliverable D5.1 of the Hiperwind project focused on formulating a component lifetime model of some of the wear components of a wind turbine to investigate the impact of the environmental conditions such as wind speed distribution, turbulence intensity distribution, wave conditions and the operation conditions of the connected electric grid. The intention of the deliverable is to describe the drivetrain of a Siemens Wind Power SWT- 2.3-93 wind turbine, which is used in the Teesside wind farm, investigated as part of the Hiperwind project. This public description of the drivetrain has been obtained by following second hand spare-part items posted on the web site of Spared In Motion. We were able to obtain a representative description of the main bearing of the SWT-2.3-93 turbine; and by combining this with an aeroelastic model of the SWT-2.3-93 turbine created by EDF, owner of the Teesside wind farm, we determined the loads of the Teesside wind turbines and used these as input for the main bearing life model. It was originally also planned to investigate the high-speed bearings inside the gearbox of the SWT-2.3-93 turbine, but it has not been possible to obtain the dimensions of the gearbox, whereby the focus was the main bearing. A conceptual analysis of the high-speed bearings is planned for a future deliverable within work package 5.

The lifemodel of the main bearing of the SWT-2.3-93 turbine has also been selected. Main bearings are the most expensive to replace, because their replacements in offshore applications require disassembling the turbine rotor from the nacelle using a jack-up vessel. Thus, the main bearing is anticipated to have the greatest impact on the operation and maintenance cost of the Teesside offshore wind farm. The lifemodel developed here is based on the ISO 281 “Rolling bearings — Dynamic load ratings and rating life” standard and extended to include a model for estimating the bearing temperature, viscosity changes and the impact of the cleanliness of the bearing grease. The environmental conditions of the Teesside wind farm are not publicly available and the analysis of the main bearing life has therefore been performed by estimating the main bearing loads assuming the SWT -2.3-93 turbine is exposed to different wind classes with different turbulence intensities. The estimated loads are the input to the lifetime model. This allow for an estimate of the expected lifetime of the main bearing given a specified wind class for the original turbine and then also in different wind classes, which can reveal whether some of the turbines are affected by wakes in the Teesside wind farm.

The lifetime analysis of the main bearing for the Teesside turbines with wind class IIA has shown that the main bearing life time is estimated to be 40 years, but if the actual turbulence intensity is reduced from the design level of $I_{ref} = 0.16$ to $I_{ref} = 0.10$, then a lifetime of the model is reduced to 22 years, which is actually lower than the design life time of 25 years of the farm. The lifetime of the model specifies the so-called modified L_{10} life of the main bearings, meaning 10 % of a bearing fleet has failed at the modified L_{10} life. Thus, a simple estimate of the number of failed main bearings after 22 years will be 10 % of the 27 turbines giving approximately 3 main bearing. This prediction can now be compared with the remaining operation of the Teesside wind farm.

There are a number of assumptions behind this estimate of the bearing life and these will be examined further in the deliverable D5.3 of the Hiperwind project and finally the estimated number of failed main bearings will be provided to the Hiperwind work package on the economical analysis of the operation and maintenance schedule.

2. Introduction

The development of the modelling framework for estimating the lifetime of offshore wind turbine drivetrain bearings is the focus of this report, considering the influence of the environmental conditions and operational strategy. The wind turbine bearing failures cause significant downtime. A proper model that can predict the bearing fatigue life accurately provides the following benefits:

(i) Accurate estimation of the bearing replacement-associated costs during a lifetime of a wind farm.

(ii) By understanding the impact of the loading and environmental conditions on the bearing fatigue life, the Operation and Maintenance (O&M) strategy can be optimized to improve both the reliability and the performance of wind farms.

(iii) Possibility to reduce the unscheduled maintenance.

The bearing fatigue life is generally quantified as a L_{10} life (Budynas & Nisbett, 2011) which gives a probable lifetime for a 90 % of a group identical bearing can survive. Even though, the L_{10} life model is well developed within bearing industry, the influence of the loading and environmental conditions faced by the wind turbine bearings on the L_{10} life has not been studied in details. Watanabe & Uchida (Watanabe & Uchida, 2015) studied the influence of bearing design parameters such as load factor, basic load rating and the wind shear on the L_{10} life of a main bearing of a wind turbine and suggested the idea of the life prolongation with the sector curtailment. Watanabe & Uchida used the fundamental principles to estimate the main bearing loads and used the L_{10} life model for the rolling-contact fatigue life calculation. Yucesan and Viana (Yigit A. Yucesan, 2021) developed a hybrid physics-informed neural network model for estimating the cumulative damage of wind turbine main bearings. The calculated aeroelastic loads were used as a training set for neural network model and visual inspection approach is used for modelling the grease degradation.

In this report, a more detailed L_{10} life model is developed whereby aeroelastic loads are used to estimate the bearing operational conditions such as lubricant viscosity, temperature and contamination level of a lubricants. Together with this bearing operational conditions, the influence of the environmental conditions such as turbulence level, annual mean wind speeds and the ambient temperature are quantified on the main bearing life. Finally, a model for estimating the High Speed Shaft (HSS) bearing loads based on generator torque is also proposed in this report.

3. Methodology for lifemodel of wind turbine bearings

This chapter will outline the methodology of using the standard 281 "Rolling bearings — Dynamic load ratings and rating life" (ISO 281, 2007) to estimate the fatigue life of bearings, given bearing specifications and bearing loads. To apply this methodology to wind turbine main bearings, one will have to determine the loads from either measurements or simulations. The latter is achieved using an aeroelastic model to calculate the main bearing loads arising from operating a wind turbine in a wind climate as specified by a Rayleigh annual probability distribution of the average wind speed, specified in IEC 61400-1. Additional environmental parameters, such as the turbulence intensity and wave conditions, are also specified, whereby the influence of these environmental parameters on the main bearing lifetime can be examined.

3.1. Life modelling of bearings

Bearing fatigue life is defined as the total number of revolutions of bearing operation until the failure criterion is developed (Budynas & Nisbett, 2011). The failure criterion is defined either as the number of revolutions needed for fatigue failure or the number of hours of operation at a constant angular speed to a fatigue failure. Since this is a stochastic variable, it is quite common in the bearing industry to quantify life with certain reliability. When the bearing life is quantified with 90 % reliability, it is defined as the basic rating life of a bearing. It is defined as the number of revolutions required by a group of 90 % identical bearings to meet or exceed the failure criterion. The bearing basic rating life (L_{10}) subjected to 90 % reliability is given by (ISO 281, 2007),

$$L_{10} = \left(\frac{C}{P_d}\right)^p \text{ [in revolutions]} \tag{1}$$

where, C is the bearing-specific basic dynamic load rating [kN], P_d is the damage equivalent fatigue load (DEFL) [kN], and p is the bearing life exponent: $p = 10/3$ for roller bearings and $p = 3$ for ball bearings.

The basic rating life in terms of hours of operation (L_{10h}) is then obtained as,

$$L_{10h} = \frac{10^6}{60\ \omega}\left(\frac{C}{P_d}\right)^p \text{ [in hours]} \tag{2}$$

where, ω is the angular speed of the bearing [rpm] and the conversion factors are 60 min/h and 10^6 (1 million) revolutions.

P_d is a load with constant magnitude and direction that gives the same rating life as the combined load (radial and axial loads) acting on the bearing in practice. For time-varying axial and radial loads as in the case of wind turbine applications, the resultant time series of the equivalent load (P_e) [kN] is computed as,

$$P_e(t) = X \cdot F_r(t) + Y \cdot F_a(t), \tag{3}$$

where F_r is the radial load [kN] and F_a is the axial load [kN], and X and Y are the radial and axial load factors, respectively. The load factors depend on the specific bearing type and the axial to radial load factor, $e = \frac{F_a}{F_r}$. Finally, P_d for the time-varying load is computed by employing the load duration distribution (LDD) method (Wang, Nejad, Bachynski, & Moan, 2020) as,

$$P_d = \left[\frac{\sum_i p_i^p l_i}{\sum_i l_i}\right]^{\frac{1}{p}} \tag{4}$$

where, p_i is the load amplitude [kN] obtained by binning the equivalent load (P_e) times series at different load levels, l_i is the number of load cycles in a load bin i .

In order to account for different reliability levels and the influence of the bearing operating conditions on the bearing rating life, two life modifications are introduced in the basic rating life L_{10} as per the ISO 281 Standard (ISO 281, 2007) and the resulting life is termed as the modified rating life L_{nmh} given by,

$$L_{nmh} = a_1 a_{ISO} L_{10h} \text{ [in hours]} \tag{5}$$

where, a_1 is the life modification factor for reliability and, a_{ISO} is the life modification factor for special operating conditions such as lubrication conditions (*i.e.,* type and viscosity of the lubricant) and the contamination of the lubricants. The calculation of the life modification factors a_1 and a_{ISO} with relevant discussions will be presented in Sections 5.1.1 and 5.1.2, respectively.

3.2. Lifemodel applied to wind turbines

One has to describe to distribution of the wind speeds that the turbine will experience when installed in a specific site for estimating main bearing life. However when the site specific wind speed distribution is unavailable, the standard wind classes as specified in the IEC 61400-1standard "Wind energy generation systems –Part 1: Design requirements." (IEC 61400-1 Ed. 4, 2019) and IEC 61400-3 standard "Wind energy generation systems - Part 3-1: Design requirements for fixed offshore wind turbines" (IEC 61400-3-1, 2019) can be used. Below the basic definitions of the wind classes and the turbulence intensity will be provided. Secondly the wave conditions are specified.

3.2.1. Wind classes and wind speed distribution

The wind classes of the IEC 61400-1 (IEC 61400-1 Ed. 4, 2019) and -3 (IEC 61400-3-1, 2019) standards are specified from a Rayleigh probability distribution of the wind speed given as,

$$P_R(V_h) = 1 - \exp[-\pi(V_h/2V_{ave})^2], \quad (6)$$

where V_h is the wind speed at hub height of the turbine, V_{ave} is the annual average wind speed.

The IEC standard have defined design wind classes according to the annual average wind speeds V_{ave} as :

Design wind class I : V_{ave} = 10.0 m/s

Design wind class II : V_{ave} = 8.5 m/s

Design wind class III : V_{ave} = 7.5 m/s

These are used as design wind classes for wind turbines in case the specific wind condition of an installation site is unknown. See Figure 1a for the illustration of the IEC wind speed distributions.

3.2.2. Turbulence intensity

Besides the annual wind speed distribution of an installation site, one will also be interested in knowing the so called turbulence intensity, because this is characterizing the amount of fluctuation around the mean wind speed.

According to the IEC standard (IEC 61400-1 Ed. 4, 2019), the turbulence intensity I is related to the wind speed V_h at the hub height of the turbine by,

$$I = I_{ref}(0.75\, V_h + b)/V_h, \quad (7)$$

where I_{ref} is called the reference turbulence intensity level and the parameter b = 5.6 m/s.

The IEC standard defines the following design turbulence designations to be provided along with the wind class number.

Design turbulence class A+ $I_{ref} = 0.18$

Design turbulence class A $I_{ref} = 0.16$

Design turbulence class B $I_{ref} = 0.14$

Design turbulence class C $I_{ref} = 0.12$

Thus, the turbulence is changed in accordance with the wind speed when an IEC wind class is defined by the class number (I, II or III) and the turbulence intensity (A+, A, B or C). See Figure 1b for the illustration of turbulence intensity variation of the IEC standard.

3.2.3. Wave conditions

The waves that an offshore wind turbine is exposed to, is often described by the significant wave height H_S, the peak spectral period T_p and the water depth d_{water} as per the IEC 61400-3 standard (IEC 61400-3-1, 2019). Often the joint distributions of wind and waves are also needed for a comprehensive load survey, but a conservative investigation of the fatigue can be obtained if the significant wave parameters are applied for all the wind condition.

3.2.4. Combining lifetime from design load cases

Included in the IEC 61400-1 (IEC 61400-1 Ed. 4, 2019) and IEC 61400-3 (IEC 61400-3-1, 2019) standards, a series of design Load Cases (DLC) can be simulated to study whether a turbine design can reach 20 or 25 years of design life time, respectively. Thus, there is a need to combining the individual life consumption of each load case to provide an estimate of the combined life time of the turbine. This will be explained in a later section of this report, but the main load cases that are evaluated in relation to the fatigue life of the main bearing are listed below:

Design Load Case	Description
1.2	Power production
3.1	Start-up
4.1	Normal shutdown
2.4	Power production plus occurrence of fault – Grid loss

The DLC 2.4 describing the loss of electrical grid will be reported in the deliverable D5.2 of the Hiperwind project, but it is based on the lifetime model presented in this report.

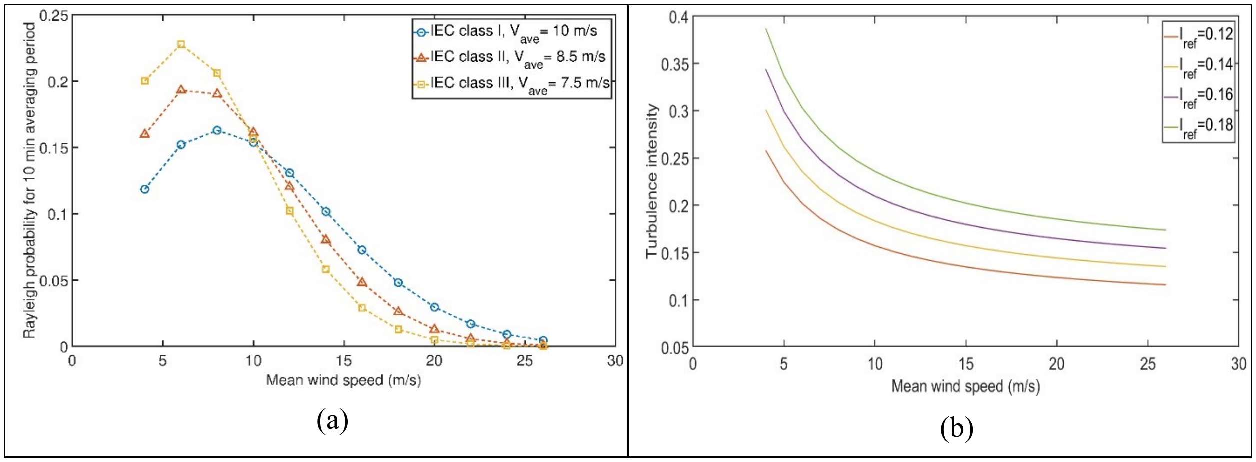


Figure 1: (a) Frequency of occurrence of 10-min mean wind speed V_h for different IEC wind classes and (b) Turbulence intensity (I) as a function of V_h,.

4. Turbine

The Teesside wind farm is an offshore wind farm located in the North sea, England and commissioned for operations on July 5, 2012. This wind farm comprises 27 Siemens SWT-2.3-93 turbines (SWT-2.3-93) with a total capacity of 62 MW. The specifications of the wind farm are given in Table 1.

The drivetrain layout of the Siemens Wind Power SWT 2.3-93 turbine has been specified to the extent that the publicly available source could provide, and the main focus has been to obtain a reasonably accurate description of the main bearing as well as the HSS bearings. Several inventory pages of second-hand spare part provider like SparesInMotion (SparesInMotion, 2023) have been investigated over the time period of the project to identify possible main bearing specifications (SWT-2.3 -93: Main bearing, u.d.), including FAG 230/800 spherical roller bearing from Schaeffler (FAG 230/800, u.d.). The lubrication of the main bearing is Klüberplex BEM 41-301 (Klüberplex BEM 41-301, 2018) as indicated by a SWT 2.3 - 93 main bearing greasing instruction Hove A/S (HOVE, u.d.). The gearbox designation was identified as Winergy PEAB 4456 and it is a three-stage gearbox with a planetary stage and two helical stages (Siemens 2.3 service manual, 2009). Similarly, it was identified that the HSS is supported by a cylindrical roller bearing (NU 2332 C3 (SKF NU 2332 ECML/C3, u.d.)) on the rotor side and a pair of tapered roller bearings (32234 Duplex set (SKF 32234, u.d.)) supporting on the generator side. Finally, the dimensions of the SWP 2.3 - 93 drivetrain have been estimated from the Siemens 2.3 Service Manual (Siemens 2.3 service manual, 2009). The drivetrain layout along with its estimated dimensions of the Siemens SWT-2.3-93 turbine is shown in Figure 2.

As in Figure 2 the main shaft is supported by a double-row spherical roller bearing (FAG 230/800), which is considered the main bearings of the Teesside wind farm. The design parameters related to the main bearing FAG 230/800 are given in the Table 2.

Table 1: Teesside wind farm site and wind turbine specifications. See notes for references.

Teesside environmental conditions	IEC wind class Turbulence Intensity Water depth Significant wave height, H_s Wave period, T_p	$V_{ave} \approx$ 7.1 m/s, assumed as class III [Note a] Not public available 13-16 m [Note b] 1 m [Note c] 10 s [Note c]
Wind turbine: Siemens SWT-2.3-93 [Note d]	Certified IEC wind class Rated power Rated wind speed Rotor diameter Hub height	II A, $V_{ave} = 8.5$ m/s, $I_{ref} = 0.16$ 2.3 MW 13-14 m/s @ 16 rpm 93 m 83.1 m
Drivetrain [Note e]	Main bearing (SWT-2.3 -93: Main bearing, u.d.) Gearbox (Siemens 2.3 service manual, 2009) HSS bearing upwind HSS bearing downwind Generator Power converter	Spherical roller bearing FAG 230/800 One planetary & two helical stages, Winergy PEAB 4456 Cylindrical roller bearing, NU 2332 C3 Tapered roller bearing, 32234 Duplex set Squirrel cage, ABB AMA 500L4/6A BAXYH Full converter, ABB ACS880
Support structures	Monopile Transition piece	

[a] Average wind speed reported in (Papatzimos, Dawood, & Thies, 2018) , [b]Global Wind Atlas (GlobalWindAtlas, u.d.) , [c]Whitby wave measurement station close to Teesside (CoastalMonitoring.org, 2023), [d]Siemens Wind Power data sheet on SWT-2.3-93 turbine (SWT-2.3-93), [e]From spare-part report of Spares In Motion (SparesInMotion, 2023).

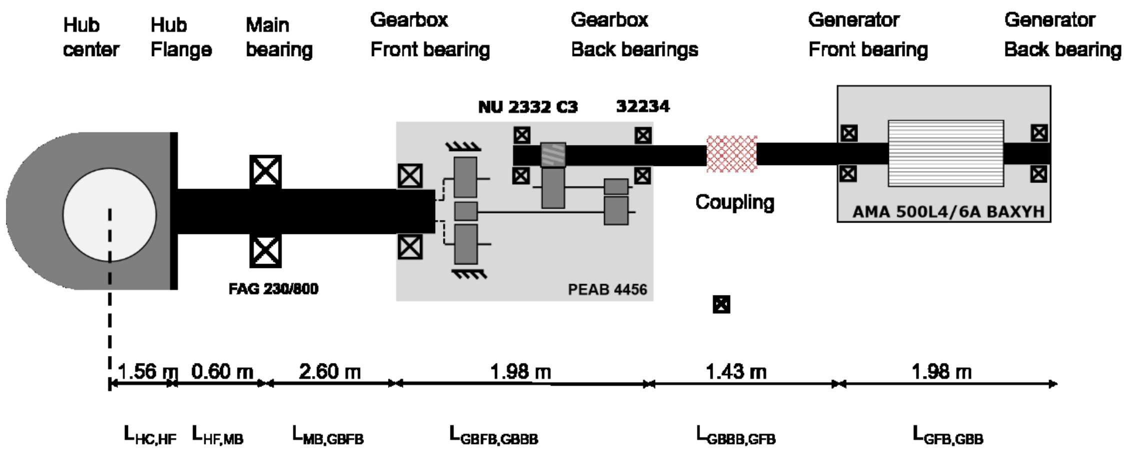


Figure 2 Proposed drivetrain layout of the Siemens SWT-2.3-93 turbine based on the spare part report of SparesInMotion (SparesInMotion, 2023)

The generator of the SWP-2.3-93 is a squirrel cage induction generator well represented by an ABB generator with the designation AMA 500 L4 / 6A BAXYH, identified by following the spare-part offers on SparesInMotion second hand spare part web page (SparesInMotion, 2023). The squirrel cage generator is connected to a full power converter, where the alternating current (AC) with a frequency of the generator is first rectified to a direct current (DC) in the DC link and then converted into an alternating current (AC) matching the frequency of the electrical grid. The output of the power converter is connected to a step up transformer, which is increasing the voltage level to 33 kV of the wind farm collection grid.

Thus, the drivetrain configuration is the so-called type 4, where the generator can operate at any frequency according to the optimal operation of the wind turbine and it is decoupled from the grid status.

From the Spares In Motion spare part report it is suggested that the power converter can be represented by the ABB ASC880 converter and more detail on the functionality in the case of Low Voltage Ride Through is provided in the Delivery report D5.2 of the Hiperwind project.

Table 2: FAG 230/800 spherical roller bearing design specifications ((FAG 230/800, u.d.); (FAG TPI 176) (FAG TPI 197)).

Basic dynamic load rating, C	9300 kN
Pitch diameter, d_P	975 mm
Fatigue load limit, C_u	1450 kN
Radial load factor, X	$= \begin{cases} 1 & \text{for } F_a/F_r \leq= 0.22 \\ 0.67 & \text{for } F_a/F_r >= 0.22 \end{cases}$
Axial load factor, Y	$= \begin{cases} 3.07 & \text{for } F_a/F_r \leq= 0.22 \\ 4.57 & \text{for } F_a/F_r >= 0.22 \end{cases}$
Speed-dependent frictional factor, f_0	3
Basic static load rating, C_0	21200 kN
Heat transfer coefficient, k_q	0.12 $kW/(mm^2\ K)$
Bearing seating surface, A_r	1.58e6 mm^2

5. Model definition

In order to obtain the main bearing operational loads, aeroelastic simulations are performed on the Siemens SWT-2.3-93 turbine according to the design load cases (DLC) 1.2 Normal operation, 3.1 start-up and 4.1Normal shutdown specified by the IEC standard (IEC 61400-3-1, 2019). Here the DLCs 1.2, 3.1 and 4.1 are the load cases for fatigue load assessments. The aeroelastic simulations are performed in the DTU in-house aeroelastic tool called HAWC2 (Larsen & Hansen, 2021). HAWC2 is an aeroelastic code used for computing the loads and displacements of wind turbine structures at given environmental conditions. It consists of a flexible multibody framework for structural modelling, blade element momentum theory combined with a dynamic stall model for aerodynamic modelling and a potential flow model combined with the Morison equation for hydrodynamic load modelling. Finally, the turbine control is employed through dynamic link libraries (DLLs). More details about the HAWC2 aeroelastic tool can be found in (Larsen & Hansen, 2021). The HAWC2 aeroelastic model of the Siemens SWT-2.3-93 turbine was created under the HIPERWIND project consortium (HIPERWIND, u.d.) and is not publicly available due to the confidentiality policy. The HAWC2 representation of the Siemens SWT-2.3-93 turbine drivetrain is shown in Figure 3.

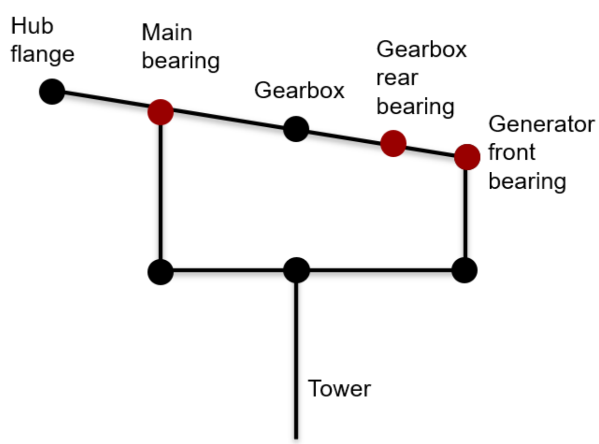


Figure 3: HAWC2 representation of the drivetrain of the SWT-2.3-93

As shown in the figure, the entire drivetrain is modelled as a lumped mass model whereby the mass and inertia of relatively larger elements such as the hub and generators are alone considered in the model. Accordingly, the mass and inertias of the main shaft, gearbox and the bearings are neglected in the model. The bearing reaction forces are modelled using an appropriate frictionless support available in HAWC2. The distance between each lumped mass elements in Figure 3 are obtained from the estimated drivetrain layout in Figure 2.

5.1.1. Life modification factor for reliability, a_1

Since the bearing rating life is defined for 90% reliability, the lifetime for different reliability levels can be obtained by assuming a three-parameter Weibull distribution for survivability (Budynas & Nisbett, 2011) (Nguyen-Schäfer, 2016) . The confidence probability or reliability at the desired lifetime L_p is given by,

$$R = \exp\left[-\left(\frac{x-x_0}{\eta}\right)^{\beta}\right] \in [0,1]; x > x_0, \tag{8}$$

where, R is the desired reliability, $x = L_p/L_{10}$ is the dimensionless life, x_0 is the position parameter, η is the scale parameter and β is the shape parameter. Accordingly, the failure probability at the desired lifetime L_p becomes,

$$P = 1 - R = 1 - \exp\left[-\left(\frac{x-x_0}{\eta}\right)^{\beta}\right], \tag{9}$$

Finally, the relationship between the desired life L_p at a given reliability R (or failure probability p) and the L_{10} life can be expressed as (Budynas & Nisbett, 2011),

$$L_p = a_1 L_{10} = \left(x_0 + \eta(-\ln R)^{\frac{1}{\beta}}\right) L_{10}. \tag{10}$$

The life modification factor (a_1) of roller bearings for different reliability levels are given in ISO standard (ISO 281, 2007) and by using these values, the Weibull parameters are estimated as, $x_0 \approx 0.05$, $\eta \approx 4.3$ and $\beta \approx 1.5$. It is assumed that the Weibull parameters of the ISO 281 standard can be applied to the main bearing of the SWT-2.3-93, since this can provide a proposal on the expected reliability of the main bearings of the Teesside wind farm.

5.1.2. Life modification factor for special operating conditions, a_{ISO}

Even though the basic rating life only accounts for the bearing load, in reality, the following factors also significantly affect the bearing fatigue life: the conditions (viscosity) of the lubricant inside the bearing, the fatigue limit and residual stress of the material, contamination of the lubricant and the ambient conditions. In order to account for all these factors in the bearing fatigue life, the life modification factor a_{ISO} is introduced in the ISO standard (ISO 281, 2007) and the same for the roller bearings is expressed as,

$$a_{ISO} = \begin{cases} 0.1\left[1-\left(1.5859-\frac{1.3993}{\kappa^{0.054381}}\right)\left(\frac{e_C C_u}{P_d}\right)^{0.4}\right]^{-9.185} & \text{for } 0.1 \leq \kappa < 0.4 \\ 0.1\left[1-\left(1.5859-\frac{1.2348}{\kappa^{0.019087}}\right)\left(\frac{e_C C_u}{P_d}\right)^{0.4}\right]^{-9.185} & \text{for } 0.4 \leq \kappa < 1 \\ 0.1\left[1-\left(1.5859-\frac{1.2348}{\kappa^{0.071739}}\right)\left(\frac{e_C C_u}{P_d}\right)^{0.4}\right]^{-9.185} & \text{for } 1 \leq \kappa \leq 4 \end{cases} \cdot \tag{11}$$

Here, where C_u is the bearing-specific fatigue load limit and its value for the FAG 230/800 bearing is given in Table. 2, and e_C and κ are the contamination factor and the actual-to-rated viscosity ratio of the lubricant, respectively. It should be noted that when $\kappa < 0.1$, the calculation of a_{ISO} is not possible and for $\kappa > 4$, the value of κ can be used as 4 (ISO 281, 2007). The viscosity ratio (κ) and the contamination factor (e_C) are calculated based on the specifications of the Klüberplex BEM 41-301 (Klüberplex BEM 41-301, 2018). In the following, the estimation of κ and e_C will be explained in detail.

5.1.2.1. Estimation of the viscosity ratio, κ

The viscosity ratio (κ) at a given operating temperature (T_B) is defined as the ratio of the kinematic viscosity (ν) and the reference viscosity (ν_1) of the lubricant (*i.e.,* $\kappa = \nu/\nu_1$). The reference viscosity (ν_1) depends on the bearing angular speed (ω) and the pitch diameter (d_P) and is given by (FAG TPI 176),

$$\nu_1 = \begin{cases} 45000 \cdot \omega^{-0.83} \cdot d_P^{-0.5} & \text{for } \omega < 1000 \text{ rpm} \\ 4500 \cdot \omega^{-0.5} \cdot d_P^{-0.5} & \text{for } \omega \geq 1000 \text{ rpm.} \end{cases} \tag{12}$$

Here, the pitch diameter (d_P) is in mm.

Given the bearing operating temperature (T_B), the operating kinematic viscosity (ν) can be estimated from the viscosity temperature relationship given by the ASTM standard (ASTM D341-20e1, 2020) as,

$$\nu = [z-0.7] - exp(-0.7487 - 3.295[z-0.7] + 0.6119[z-0.7]^2 - 0.3193[z-0.7]^3), \tag{13}$$

$$loglog(z) = A - Blog(T_B). \tag{14}$$

Here, T_B is in K, ν is in mm^2/s, A and B are constants, and the logarithm base is 10. The kinematic viscosity of the Klüberplex BEM 41-301 is approximately 300 mm^2/s at 313.15 K (40° C) and 23 mm^2/s at 373.15 K (100° C) (Klüberplex BEM 41-301, 2018) and with that, the constants A and B are estimated as 8.78 and 3.36, respectively.

5.1.2.2. Estimation of main bearing operating temperature, T_B

In general, the main bearing operating temperature (T_B) can be obtained from SCADA if available, else it can be estimated with the following assumptions:

1. The system (main bearing) is in a steady state equilibrium, *i.e.,* the amount of heat generated by the system is the same as that of the amount of heat dissipated from the system.

2. The heat dissipation from the lubricant is negligible.

By following these assumptions, the main bearing operating temperature (T_B) can be estimated by equating the amount of heat generated inside the bearing with the amount of heat dissipated from the bearing. The heat generated during the bearing operation is attributed to the bearing friction. Accordingly, the heat flow generated ($\dot{Q}_\mathrm{B}$) by the bearing friction is (FAG TPI 176),

$$\dot{Q}_\mathrm{B} = N_R = M_R \frac{\omega}{9550} = (M_0 + M_1)\frac{\omega}{9550}, \tag{15}$$

where, N_R is the bearing frictional power in W, M_R is the bearing frictional torque in N-mm, M_0 is the speed-dependent frictional torque in N-mm, and M_1 is the load-dependent frictional torque in N-mm.

The speed-dependent frictional torque M_0 is given by (FAG TPI 176),

$$M_0 = \begin{cases} f_0 \cdot (\nu \cdot \omega)^{(2/3)} \cdot d_P^3 \cdot 10^{-7} & \text{for} \quad \nu \cdot \omega \geq 2000, \\ f_0 \cdot 160 \cdot d_P^3 \cdot 10^{-7} & \text{for} \quad \nu \cdot \omega < 2000, \end{cases} \tag{16}$$

where, f_0 is the speed-dependent frictional factor and its value is given in Table. 2.
The load-dependent frictional torque M_1 is given by (FAG TPI 176),

$$M_1 = f_1 P_1 d_P. \tag{17}$$

The parameters in Eq. (17) are calculated as (FAG TPI 176),

$$f_1 = 0.00075 \cdot \left(\frac{P_0}{C_0}\right)^{0.5}. \tag{18}$$

$$P_0 = F_r + 3F_a. \tag{19}$$

$$P_1 = \begin{cases} 1.6 \quad if \ \dfrac{F_a}{F_r} > e \\ F_r \cdot \left\{1 + 0.6 \cdot \left(\dfrac{F_a}{eF_r}\right)^3\right\} \quad if \ F_a/F_r \leq e \end{cases} \tag{20}$$

where, f_1 is the load-dependent frictional factor, P_0 is the equivalent static load, C_0 is the basic static load rating and its value is given in Table 2, and P_1 is the decisive load for frictional torque.

The heat flow dissipated $\dot{Q}_s$ to the environment is obtained by (FAG TPI 176),,

$$\dot{Q}_s = k_q A_r \Delta T. \tag{21}$$

where, k_q is the heat transfer coefficient in $W/(mm^2\ K)$ and A_r is the bearing seating surface in mm^2. These values are given in Table 2. ΔT is the temperature difference given by, $\Delta T = T_A - T_B$, where, T_A is the ambient temperature in K.

When the system is in equilibrium,

$$\dot{Q}_s = \dot{Q}_B. \tag{22}$$

Accordingly, for a given ambient temperature (T_A) and bearing loads (F_r and F_a), the bearing operating temperature (T_B) and the operating kinematic viscosity (ν) is obtained from Eq. (22) subjected to a constraint in Eq. (14) as,

$$T_B = \frac{N_R(\nu, F_r, F_a)}{k_q A_r} + T_A. \tag{23}$$

Where $N_R(\nu,F_r,F_a)$ is the bearing frictional power described as a function of the operating viscosity and the radial and axial loads, k_q is the heat transfer coefficient, A_r is the bearing seating surface area and T_A is the ambient temperature.

5.1.2.3. Estimation of the contamination factor, e_C

In order to account for the lubricant contamination on the bearing life, a factor called contamination factor e_C is included in the calculation of the life modification factor (a_{ISO}) by ISO standard. A lubricant is said to be contaminated when there are solid particles in it. The presence of these solid particles results in permanent indentations in the raceway. These indentations areas result in a rise in localized stress levels which will eventually reduce the life of the bearing. Depending on the type, size and quantity of the solid particles, the level of contamination is determined. The ideal way to determine the level of contamination is to test the samples in a laboratory for a detailed analysis. Even though the level of accuracy is high in laboratory tests, they are often expensive and time-consuming. As a result, the wind farm operators would often choose a borescope inspection of the lubricants by qualified technicians. More details about the errors and methods of quantifying the contamination can be found in (Yucesan & Viana, 2020).

The level of grease lubricant contamination is classified into the following six categories in ISO standard (ISO 281, 2007)

- Extreme cleanliness
- High cleanliness
- Normal cleanliness
- Slight to typical contamination
- Severe contamination
- Very severe contamination

After identifying the level of contamination, the contamination factor (e_C) for grease lubricants can be computed as a function of κ and d_P (ISO 281, 2007). In order to understand the correlation between the level of contamination with the bearing life, all the levels of contamination are considered in this article. The algorithm of computing the life modification (a_{ISO}) explained in Section (13) is given as a flowchart in Figure 4.

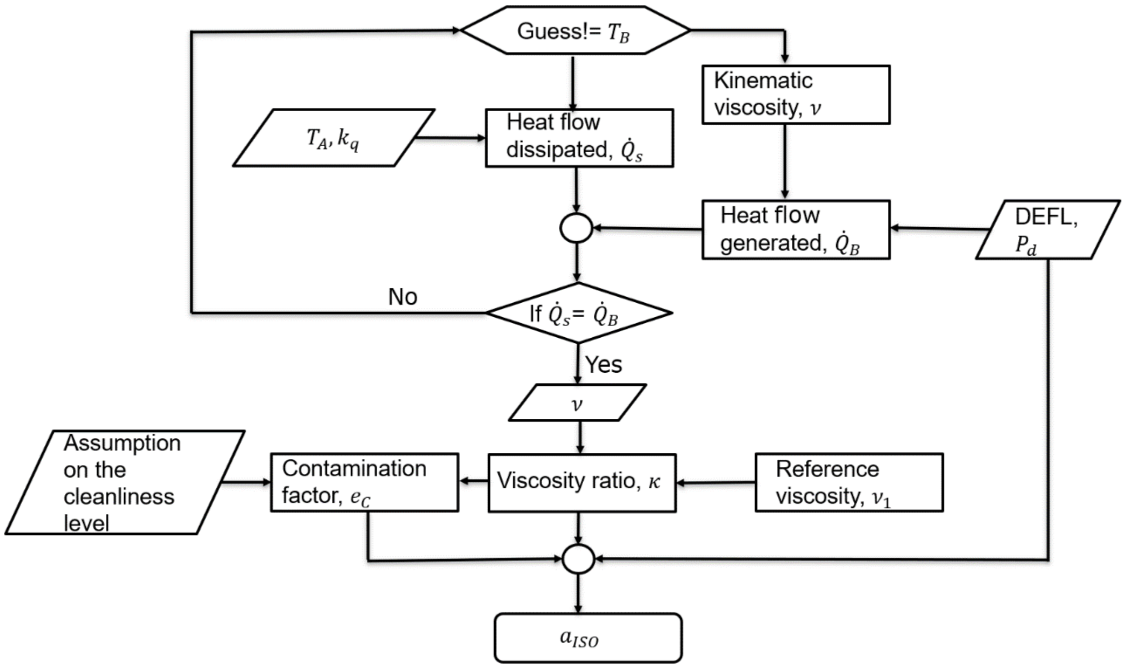


Figure 4: The life modification factor, a_{ISO} calculation algorithm

6. Results

The influence of environmental, and operational conditions on the bearing life will be discussed in this section. Using HAWC2, aeroelastic simulations are performed for the design load cases (DLCs) 1.2, 3.1 and 4.1, and the main bearing axial F_a, lateral F_x and vertical F_y loads are obtained for all these load cases. All loads are in the bearing rotational coordinate system and then the main bearing radial load is obtained as $F_r = \sqrt{F_x^2 + F_y^2}$. Finally, the resultant DEFL (P_d) is obtained by employing Eq. (4) for each simulation and then the modified rating life (L_{10mh}) for all these simulations can be computed using Eq. (5). After that, the equivalent life (L_{10mv}) for each mean wind speed (V_h) is obtained as,

$$L_{10mv}(V_h) = \sum_{i=1}^{N_1} \frac{L_{10mh,i}}{N_1 \cdot 365 \cdot 24} \; [in\ years], \tag{24}$$

where, N_1 is the total number of ten min or 100 second simulations for each mean wind speed under a particular load case, and $L_{10h,i}$is the modified rating life of i^{th} simulation of either 10 minute duration for fatigue load cases and only 100 second for start up or shut down loads cases as describe in the next chapters. The factors in the denominator reflect a conversion from hour to years by the number of hours per year.

Finally, the total modified rating life L_{10mt} for each load case can be obtained by combining the equivalent life (L_{10mv}) for each mean wind speed with its annual frequency occurrence $(P_R(V_h))$ computed using Eq. (6) as,

$$L_{10mt} = \frac{\sum_{j=1}^{N} P_R(V_{h,j})}{\sum_{j=1}^{N} \frac{P_R(V_{h,j})}{L_{10mv}(V_{h,j})}} \; [\text{in years}], \tag{25}$$

where, N is the total number of mean wind speeds for a load case considered, $(P_R(V_{h,j}))$is the annual frequency of occurrence of j^{th} mean wind speed, and $L_{10mv}(V_{h,j}))$ is the total modified rating life of j^{th} mean wind speed. The results of all DLCs will be discussed in detail next.

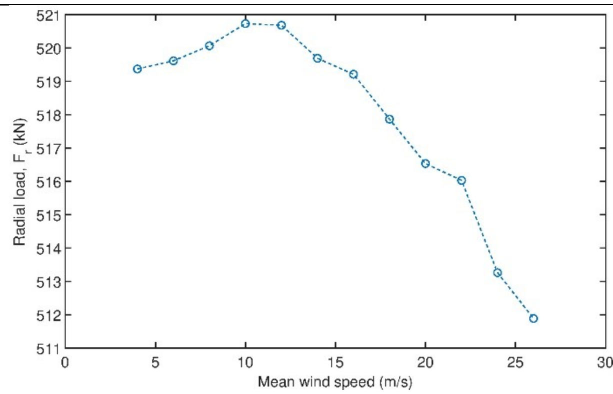
6.1. Design load case DLC 1.2 Normal operation

The description of the DLC 1.2 as per IEC standard (IEC 61400-3-1, 2019) is given in Table 3. It consists of 54 ten minute simulations (three wind yaw misalignments, three wave yaw misalignments and six random turbulent seeds) for each mean wind speed. This results in a total number of 648 ten minute simulations with a 50 Hz sampling frequency for DLC 1.2.

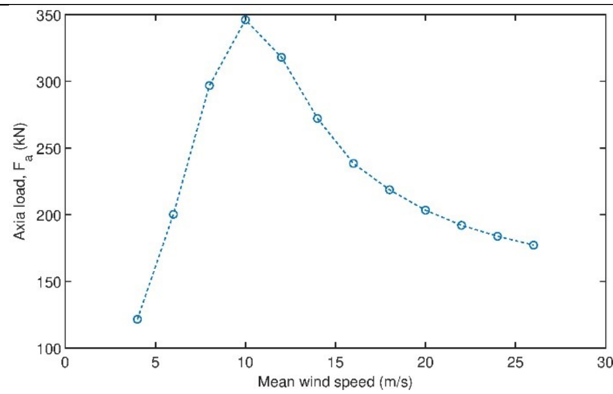
Table 3: DLC1.2 load case Normal Operation description (IEC 61400-3-1, 2019).

Load case	DLC 1.2
Design situation	Normal power production
Mean wind speed at hub height (V_h)	4-26 m/s *in steps of* 2 m/s
Turbulence model	Normal turbulence model (NTM)
Wind yaw	$-10/0/+10$ [deg]
Wind shear	0.14
Waves	Normal sea state (NSS), $H_s = \mathrm{E}[H_s \vert V_h]$
Wind and wave directionality	Misaligned, multidirectional
Wave yaw	$-10/0/+10$ [deg]
Sea currents	No currents
Simulation time and sampling frequency	600 s (Without transients) and 50 Hz

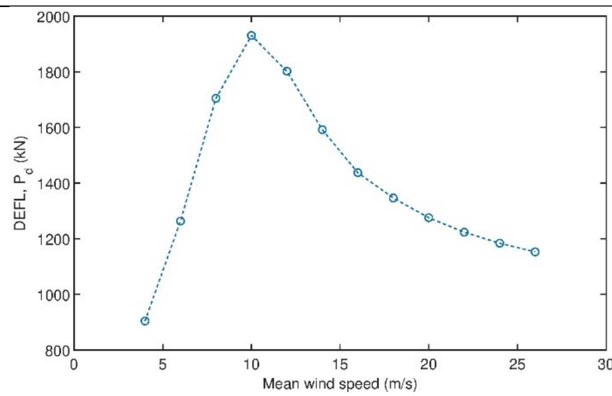
Aeroelastic simulations are performed as per DLC 1.2 load case to obtain the main bearing axial (F_a), lateral (F_x) and vertical (F_y) loads from these 648 simulations in HAWC2. Subsequently, the DEFL (P_d) is estimated using Eq. (4) for each simulation. The computed DEFL is constant in magnitude and direction. For the illustration purpose, the mean values of the axial load, radial load and DEFL is computed for each mean wind speed from its 54 ten-minute simulations and then the resultant average as a function of the mean wind speed as shown in Figure 5. Since the lateral and vertical main bearing loads are directly correlated with the blade edgewise loads, they do not change with the mean wind speed. This is also evident from Figure 5(a) that the change in the radial load with the mean wind speed is insignificant. On the other hand, the main bearing axial load is correlated with the rotor thrust load, and as a result, the mean value of the axial load increases till the rated mean wind speed and then decreases with the mean wind speed as shown in Figure 5(b), which is typical behavior of the rotor thrust with the mean wind speed. The axial load to radial load ratio for the entire span of operating conditions is always higher than 0.22, and this results in a higher contribution of the axial load to the equivalent load (P_e). Hence, the resultant DEFL (P_d) resembles the behavior of the axial load with the mean wind speed as seen in Figure 5(c).



(a) Radial load (F_r) variation



(b) Axial load F_a variation



(c) Damage equilent fatigue load (DEFL) P_d variation

Figure 5: Main bearing loads as a function of mean wind speed for DLC 1.2 load case with $V_{ave} = 8.5$ m/s, and $I_{ref} = 0.16$.

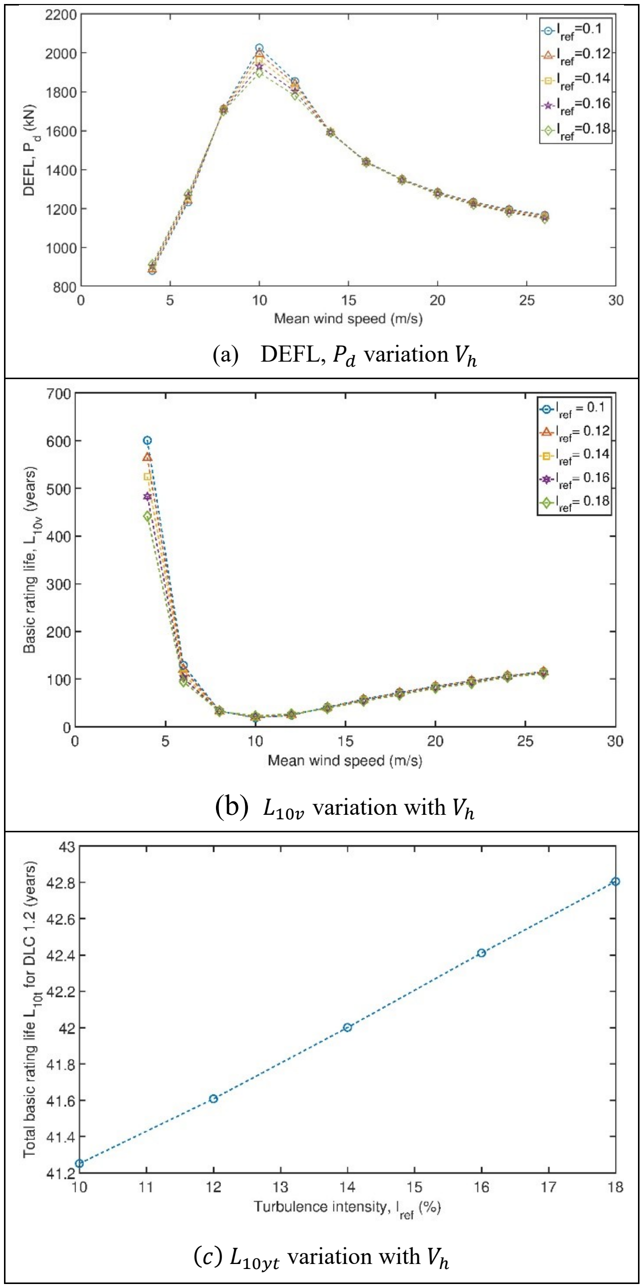
The bearing loads are the first and foremost factor affecting the bearing life and it is mainly dictated by the environmental conditions about which a wind turbine operates. The following two parameters are chosen as key parameters through which the influence of environmental conditions on bearing life can be quantified:

i. Reference Turbulence intensity, I_{ref}
ii. Annual mean wind speed, V_{ave}

6.1.1. Influence of turbulence intensity I_{ref}

Since the site-specific turbulence intensity for the Teesside wind farm is not publicly available, the IEC 61400-1 ED 4 (IEC 61400-1 Ed. 4, 2019) specified turbulence categories $\mathrm{A+}\ (\mathrm{I}_{ref} = 0.18), \mathrm{A}\ (\mathrm{I_{ref}} = 0.16), \mathrm{B}\ (\mathrm{I_{ref}} = 0.14)$ and $\mathrm{C}\ (\mathrm{I}_{ref} = 0.12)$ are considered in this study. In addition, $I_{ref} = 0.1$ is also considered in this study. The DLC 1.2 load case is simulated for all these five I_{ref} values and it resulted in five sets of 648 ten minute time series for the bearing loads (*i.e.,* F_x, F_y and F_z). The input parameters for these simulations are given in Table 2 and Table 3.

The resultant DEFL (P_d) is obtained by employing Eq. (4) for each simulation and the mean values for each mean wind speed is shown in Figure 6(a). Upon computing the DEFL (P_d) from the five sets of 648 simulations corresponds to DLC 1.2, the basic rating life is obtained using Eq. (2) and the resultant average in years for each mean wind speed is shown in Figure 6(b). As in Eq. (2), the basic rating life is inversely proportional to the DEFL (P_d). As a result, the individual life follows the inverted pattern of P_d with mean wind speed. The basic rating life L_{10v} reaches its minimum around the rated mean wind speed ($V_h = 12$ m/s) as seen in Figure 6(b). For each mean wind speed, this individual life can be regarded as a basic rating life resulting from the continuous operation of a wind turbine at that mean wind speed. The total basic rating life L_{10yt} under DLC 1.2 load case is obtained by combining the individual basic rating life L_{10v} of each mean wind speed with its annual frequency of occurrence \similar to Eq. (25). The results for the chosen I_{ref} values are shown in Figure 6(c). As seen in the figure, the total lifetime increases with turbulence intensity. The increase in L_{10yt} with I_{ref} is due to the fact that the DEFL (P_d) is decreasing with an increase in I_{ref} around the peak thrust load as shown in Figure 6(a). The higher the I_{ref}, the higher will be spread in the wind speed from its mean value and as a result, the lower will be the P_d due to the peaked nature of the curve around the rated mean wind speed. However, the change in the total basic rating life with I_{ref} is insignificant. It is important to note that the mean wind speed range (*i.e.,* 8 - 14 m/s) dominate the total life of a bearing due to their higher frequency of occurrence and a relatively larger P_d as compared to the other mean wind speeds.



(a) DEFL, P_d variation V_h

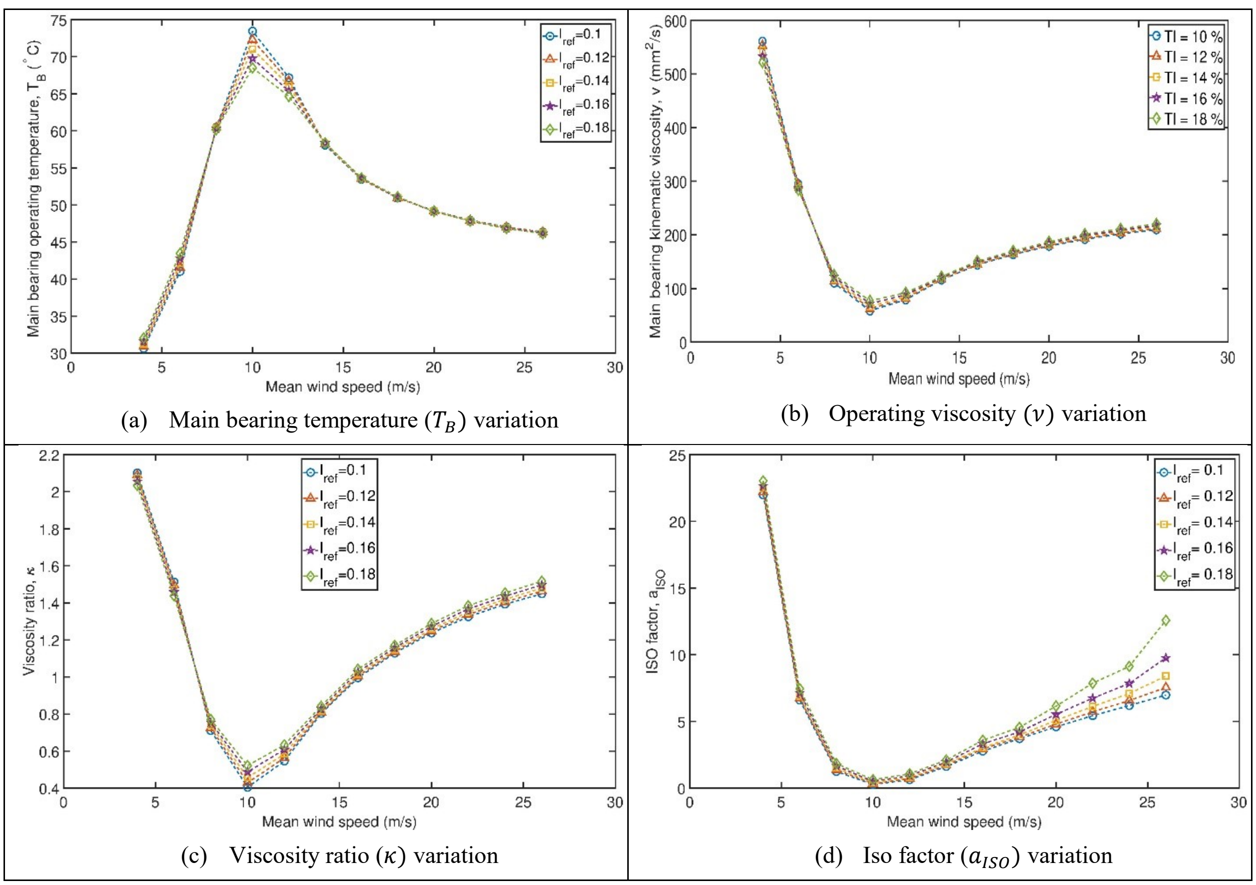
(b) L_{10v} variation with V_h

(c) L_{10yt} variation with V_h

Figure 6: Influence of the turbulent intensity on the main bearing basic rating life for DLC 1.2 load case with $V_{ave} = 8.5$ *m/s.*

The influence of the turbulence intensity on the modified rating life will be discussed hereafter. As shown in Eq. (5) there are two life modification factors (a_1 and a_{ISO}) that need to be computed to obtain the modified rating life. The life modification factor for reliability a_1 is considered to be unity for this part of the subsection and the resulting modified life is then with 90 % reliability. For all the five sets of 648 ten minute simulations, the time series of a_{ISO} is obtained by following the procedure outlined in Section 5.1.2 (Figure 4). For this part of the calculations, it is considered that the ambient temperature is $T_A = 22\,°\text{C}$, the annual average mean wind is $V_{ave} = 8.5$ m/s and the level of cleanliness is assumed to be normal cleanliness ($e_C = 0.7$ (ISO 281, 2007)).

The resultant mean values of T_B, ν, κ and a_{ISO} for each mean wind speed at a given I_{ref} are shown in Figure 7. As seen in Figure 7(a), the bearing operating temperature (T_B) is exhibiting similar behaviour as that of the DEFL (P_d) with respect to mean wind speeds. For a given ambient temperature (T_A), the bearing operating temperature is directly proportional to the frictional power (N_r) as in Eq. (23) and thereby to the DEFL (P_d) as well. As a result, T_B reaches its maximum around the rated mean wind speed as seen in Figure 7 (a). Also, the higher the P_d higher will be the T_B around the rated mean wind speed (12 m/s). As a result, the operating temperature (T_B) is highest for the load case with $I_{ref} = 0.1$ and lowest for $I_{ref} = 0.18$. As per the viscosity-temperature relationship (ASTM D341-20e1, 2020), the operating viscosity (ν) is inversely proportional to T_B. As a result, ν decreases with mean wind speed (V_h) till the rated mean wind speed and increases thereafter with V_h as shown in Figure 7(b). This similar behavior is also seen in for κ and a_{ISO} with the mean wind speed as in Figure 7(c) and Figure 7(d).



(a) Main bearing temperature (T_B) variation

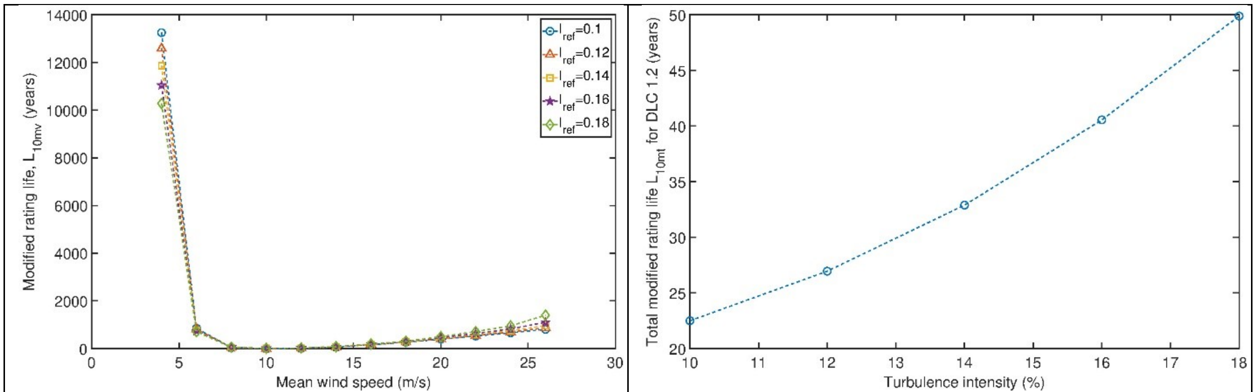
(b) Operating viscosity (ν) variation

(c) Viscosity ratio (κ) variation

(d) Iso factor (a_{ISO}) variation

Figure 7: Variation of the main bearing operational parameters with mean wind speed ($e_C = 0.7$ (normal cleanliness), $T_A = 22°$ C).

Upon computing a_{ISO} with $a_1 = 1$, the modified rating life (L_{10mv}) for each mean wind speed is obtained by using Eq. (24) and then subsequently the total modified rating life (L_{10mt}) under DLC 1.2 for the chosen I_{ref} values are computed by using Eq. (25). These results are shown in Figure 8(a). Since the modified rating life is obtained by multiplying the life modification factors with the basic rating life (as in Eq. (5)). For the moderate mean wind speeds (*i.e.,* 8 - 14 m/s), a_{ISO} is either close to one or less than one and it is much greater than one for all other wind speeds as seen in Figure 7(d). Accordingly, the modified rating life is lower than the basic rating for the cases where $a_{ISO} < 1$. Moderate wind speeds dominate the total lifetime due to their high frequency of occurrence, lower basic rating life and a_{ISO}. As a result, the total modified rating life decreases with an increase in I_{ref} as seen in Figure 8(b).

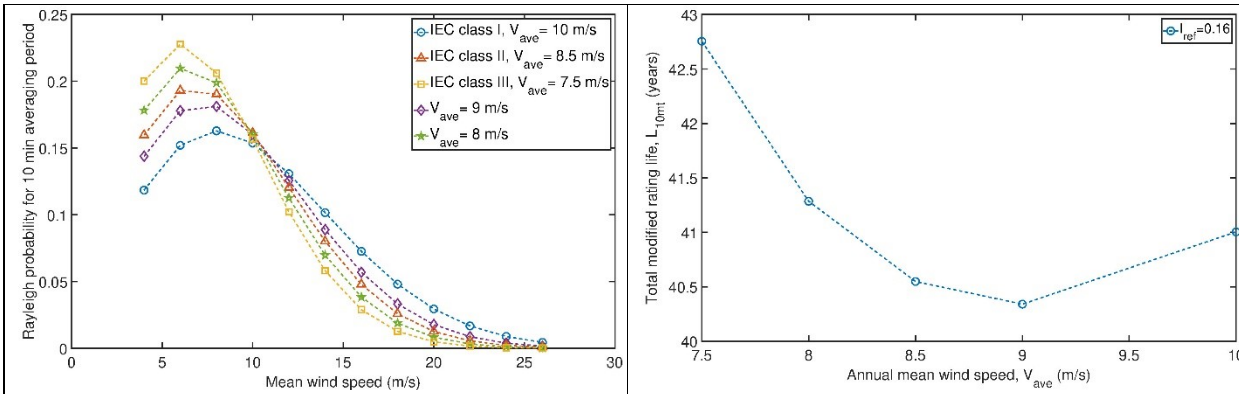


(a) L_{10mv} variation with V_h (b) L_{10mt} variation with V_h

Figure 8: Influence of the turbulence intensity on the main bearing modified rating life for DLC 1.2 load case ($e_C = 0.7$ (normal cleanliness), $T_A = 22°$ C, $V_{ave} = 8.5\ m/s$).

6.1.2. Influence of the annual mean wind speed

Even though the Teesside wind farm is designed for IEC wind class III (Papatzimos, Dawood, & Thies, 2018), the influence of other wind classes given by IEC 61400-1 ED 4 (IEC 61400-1 Ed. 4, 2019) on the main bearing life is studied in this report. It helps to understand the correlation between the main bearing life with the annual mean wind speed as well as the turbulence level. The frequency of occurrence of each mean wind speed V_h given the 10 min annual average wind speed (V_{ave}) is modelled using the Rayleigh distribution (IEC 61400-1 Ed. 4, 2019) and is given in Eq. (6).In addition to the IEC-specified wind classes I $(V_{ave} = 10\ m/s), II\ V_{ave} = 8.5\ m/s)$ and $III\ (V_{ave} = 7.5\ m/s)$,two more values for $V_{ave} = 8$ and 9 m/s) are also considered in this study. The annual mean wind speed (V_{ave}) determines the frequency of occurrences of load conditions of each mean wind speed and thereby influencing the bearing life as shown in Eq. (25). The frequency of occurrence of different hub-height mean wind speeds (V_h) for the chosen values of V_{ave} is shown in Figure 9(a). A total number of 648 ten min load cases corresponding to $I_{ref} = 0.14, e_C = 0.7$ and $T\_A = 22°$ C are considered for this part of the study. The basic rating life and the modified rating life of each mean wind speed remain the same for all different annual mean wind speeds. By combining the modified rating life of each mean wind speed with its frequency of occurrence as given by Eq. (25), the total modified rating life is computed for each V_{ave} and the results are shown in Figure 9(b). The result signifies that the change in V_{ave} can bring a maximum change in the lifetime of around 5 % for the same loading conditions.



(a) Frequency of occurrence of V_h (b) L_{10mt} variation with V_{ave}.

Figure 9: Influence of the different IEC wind classes on the main bearing modified rating life for DLC 1.2 load case ($I_{ref} = 0.16, e_C = 0.7, T_A = 22° C$).

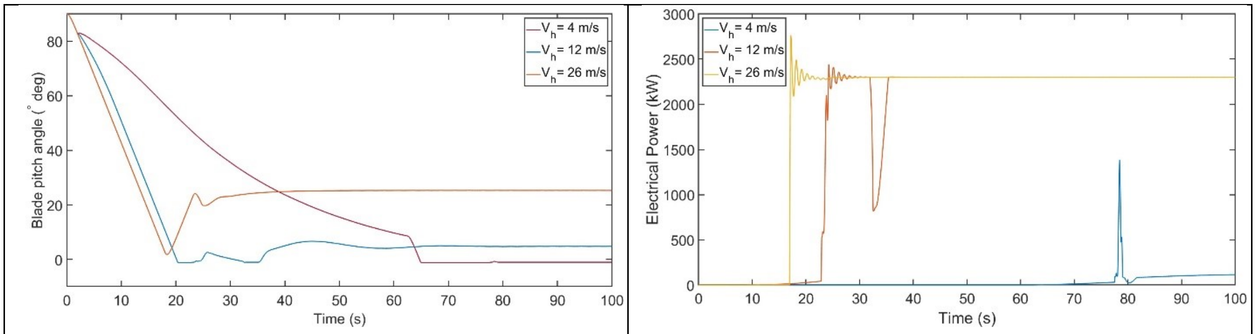
6.2. Design Load Case DLC 3.1 – Start-up

The description of the load case DLC 3.1 as per IEC standard (IEC 61400-3-1, 2019) is given in Table 4 and the aeroelastic simulations are performed as per the description in Table 4 in HAWC2. It consists of three 100 second simulations for three wind speeds. It is a steady wind case as per IEC standard and the wind profile at different heights can be modelled using normal wind profile (NWP) model given by IEC standard (IEC 61400-1 Ed. 4, 2019).

Table 4: DLC3.1 Start-up load case description (IEC 61400-3-1, 2019).

Load case	DLC 3.1
Design situation	Start-up
Wind speed at hub height (V_h)	$V_{in} = 4\frac{m}{s}, V_{rated} = 12\frac{m}{s}$ and $V_{out} = 26\frac{m}{s}$
Turbulence	None
Wind profile	Normal wind profile (NWP) model
Wind yaw	None
Wind shear	0.14
Waves	Normal sea state (NSS), $H_s = E[H_s\|V_h]$
Wind and wave directionality	Unidirectional waves
Wave yaw	None
Sea currents	No currents
Simulation time and sampling frequency	100 s (without transients) and 50 Hz
Annual frequency of occurrence	1000 start-up procedures at V_{in} 50 start-up procedures at V_{rated} 50 start-up procedures at V_{out}

By means of pitch action, the startup scenario is simulated in HAWC2 and the pitch response and the electrical power with time is shown in Figure 10.

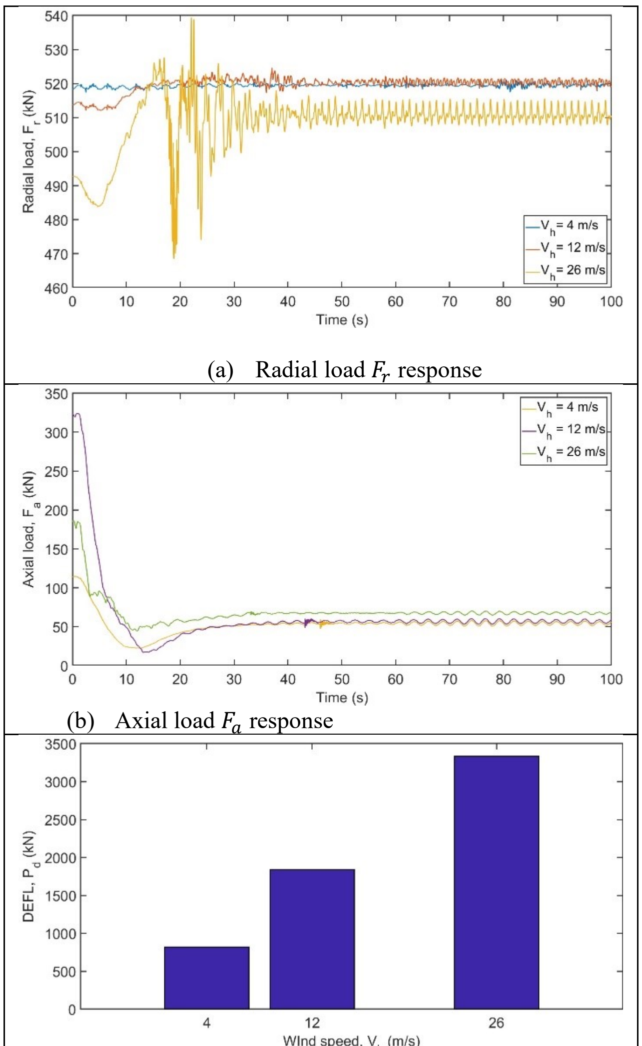


(a) Blade pitch angle response

(b) Electrical power response

Figure 10: DLC 3.1 start-up load case

The main bearing loads obtained from these simulations and the computed DEFL under DLC 3.1 are shown in Figure 11.



(a) Radial load F_r response

(b) Axial load F_a response

(c) DEFL P_d for three V_h values

Figure 11: DLC 3.1 load case, maximum main bearing loads.

Subsequently, the individual modified life (L_{10mv}) for these wind speeds are computed using Eq. (24) and the total modified rating life (L_{10mt}) using Eq. (25) and the results are presented in Table 5.

Table 5. Main bearing lifetime under DLC 3.1 start-up load case

Individual modified rating life	$L_{10mv} = 27043$ years for $V_h = 4$ m/s
	$L_{10mv} = 97$ years for $V_h = 12$ m/s
	$L_{10mv} = 53$ years for $V_h = 26$ m/s
Total modified rating life	$L_{10mt} =$ 223 years

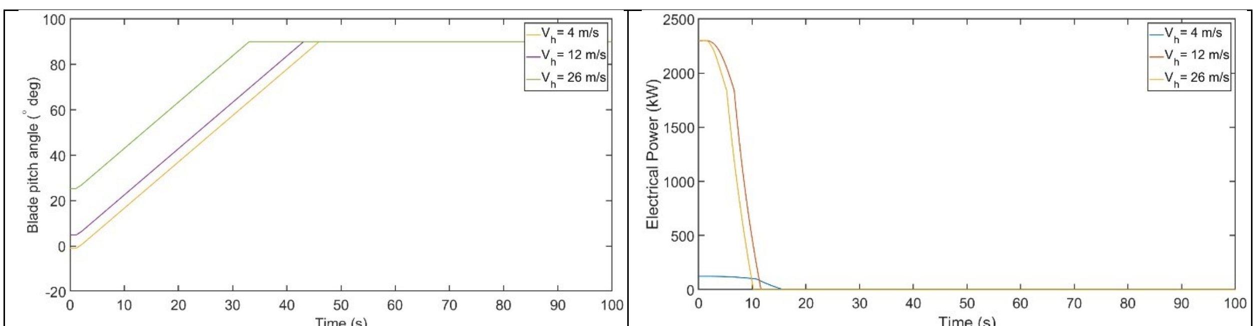
6.3. Design Load Case DLC 4.1 – Normal shutdown

The description of the load case DLC 4.1 corresponds to normal shutdown is given in Table 6 as per IEC standard (IEC 61400-3-1, 2019) and the aeroelastic simulations are performed as per the description given in Table 6 in HAWC2. It consists of three 100 second simulations for three wind speeds.

Table 6: DLC4.1 Normal shutdown load case description (IEC 61400-3-1, 2019).

Load case	DLC 4.1
Design situation	Normal shutdown
Wind speed at hub height (V_h)	$V_{in} = 4\frac{m}{s}, V_{rated} = 12\frac{m}{s} and\ V_{out} = 26\frac{m}{s}$
Turbulence	None
Wind profile	Normal wind profile (NWP) model
Wind yaw	None
Wind shear	0.14
Waves	Normal sea state (NSS), $H_s = E[H_s \vert V_h]$
Wind and wave directionality	Unidirectional waves
Wave yaw	None
Sea currents	No currents
Simulation time and sampling frequency	100 s (without transients) and 50 Hz
Annual frequency of occurrence	1000 shutdown procedures at V_{in} 50 shutdown procedures at V_{rated} 50 shutdown procedures at V_{out}

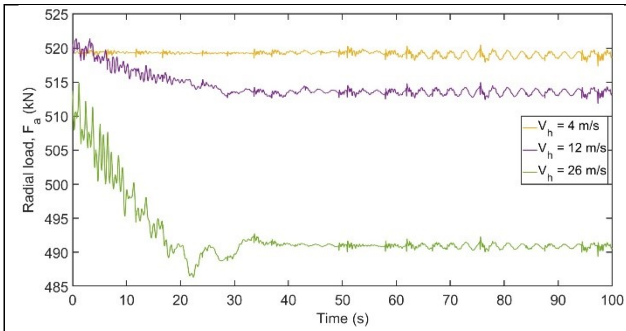
Similar to the DLC 3.1, the normal shutdown scenario is simulated in HAWC2 by means of the pitch action and the resulting pitch response and the electrical power responses are shown in Figure 12.



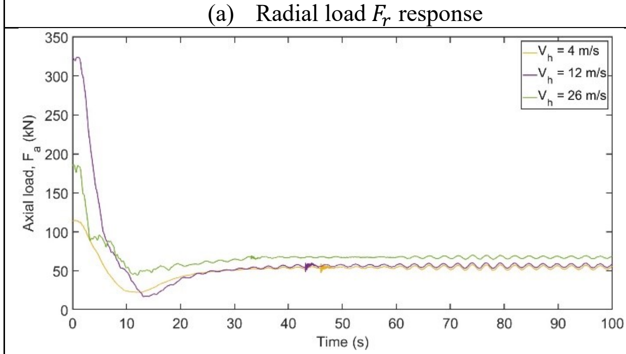
(a) Blade pitch angle response (b) Electrical power response

Figure 12: DLC 4.1 normal shutdown load case

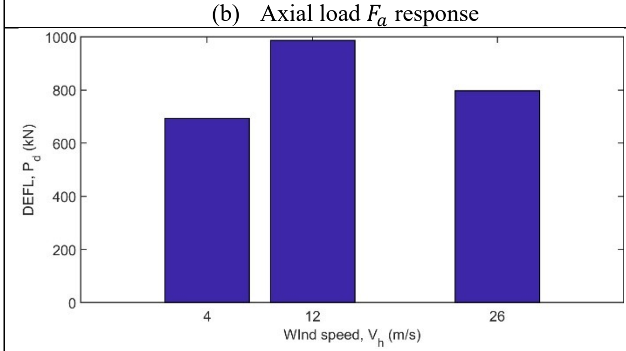
The main bearing loads obtained from these simulations and the computed DEFL under DLC 4.1 are shown in Figure 13.



(a) Radial load F_r response



(b) Axial load F_a response



(c) DEFL P_d for three V_h values

Figure 13: DLC 4.1 load case main bearing loads

Subsequently, the individual modified life (L_{10mv}) for these wind speeds are computed using Eq. (24) and the total modified rating life (L_{10mt}) using Eq. (25) and the results are presented in Table 7.

Table 7. Main bearing lifetime under DLC 4.1 Normal shut down load case

Individual modified rating life	L_{10mv} = 98862 years for V_h = 4 m/s
	L_{10mv} = 22054 years for V_h = 12 m/s
	L_{10mv} = 38021 years for V_h = 26 m/s
Total modified rating life	L_{10mt} = 39591 years

6.4. Bearing life from combined load cases

The influence of the three different load cases (*i.e.,* 1.2, 3.1 and 4.1) on the main bearing fatigue life was discussed so far. However, all these load cases will be encountered by a wind turbine during its lifetime and hence it is important to quantify the main bearing fatigue lifetime under the combined action of these three loadings. In order to do so, the duration of the each load case experienced by the wind turbine needs to be computed and the same is computed as,

Table 8 Specification of duration of the fatigue load cases DLC 3.1 , DLC 4.1 and DLC 1.2

Total number of hours in a year	$365\frac{days}{year}\cdot 24\frac{hour}{day} = 8760$ hours/year
Duration of startup events (DLC 3.1) in a year as per IEC 61400-1 (IEC 61400-1 Ed. 4, 2019)(1000 startup-procedures at 4 m/s., 50 start-up procedures each at 12 m/s and 26 m/s with a duration of 100 s)	30.56 hours (P_{L31} =0.35 %)
Duration of shutdown events (DLC 4.1) in a year as per IEC 61400-1 (IEC 61400-1 Ed. 4, 2019) (IEC 61400-1 Ed. 4, 2019)(1000 shutdown procedures at 4 m/s., 50 shutdown procedures each at 12 m/s and 26 m/s with a duration of 100 s)	30.56 hours (P_{L41} =0.35 %)
Duration of DLC 1.2 (By assuming that the wind turbine operated under DLC 1.2 for the remaining hours in a year)	8699 hours (P_{L12} =99.3 %)

Upon computing the duration of each load cases, the resulting individual lifetime of each load cases are combined with its duration to obtain the total life under the combined loading as similar to Eq. (25) and it is given by,

$$L_{10mT} = \frac{\sum_{k=1}^{N_2} P_{L,k}}{\sum_{k=1}^{N_2} \frac{P_{L,k}}{L_{10mv,k}}} \text{[in years]}, \tag{26}$$

Where, N_2 is the total number of design load cases considered, $P_{L,k}$ is the annual frequency of occurrence of each load, and $L_{10mv,k}$ is the total modified rating life of k^{th} design load case.

Accordingly, the total modified life (L_{10mT}) under the combined loading is obtained using Eq. (26) as,

$$L_{10mT} = \frac{(P_{L31}+P_{L41}+P_{L12})}{\left(\frac{P_{L31}}{223}+\frac{P_{L41}}{39591}+\frac{P_{L12}}{42}\right)} = 42.3 \text{ years}. \tag{27}$$

7. Discussion

Among the four operational load cases (*i.e.,* DLCs 1.2, 2.4, 3.1, 4.1) used for fatigue load assessments, only the results of the mechanical load cases (*i.e.,* DLCs 1.2, 3.1, and 4.1) are presented in this report. The electromechanical load case (DLC 2.4) results will be presented in the Deliverable D5.2.

The influence of different turbulence intensity values and the annual mean wind speeds on the main bearing life is quantified for the DLC 1.2 load cases. The study shows that the total basic rating life (L_{10yt}) variation with respect to the turbulence intensity (I_{ref}) is insignificant. It means that the changes in the basic rating life are only due to the variation in the main bearing aeroelastic loads. On the other hand, the modified rating life (L_{10mt}) changes significantly with respect to the turbulence intensity. This is due to a significant change in the main bearing operational conditions quantified by the life modification factor (a_{ISO}) with respect to I_{ref}. The study revealed that higher the I_{ref}, higher will be the L_{10mt}. However, if there is a change in the turbulence class, then the main bearing lifetime will change significantly as shown in the report. Accordingly, careful consideration needs to be taken while installing the wind turbine in a low turbulence terrain. Also, higher damage is seen around the rated mean wind speeds, and it is highest for the lowest I_{ref}. Hence, by applying the sector curtailment around the rated mean wind speed, one can achieve a higher fatigue life by lowering the DEFL of the main bearing. More detailed analysis is needed to quantify the increase in the fatigue lifetime due to the sector curtailment.

Similar to the turbulence intensity, the influence of the annual mean wind speed on the main bearing fatigue life was also studied in this report. Though the annual mean wind speeds (V_{ave}) do not affect the aeroelastic loads directly, it affects the main bearing fatigue life by means of the frequency of occurrence of individual mean wind speeds. It can bring a maximum change of 5 % in the modified rating life when going from $V_{ave} = 7.5$ m/s to $V_{ave} = 9$ m/s. The influence of the other factors such as ambient temperature, lubricant cleanliness levels and the life for different reliability levels (*i.e.,* different a_1 values) is given in a manuscript submitted to WES journal authored by W. Dheelibun and Asger. If the wind turbine operates at DLC 1.2 throughout its lifetime with its certified design condition (II A), then the main bearing life is coming out to be 42 years.

The results for the transient events such as start-up and normal shutdown are also presented in this report. For DLC 3.1 results revealed that the main bearing is experiencing a higher P_d for $V_h = 26$ m/s, whereas highest P_d always occurs around the rated V_h for DLC 1.2. If the wind turbine operates at DLC 3.1, then the main bearing life is coming out to be 223 years. Similarly, for DLC 4.1, it is almost an infinite life (39591 years).

The main bearing fatigue life under the effects of the combined normal power production and transient loading was also presented. The computed total fatigue life of the main bearing is 42.3 years which is almost the same as that of the fatigue life under DLC 1.2. This shows that the start-up and shutdown procedures have less contribution to the main bearing fatigue life, whereas a significant contribution is resulting from DLC 1.2 due to its larger duration.

8. Conclusion

A model for predicting the lifetime of the main bearing of the Siemens Wind Power SWT-2.3-93 turbine installed in the Teesside offshore wind farm has been created by combining the methodology of the ISO 281 standard with aeroelastic simulations of the bearing loads obtained from a model of the SWT-2.3-93 represented in the HAWC2 aeroelastic code.

If the design wind class of the SWT-2.3-93 is used as input to the model then the modified life time of the main bearing is found to be about 42 years, which is higher than the design life time of 25 years. The analysis has however shown that if the turbulence intensity is reduced from $I_{ref} = 0.16$ to $I_{ref} = 0.1$ then the modified life time is expected to decrease to 22 years, which is lower than the design life time. These changes in main bearing life time can be explained by the fact that the main bearing loads are dominated by the thrust force of the turbine rotor and is peaking at rated wind speed. Decreasing the turbulence is causing more hours spend around rated wind speed compared to a high turbulence scenario, which is moving more operation hours away from rated wind speed.

The model will be used in the Hiperwind project to investigate loss of electrical grid in terms of Low Voltage Ride Through in deliverable 5.2 as well as providing input to the economical work package WP6 of the Hiperwind project.

9. Relation to other work packages of Hiperwind

The main bearing life time model of this deliverable will be used as input for the validation task T5.3, where comparison to the Teesside wind farm SCADA data will be investigated and for the statistical model of task 5.4.

Secondly the Weibull distribution as specified by the life modification factor a1 in chapter 5.1.1will be used to provide input to the Levelized Cost of Energy (LCoE) estimation in Work package 6. This is done by indicating when a certain number of main bearings are expected to fail for the Teesside wind farm according to the life model and then one can determine the expected Operation and Maintenance (O&M) expenses as function of time. It will be very interesting to compare this prediction with the actual status of the main bearings of the Teesside wind farm as part of Task 5.3 as well as following the Teesside wind farm for the next decade.

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11. Appendix Aeroelastic load model for HSS bearing

As mentioned in Section 5, the entire drivetrain is modelled as lumped mass model in HAWC2. Since there is only one main bearing in the drivetrain, it is straightforward to obtain the aeroelastic loads from HAWC2 exactly at the main bearing location. However, the high-speed shaft (HSS) is supported by two bearings as shown in Figure 2. As a result, additional a mathematical model is needed to obtain the loads on these two HSS bearings from HAWC2 and the same will be presented here.

The following assumptions are made while developing the mathematical model:

(i) The HSS is rigid and its weight is negligible.

(ii) The generator coupling does not transmit any loads to the HSS side.

(iii) All the axial loads are carried by the TRB and both the TRB and CRB take the same amount of radial loads.

(iv) Both the bearings are located at an equal distance from the HSS pinion.

(v) No shaft misalignments are considered.

Accordingly, the free body diagram of the HSS is given in Figure 14.

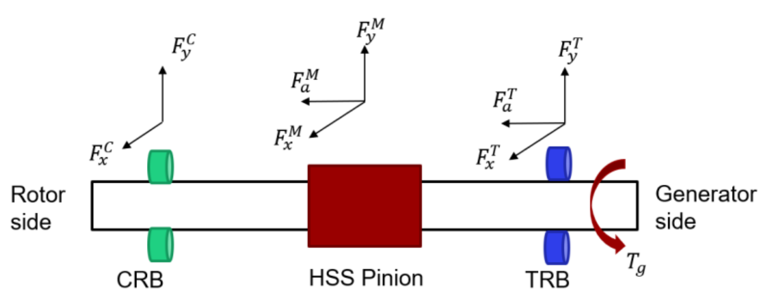

Figure 14: Freebody diagram of HSS.

By following the procedures outlined by Yi and Keller (Yi & Jon, 2018) the forces in the HSS pinion are obtained as,

$$F_y^M = \frac{T_g}{d_H}\cos(\gamma)\,, F_x^M = -\frac{T_g}{d_H}\sin(\gamma)\,, F_a^M = \frac{T_g}{d_H}\tan(\beta), \quad (28)$$

Where, T_g is the generator torque obtained from HAWC2, R is the HSS bearings pitch diameter, β is the angle between the line of action and the Y axis and γ is the helix angle. Typical values of gamma and Beta are ….

By resolving the forces along the radial and axial directions,

$$F_R^C + F_R^T = -F_r^M, \quad (29)$$

$$F_a^T = -F_a^M, \quad (30)$$

Where,

$F_R^C = \sqrt{{F_x^C}^2 + {F_y^C}^2}$, $F_R^T = \sqrt{{F_x^T}^2 + {F_y^T}^2}$, $and\ F_R^M = \sqrt{{F_x^M}^2 + {F_y^M}^2}$.

With the assumption of $F_r^C = F_r^T = F_r$, Eq. (29) becomes, $F_r = -F_r^M/2$. With F_r and F_a^T, the HSS bearings fatigue life can be estimated by following the procedure explained in this report. The outcome of the HSS bearing fatigue life will be presented in the WESC conference.