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Universidad del País Vasco — Euskal Herriko Unibertsitatea

TRABAJO DE FIN DE MASTER

DESIGN OF TEST CYCLES FOR A WIND TURBINE GEARBOX

Diseño de los ciclos de ensayo para la multiplicadora de una turbina eólica

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Resumen

Inglés:

The premise of this project work has been defining the test cycles for the gearbox of a wind turbine that will be tested in the just mentioned nacelle test rig. As it does not exist a standard methodology to test a wind turbine or any of its components, the work described in this thesis pretends to be a structured methodology which can be implemented to test a gearbox of a wind turbine, in order to proof the correct operation of it and determine its critical operating conditions, before it is put in a field to produce power. In total, four different tests have been designed making a total duration of the measurement campaign of the gearbox of 19 days. In those tests, different parameters of the gearbox will be measured and analyzed.

Castellano:

El objetivo de este trabajo ha sido definir los ciclos de ensayo para la caja de engranajes de un aerogenerador, cuyo nacel será probado íntegramente en un banco de ensayos. Como no existe una metodología estándar para ensayar una turbina eólica o cualquiera de sus componentes, el trabajo descrito en esta tesis pretende ser una metodología estructurada que pueda ser implementada para probar una caja de engranajes de un aerogenerador (multiplicadora). Durante el rodaje del nacel en el banco de ensayos, se comprueba su correcto funcionamiento, así como sus condiciones críticas de funcionamiento, antes de ser instalado en su empacamiento final, con el fin de producir energía. En total, han sido diseñados cuatro ciclos de ensayo, con una duración total de la campaña de medida de la multiplicadora de 19 días. En esas pruebas, serán medidos y analizados diferentes parámetros de la multiplicadora.

Euskera:

Lan honen helburua aerogeneradore baten engranajeen entsegu zikloak diseinatzea izan da. Aerogenadorean “nacel”a entsegu banku batean probatuko da. Gaur egun, turbina eoliko oso bat edo haren konponenteak entsaiatzeko metodologia estandarra ez dagoenez, tesis honetan azaldutako lanaren helburua aerogeneradore baten engranajeak entsaiatzeko egituratutako metodogia izatea da. Behin aerogeneradorea, entsegu bankuan instalatuta, bere funtzionamendua eta bere baldintza kritikoak frogatuko dira. Guztira, lau entsegu ziklo desberdin diseinatu dira eta parametro guztien neurketak egiteko eta aztertzeko 19 egun behar izango dira, tesis honen arabera.

Aachen, 10. July 2016

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Master Thesis

for Mrs. Cand.-Ing. Irati Bilbao

Matrikel Number: 365758

Topic: *Design of test cycles for a wind turbine gearbox*

Wind turbines represent an important contribution in the modern energy supply. The design and development of wind turbines is subject to special requirements. The required durability of wind turbines is more than 20 years. The wind loads act in six degrees of freedom, are transient and highly dynamic. Spillover from the electrical grid, also impact on the drivetrain. These external influences have to be considered in the development tools and simulation methods for the design of drivetrain components and wind turbines. Analyzing the interaction between the drivetrain components and the external influences and making usable in the simulation methods, the Center for Wind Power Drives (CWD) of the RWTH Aachen has developed a test bench for wind turbines. The test bench provides the option wind turbine nacelles under real conditions to operate. By simulating wind conditions in combination with a dynamic load unit loads may be imposed by the rotor. The electrical grid is also represented by a real-time simulation, allowing the resulting loads on the drivetrain impart. The system controller is integrated in the test bench control and operates the facility as the real wind turbine operating in the field.

For targeted investigation of wind turbines are testing cycles necessary for one to reflect the real behavior of a wind turbine and other test cycles, which are comparable to component tests. In this master thesis, test cycles for the investigation of a wind turbine on the test bench has to be developed. The focus is on the investigation of the transmission in context of the overall system. For one, the behavior of the transmission system in operation must be examined under real wind and grid conditions. Another includes traditional transmission tests are used to investigate the behavior of the transmission in the system. The results compared to a better understanding with regard to the interactions of the components give to each other and a definition of loads on the component interfaces for the different simulation methods.

In detail the following tasks has to be done:

- Familiarization with the state of the art
- Overview about current test procedures and design process of wind turbines
- Developing test cycles for the gearbox for the HIL Operation
- Developing test cycles for the gearbox for the investigation of the operational behavior
- Documentation of the work

Prof. Dr.-Ing. Christian Brecher

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II Symbols and abbreviations

Symbols	Unit	Description
A	m^2	Rotor swept area
C_p		Power coefficient
D	m	Rotor diameter
E	J	Wind energy
F	N	Force
i		Gear ratio
i_{HSS}		Gear ratio of the High Speed Stage
i_{ISS}		Gear ratio of the Intermediate Speed Stage
i_k''	A	Short circuit current
i_N	A	Nominal current
i_r	A	Rotor current
i_s	A	Stator current
L_M	Ω	Mutual (main) inductance
L_r	Ω	Rotor inductance
L_s	Ω	Stator Inductance
m	kg/s	Air mas per time unit
n	Rpm	Rotatory speed
P	W	Wind Power
R	m	Rotor radius

Symbols	Unit	Description
R_r	Ω	Rotor resistance
R_s	Ω	Stator resistance
T	Nm	Torque
T_r		Rotor time constant
T_s		Stator time constant
T_σ		Total leakage time constant
t	s	Time
t_{dip}	s	Voltage dip duration
v	m/s	Wind speed
v_{ave}	m/s	Average wind speed
v_{hub}	m/s	Wind speed at the hub
v_{in}	m/s	Cut-in wind speed
v_{out}	m/s	Cut-out wind speed
v_r	V	Rotor voltage
v_{rated}	m/s	Rated wind speed
v_s	V	Stator voltage
Z	Ω	Impedance
z1		Teeth number of the pinion
z2		Teeth number of the wheel
ρ	kg/m^3	Air density

Symbols	Unit	Description
η_t		Efficiency of the drivetrain
η_g		Efficiency of the generator
λ		Tip speed ratio
ω	Rad/s	Rotatory speed
Ψ_r	V	Rotor flux linkage
Ψ_s	V	Stator flux linkage
$\Delta\varphi_{HSS}$		Transmission error of the High Speed Stage
$\Delta\varphi_{ISS}$		Transmission error of the Intermediate Speed Stage
φ_{HSS}		Rotation of the High Speed Stage
φ_{ISS}		Rotation of the Intermediate Speed Stage
σ		Leakage factor

Abbreviation	Description
AEP	Annual Energy Production
CAPEX	Capital Investment
CoE	Cost of Energy
CRB	Cylindrical Roller Bearing
CWD	Center for Wind Power Drives
DC	Direct Current

Abbreviation	Description
DFIG	Double Feed Induction Generator
EWM	Extreme Wind Speed Model
EWS	Extreme Wind Shear
FE	Finite Elements
FRT	Fault Ride Through
FVA	Forschungsvereinigung Antriebstechnik e. V
HALT	Highly Accelerated Life Tests
HAWT	Horizontal Axis Wind Turbine
HIL	Hardware In the Loop
HSS	High Speed Shaft/Stage
IEC	International Electro-technical Commission
ISS	Intermediate Speed Shaft/Stage
LSC	Line Side Converter
LSS	Low Speed Shaft/Stage
LVRT	Low Voltage Ride Through
MLC	Measurement Load Case
MSC	Machine Side Converter
NREL	National Renewable Energy Laboratory
NTL	Non Torque Unit
NTM	Normal Turbulent Model

Abbreviation	Description
NWP	Normal Wind Profile Model
OEM	Own Equipment Manufacturer
OPEX	Annual Costs of Operation and Maintenance
O&M	Operation and Maintenance
PCC	Point of Common Coupling
RTDS	Real Time Digital Simulator
RWTH	Rheinisch-Westfälische Technische Hochschule Aachen
R&D	Research and Development
SRB	Spherical Roller Bearing
TRB	Tapered Roller Bearing
VAWT	Vertical Axis Wind Turbine

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1 Introduction

Gearboxes are a type of transmission widely used to provide speed and torque conversions from a rotating source to another machine, using gear ratios. [UICK03], [PAUL79]. Its application ranges from installations in vehicle transmission, aircraft engines, marine drives and wind turbines to name a few. Wind turbine gearboxes are used to increase the low speed of the rotor into a higher speed in order to produce electricity in an efficient way in the generator. Although there is no single reason why wind turbine gearboxes fail prematurely, it continues happening that wind turbine gearboxes tend to fail prematurely more than the gearboxes used in other applications. The relative newness of the industry, the rapid evolution to large size wind turbines and the poor understanding of turbine loads are some of the reasons for the premature failure of wind turbine gearboxes. [MERV16], [BUDN14]

Nowadays, wind energy is the fastest growing type of renewable energy source in the world [MOTA04]. The premature failures of the gearbox and of other components of the wind turbine increases the Operation and Maintenance (O&M) costs, and threfore, the Cost of Energy (CoE) for wind power. In order to make wind power more competitive, there is a need to improve turbine reliability and availability. [SHEN11]

A wind turbine lifetime is normally about 100 million revolutions of the main-shaft over 20 years, but most of the manufactured wind turbines never achieve those 20 years without presenting any problem. The gearbox is the subsystem of the wind turbine that costs more to maintain throughout that expected design life of 20 years of the turbine. So, wind turbine manufacturers are working to boost reliability and availability of the gearbox and so, of the wind turbine. [BUDN14]

However, after the development process, test and certification procedures have to be conducted to ensure a successful product development. Since in-field testing and certification depend on wind and grid conditions, a workshop test rig accelerates the process of product development considerably. Also, in contrast to in-field tests, the reproducibility of measurement is given.

In reality, stochastic and unsteady wind loads influence wind turbines, which results in high forces in the six degrees of freedom of the wind turbine. With simulation models, the impact of the loads and deformations on the wind turbine cannot be described adequately. Even for pure torque, local load distribution has to be verified on test rigs, from idle to overload. For a design optimization of a wind turbine and of its components, which means a better load distribution, an optimal optimization of the material, reduced costs and an increased availability and better knowledge of load distribution is required. [RADN15]

A part from making changes in the design of gearboxes and wind turbines in general, testing those machines can make then more reliable and available. What is more, testing machines (normally called test rigs or test benches) are designed in order to evaluate the behavior of the components when they are under load conditions as well as to analyze and detect the possible failures that can occur in the mechanical components before they are put in their real working in a field. Therefore, it is understood that with a test bench for the whole wind turbine and/or to for components, as gearboxes, it is possible a better understanding about how a wind turbine works as well as check that the mechanical elements will properly perform the function for which they have been designed.

The goal of this master thesis is to design the test cycles for a wind turbine assembled in a nacelle test rig (a test rig in which a complete nacelle is tested), focusing on the gearbox, in order to analyze the behavior of the gearbox under different load conditions and to predict and inspect possible failures of it, and so, increase the reliability and availability of the gearbox and the wind turbine.

2 State of the Art

Wind is the movement of the air across the surface of the Earth, affected by areas of high pressure and low pressure. The surface of the air is heated unevenly by the sun, depending on factors such as the angle of incidence of the sun's rays at the surface, weather the land is open or covered with vegetation, and the atmospheric conditions (clouds, storms, rain). Also large masses of water such as the oceans heat up and cool down slower than the land. During the day, the air above the land heats up more quickly than the air over the water. The warm air over the land expands and rises, and the cooler (and heavier) air moves in to take its place, creating winds. At night, the winds are reversed because the air cools more rapidly over the land than over water. [SPPO16]

Wind energy is the kinetic energy of air in motion. Total wind energy flowing through a given area A (perpendicular to the flow of wind) during a time t is calculated as shown below:

$$E = \frac{1}{2}mv^2 = \frac{1}{2}(Avt\rho)v^2 = \frac{1}{2}At\rho v^3 . \tag{1}$$

E = Wind energy (J)

v = Wind speed (m/s)

ρ = Air density (kg/m^3)

m = Air mass passing per time unit (kg/s)

A = Rotor swept area (m^2)

Power is energy per unit time, so the incident wind power on a given area A (equal to the rotor area of a wind turbine) is:

$$P = \frac{E}{t} = \frac{1}{2}A\rho v^3 . \tag{2}$$

P = Wind power (W)

E = Energy (J)

t = Time (s)

Wind power in an open air stream is thus proportional to the wind turbine effective rotor area and to the third power of the wind speed, so when the wind speeds doubles, the available power increases in eight times. [REVO16]. Therefore, wind energy turbines are requested to provide bigger rotors and to be especially efficient at high wind speeds in order to capture the most of the wind power. Considering the efficiencies for the drivetrain and the generator and Cp as the power coefficient (max=0.59) due to the Betz limit [DNVR02], the wind turbine power capture expression is:

$$P_{WTG} = \frac{1}{2}\eta_t \eta_g \rho C_p v^3 \pi \frac{D^2}{4} \,. \tag{3}$$

η_t = Efficiency of the drivetrain

η_g = Efficiency of the generator

ρ = Air density (kg/m^3)

C_p = Power coefficient

v = Wind speed (m/s)

D = Rotor diameter (m)

Wind development constraints have also demanded the deployment of the latest turbine technology. Optimal wind sites combining excellent wind resources, close proximity to transmission, and local technical support are already populated with wind turbines in mature markets. The need for larger rotors, taller towers and more efficient drivetrains is critical to developing suboptimal wind sites profitably. But these challenges come at a cost: the deployment of carbon-fiber blades, the increased steel usage in 100-meter steel towers, the reliability features of the whole system and the associated operational and maintenance costs. So wind turbine manufacturers face difficult design choices requiring careful costs benefit analysis to properly meet the expectations of asset owners while achieving the financial goals of their stakeholders.

Design philosophies and R&D strategies are varied, but all turbine Own Equipment Manufacturers (OEMs) are focused on strategic components that have the most significant impact on cost of energy (CoE).

2.1 Global market figures

After passing the 50 GW record achieved 7in 2014, the global wind power industry installed 63 GW in 2015, powered by an astonishing 30 GW of new installations in China. This represents a growth of 22% of the annual market. The US market reached 8,6 GW and Germany led a strong performance in Europe with a record of 6 GW of new installations, including more than 2 GW offshore. At the end of 2015, the total global capacity reached 432.5 GW, representing cumulative growth of 17% [GWEC15]. In **Figure 2.1** is shown the global installed wind capacity between the years 2000 and 2015, in MW.

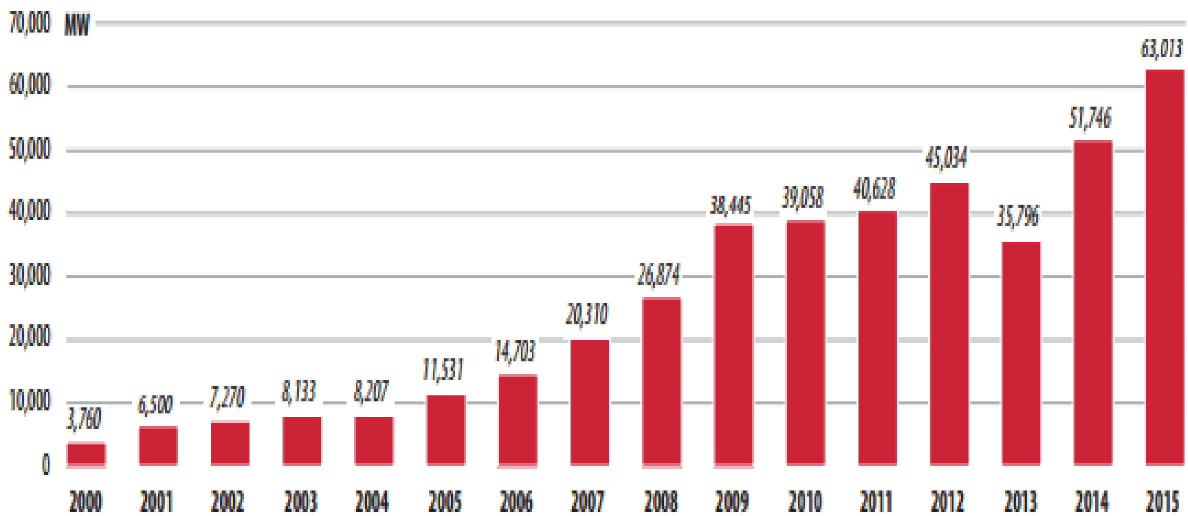


Figure 2.1: Global annual installed wind capacity 2000-2015 (MW)

As a result of its extraordinary annual market, China has edged past the European Union in terms of total installed capacity, with 145.1 GW to the EU's 141.6 GW. Moreover, India registered 2,623 MW pushing past Spain into fourth place in terms of cumulative capacity, after China, the US and Germany. [GWEC15]

Regarding Europe, there are now 16 countries with more than 1 GW installed and 9 countries with more than 5 GW. In the US, more than 5 GW were installed, which resulted in an annual market of 8.6 GW and a cumulative total of 74.4 GW. Despite its economic and political woes, Brazil installed 2.7 GW with a cumulative capacity of 8.7 GW. In addition, the Middle East and Africa was led by South Africa's 483 MW market in 2015, making a total of 1 GW mark in the country. Australia added only 380 MW in new installations to the total of the country of 4 GW. [GWEC15]

In the plot illustrated in **Figure 2.2** is shown the installed annual capacity in MW by regions between the years 2007 and 2015, in which it can be seen the astonishing growth of the Asian countries.

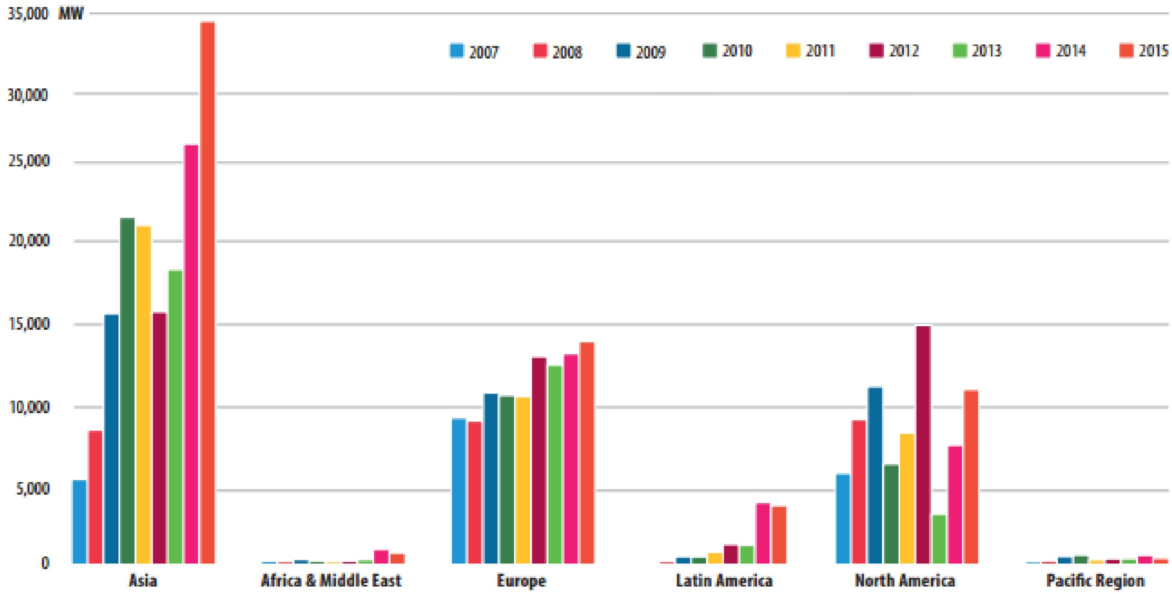
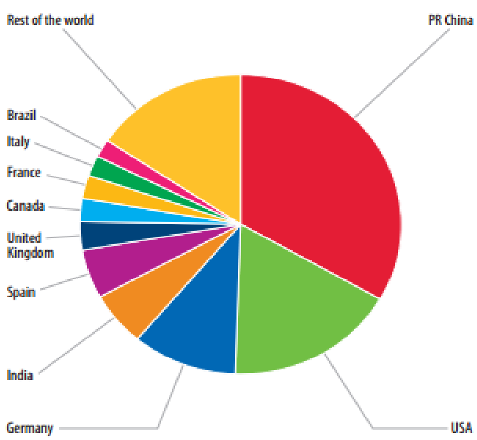


Figure 2.2: Annual installed capacity by regions 2007-2015 (MW)

Moreover, in **Figure 2.3** is shown the top 10 countries in terms of cumulative capacity up to 2015 while in **Figure 2.4** is illustrated the evolution of the wind turbine rotor size and the capacity of the wind turbines.



Country	MW	% Share
PR China**	145,104	33.6
USA	74,471	17.2
Germany	44,947	10.4
India	25,088	5.8
Spain	23,025	5.3
United Kingdom	13,603	3.1
Canada	11,200	2.6
France	10,358	2.4
Italy	8,958	2.1
Brazil	8,715	2.0
Rest of the world	66,951	15.5
Total TOP 10	**365,468**	**84.5**
World Total	**432,419**	**100**

Figure 2.3: Tope 10 cumulative capacity 2015

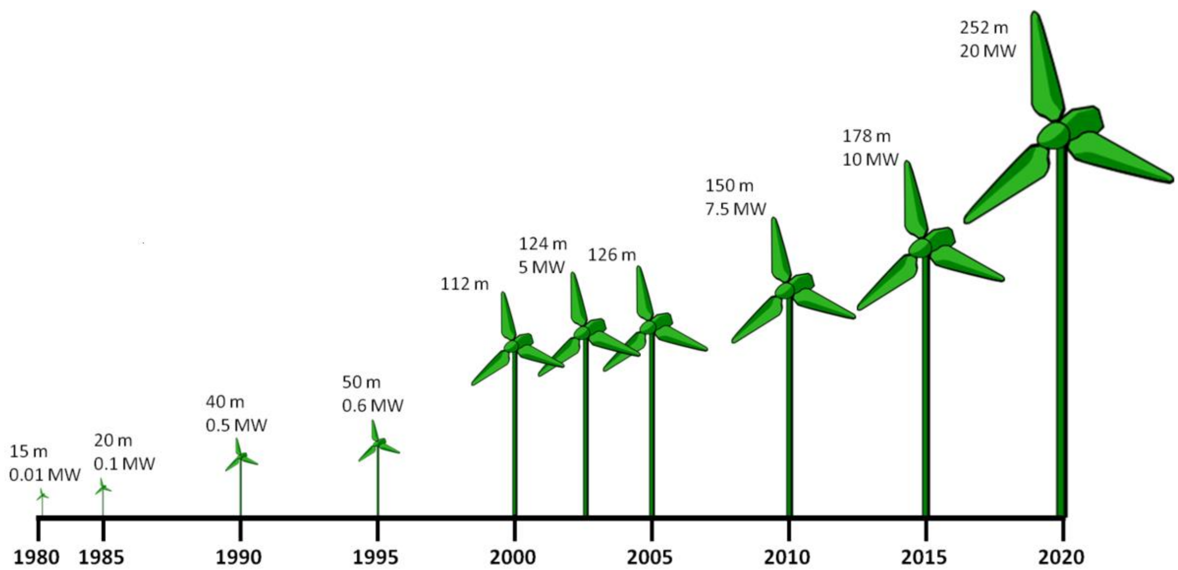


Figure 2.4: WTG rotor size & rated technological evolution

The figure above clearly shows the trend in Europe towards a mainstream of >3MW WTG installations during the last years. This is due to the maturity of the market and the existing grid connection networks in comparison to the rest of the markets (above <2.5 MW).

From the drivetrain and gearbox perspective, the market shows currently a distribution of technologies, organizing the mechanical drive in three categories:

- Direct drive: no gearbox, the rotor hub is connected directly to generator.
- Medium Speed gearbox ratio (1/2 stages)
- High Speed gearbox ratio (3/4 stages)

Globally, the geared systems for wind turbines represent about 80% of the current WTG world fleet.

2.2 Cost of Energy (COE) and Annual Energy Production (AEP)

Wind turbine costs on land (onshore turbines) represents more than half of the total installed cost, whereas for offshore applications, the turbine cost represents only one-third part of the total installed cost of the wind project. These costs, include foundations, electrical grids, operation and maintenance (O&M), installation and staging costs, dominate the system CoE. Turbine improvements that make turbines more reliable, more maintainable, more rugged and larger will still be needed to achieve cost goals. Although, none of these improvements are likely to lower in a significant manner the turbine costs but the net result will lower overall system costs.

2.2.1 Cost of Energy

The cost of energy (COE) is defined as the unit cost to produce energy with a wind power system, expressed in €/kWh. It is accepted by the wind energy research community that the COE can be estimated using the following expression (some small differences can be found between different authors) [MONE13]:

$$COE = \frac{CAPEX + OPEX}{AEP}. \tag{4}$$

CAPEX = Capital investment. Initial capital cost x interests (%); [€]

OPEX = Annual costs of operation and maintenance; [€]

AEP = Annual energy production; [kWh]

The main requirement for wind power is to produce wind energy at the lowest possible cost. To achieve that, capital and operational expenditures must be optimized. Capital expenditure (CAPEX) for an onshore turbine is mainly the cost of the turbine itself plus foundations and electrical infrastructure. Operational expenditure (OPEX) for a wind turbine includes the cost of running the site, planned operation and maintenance and unplanned maintenance due to poor reliability.

The figure on the left shows some factors that have influence on the CoE. Optimizing the drivetrain can significantly decrease both CAPEX and OPEX, improving reliability and availability. Focusing on the CAPEX, the gearbox, generator and power converter make about the 18% of the total cost of the turbine.

Historically, the market has forced wind turbine manufacturers to deliver products with the lowest price (CAPEX). This is now changing, as the industry matures and the operator’s purchasing drivers change from “low cost” to “low CoE”. This is specially happening on offshore turbines.

For any turbine design, it should be possible to improve reliability, and hence reduce OPEX, by increasing the cost of the turbine (CAPEX). For onshore wind turbines, the CoE is largely proportional to the cost of the turbine, which means that an increase in the turbine cost can have a significant negative impact on CoE. So, in order to reduce the CoE, the increase of the CAPEX should be balance by an even larger reduction in OPEX. However, the OPEX is a small portion of the CoE (over 21%), so a large reduction in the OPEX will be difficult to achieve. On the other hand, for offshore turbines the CoE equation changes. In this case, the CAPEX is a smaller portion of the total CoE compared to the OPEX. As a result, an increase on the turbine cost is less significant in offshore turbines; and if it leads to a corresponding increase in reliability, hence reducing OPEX, the total CoE can be reduced.

Normally, an increase in drivetrain costs can be associated with a heavier, larger and stiffer nacelle and to a more expensive main bearing, gearbox or generator. Moreover, it is known, that wind turbine reliability will improve with more advanced and more intensive drivetrain testing. There are being developed some test facilities in which is possible to test the whole drivetrain. Additional testing will have the impact of increasing CAPEX due to the increased development cost, but on the other hand, if successful, it will reduce OPEX and CoE through improved reliability [ZIPP12].

2.2.2 Annual Energy Production

The Annual Energy Production (AEP) indicator of a wind turbine is a key parameter demanded by the owners of the wind farms. The AEP is calculated as the maximum potential energy to be produced considering these factors:

- Specific wind turbine efficiency (capacity factor) and power curve
- Specific site wind conditions: Raileigh or Weibull wind speed distribution, air density, tower height and terrain class.
- Availability factor (function of equipment reliability, associated uptime and maintainability)

In **Figure 2.5** is represented a graphical definition of the concept of the Annual Energy Production.

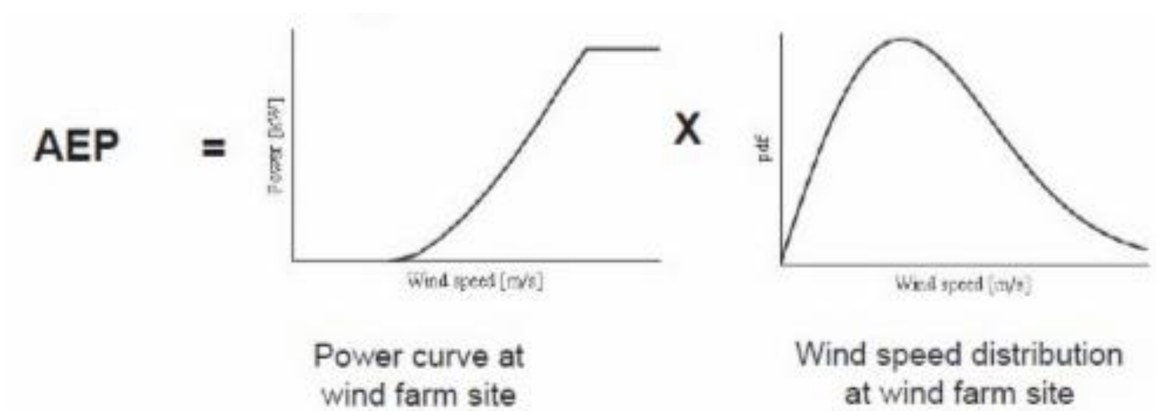


Figure 2.5: Graphical definition of AEP

The capacity factor is the actual annual energy output divided by the theoretical maximum output, if the machine was running at its rated (maximum) power during all of the 8766 hours of the year. This factor theoretically varies from 0-100%, but in practice they will usually range from 20-70%, and mostly be around 25-30%. [TURB16]

Although it seems logical to have preference for a large capacity factor, it may not always be an economic advantage. In a very windy location it may be an advantage to use a larger wind turbine with the same rotor diameter, or a smaller rotor diameter for a given wind turbine size. This would tend to lower the capacity factor (using less of the capacity of a relatively larger generator), but it may mean a substantially larger annual production. Whether it is worthwhile to o for a lower capacity factor with a relatively larger WTG, depends both on wind conditions and on the price of the different turbine models. So, there are two possibilities in order to choose the capacity factor:

- A stable power output, close to the design limit of the wind turbine, with a high capacity factor.
- A high energy output with a low capacity factor.

2.3 Wind turbines

A wind turbine is a rotatory engine that converts kinetic energy from the flow of wind into mechanical energy. If the mechanical energy is used to produce electricity, the machine can be called wind turbine generator (WTG). So, a WTG is a rotatory engine that extracts kinetic energy from the flow of wind and use it to generate electricity. The simplest turbines have one moving part, a rotor assembly, which is a shaft with blades, normally three, attached. Wind energy acts on the blades, or what is the same, the blades react to wind, so that they rotate and transmit energy to the rotor. So, the kinetic energy of the wind turns into mechanical energy of the rotating blades. This mechanical energy is converted into electrical energy in the generator, producing as a result electricity. The generated electricity can be stored in batteries or it can be used directly. [FTEX16], [PESW16]

There are three basic physical laws that govern the quantity of energy that can be extracted from the wind. The first law indicates that the generated energy is proportional to the square of the wind speed. The second one, says that the available energy is directly proportional to the rotor´s swept area and that the extractable energy of the wind is proportional to the square of the length of the blades. The last law, indicates that exists a theoretical maximum efficiency of the wind turbine of 59%, which means that a wind turbine can only extract the 59% of the energy contained in the flow of wind. In practice, most of the wind turbines are much less efficient than that theoretical efficiency. [FTEX16]

Wind turbines can be classified into horizontal axis wind turbines (HAWT) or vertical axis wind turbines (VAWT). The horizontal axis turbines have the main shaft and the drivetrain at the top of a tower while the vertical axis turbines have the main shaft arranged vertically. In the wind industry, the HAWT are the most common ones and its technology has been extensively developed to supply turbines compliant to technical requirements and business economic factors. Moreover, the rotor location can be placed upwind or downwind, being the first one the most used. [CENT16]

Even if different HAWT configurations can be found, all of them contain the following subsystem groups:

- The rotor, including blades, hub and pitching system
- The nacelle, including the main-shaft, bedplate, gearbox, electrical generator, frequency converters, electrical transformers, control system, brake system, yaw system and auxiliary systems
- The tower
- The foundation

In **Figure 2.6** can be seen the subsystems that compose a common horizontal axis wind turbine.

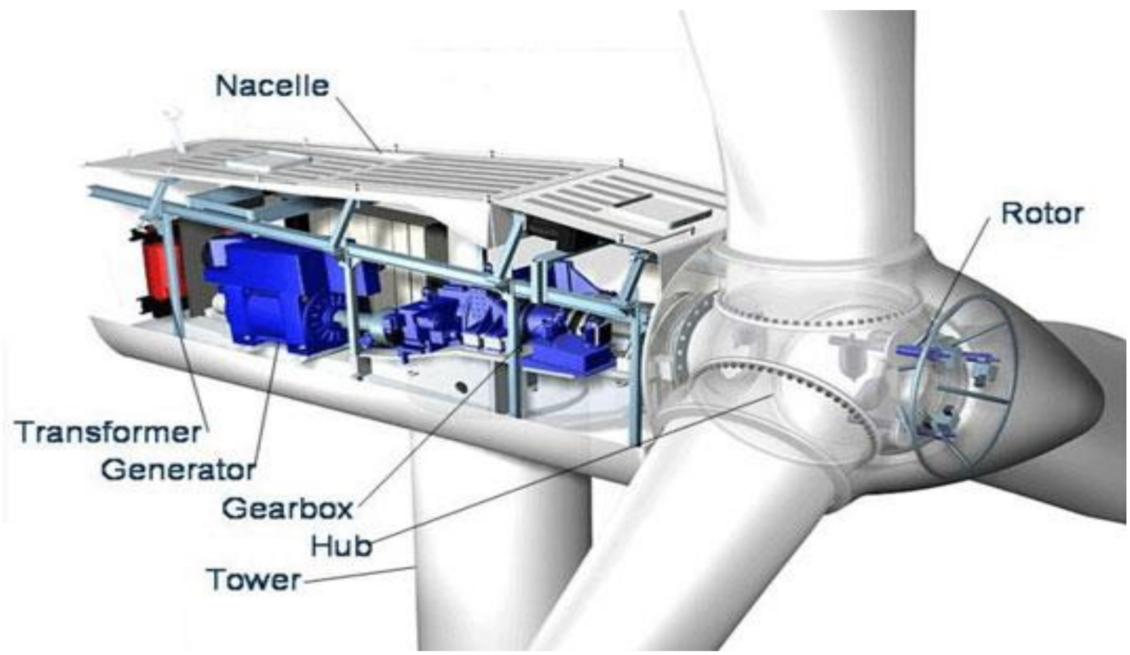


Figure 2.6: Overview of a wind turbine

Different manufacturers use different terms for the same subsystems, but it is only a matter of definition.

WTG are designed to capture the energy of the wind, according to an internal power curve set to nominal conditions named WTG Rated power (e.g.: 4 MW), as it is shown in **Figure 2.7.**

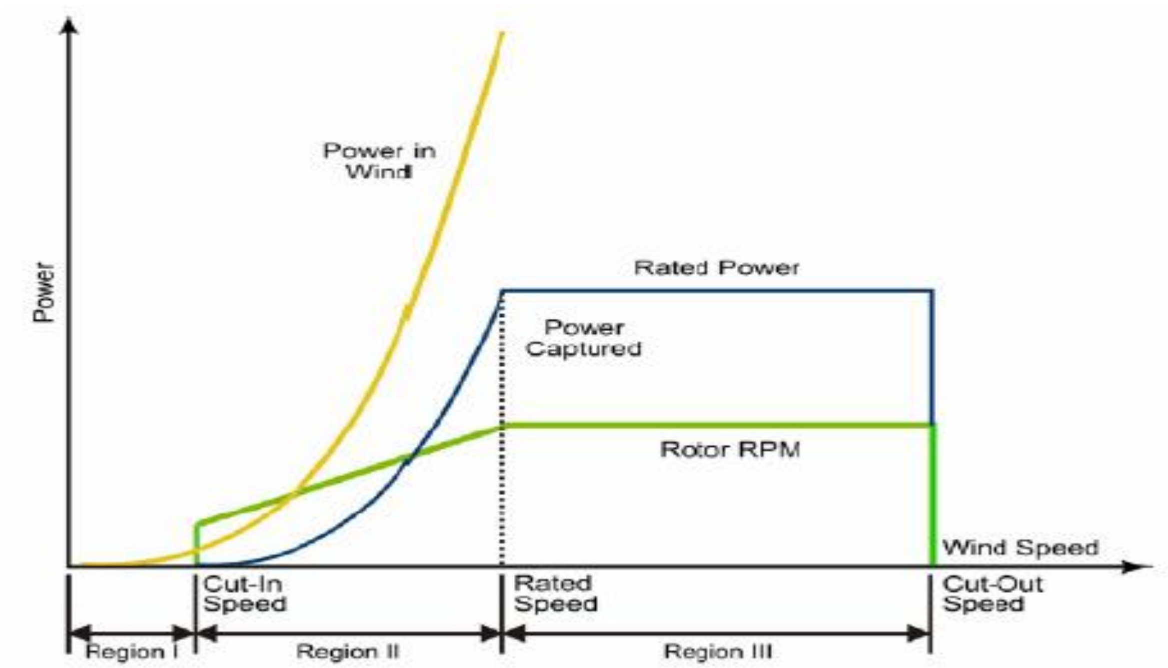


Figure 2.7: WTG power output vs. wind speed curve

In practice, wind turbines are designed to work between certain wind speeds. The lowest speed is called cut-in speed and is generally around 4-5 m/s, and is the speed at which the turbine starts producing electricity. At very low wind speeds, there is insufficient torque exerted by the wind on the turbine blades to make them rotate. However, as the speed increases, the wind turbine will begin to rotate and generate electrical power. When each turbine reaches its rated power (maximum nominal power), the pitch control system begins to limit the power output and prevent generator and drivetrain overload. The maximum operating wind speed is determined by the maximum speed that each wind turbine can support; the control system pitches the blades to stop rotation, feathering the blades to prevent overloads and damage to the turbine's components. This maximum speed is called the cut-out speed and is around 22 m/s to 25 m/s. [WIND16]

2.3.1 WTG control algorithm

Most of multi-MW class wind turbines uses blade pitch and generator torque controllers for power control with variable rotor speed. Pitch control is used mainly to regulate aerodynamic power in high wind speeds while the generator torque controller is used to maximize energy capture in below-rated wind. It is well known that the control action can have a major influence on the loads of the wind turbine. Moreover, the control algorithms have to be designed to play a role of reducing structural loadings of wind turbine. Consequently, it is clear that the algorithms for pitch ad torque controller should be designed carefully [KUSI09].

When it comes to offshore wind turbines, additional effects coming from wave loads to the foundations must be taken into consideration for the accurate control of the system, especially for floating wind turbines which very much affect the structural response and thus, reliability of the wind turbine.

In **Figure 2.8** can be seen the scheme of the control algorithm of a WTG.

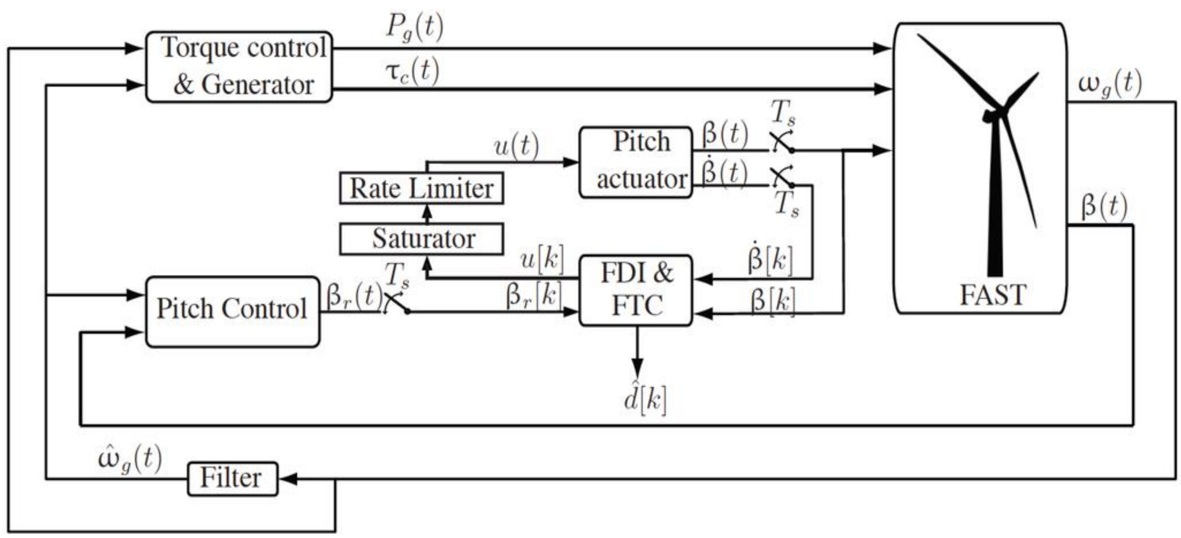


Figure 2.8: Scheme of the control algorithm of a WTG

2.3.2 WTG Drivetrain

As the aim of this thesis is to design the test cycles for the gearbox, the following lines will be about the drivetrain and the different drivetrain configurations that can be found inside the nacelle.

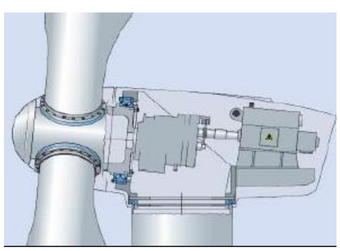
The wind turbine's drivetrain converts the rotation of the rotor into electricity. The most common configuration consists of a Low Speed Shaft (LSS), which is also called main shaft, a gearbox, a High Speed Shaft (HSS) and a generator. The rotor hub is connected to the Low Speed Shaft, which is supported by one or two main bearing. The gearbox steps up the rotational speed of the Low Speed Shaft, with two or three gear stages. The typical gearbox consists on one planetary gear stage and two parallel gear stages. After, a High Speed Shaft transmits the mechanical power of the gearbox to a generator, in which that mechanical power is converted into electrical power. The generator needs high rotational speeds in order to generate electricity in an efficient way. [SHEN11]

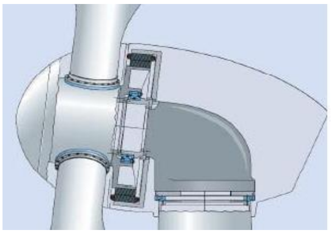
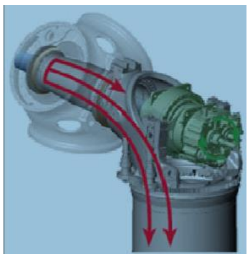
The arrangement of the main shaft and main bearings plays critical role not only determining the efficient non-torque load transfer to the gearbox and/or generator but also in determining the physical size and weight of the components. The conventional four-point and three-point suspension systems that have dominated the modular nacelle architecture in previous years are being replaced by more compact architectures, as turbine OEMs try to reduce the part count and the weight of the nacelle.

In **Table 2.1** can be shown a summary of the different drivetrain configurations that can be assembled inside the nacelle [DINN14], [WIND10].

Table 2.1: Drivetrain configurations

Sketch	Configuration	Key characteristics
	Four-point suspension	• Main shaft with two bearings and gearbox torque arms • SRB+SRB or DTRB+CRB • Smaller diameters, preloaded bearings to limit axial shaft movement • Requires precise mounting

	Three-point suspension	Main shaft with single bearing and gearbox torque arms • SRB • Smaller bearing • Design coordination to ensure that gearbox does not receive axial load • Allows for greater flexibility and misalignment • Flexibility tolerates slight mounting errors
	Two-point suspension	• Tow bearings with gearbox suspended on housing • TRB+TRB • Medium to large bore bearings • Heavy structural support required • Gearbox and generator as one structure
	Direct coupling	• Single bearing integrated into hub and gearbox • TRB or DTRB • Large bore diameters • Expensive • Used when extreme stiffness required • Able to take on radial and axial loads • DTRB configuration generally employs a small stub shaft

	Direct drive	• No main-shaft • No gearbox
	Pure torque concept	• Main-shaft decouples non-torque loads towards gearbox

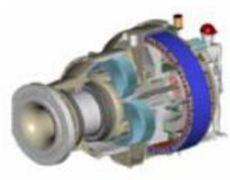
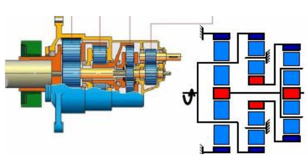
SRB=Spherical Roller Bearing, TRB=Tapered Roller Bearing, DTRB=Double Row Tapered Bearing, CRB=Cylindrical Roller Bearing

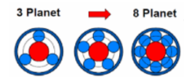
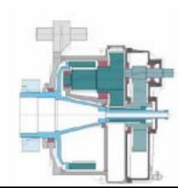
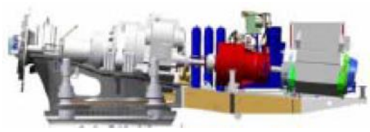
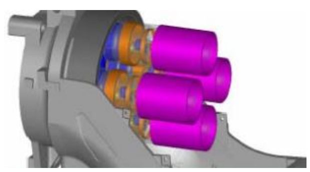
2.3.3 Wind turbine gearbox

The gearbox is used to increase the rotational speed from the Los Speed Shaft (LSS) to a higher speed, in order to generate electricity in an efficient way. The design of the gearbox presents many challenges due to the loading and the operational conditions in which the gearbox must work. In addition to the torque from the rotor that generates power the turbine rotor also introduces large moments and forces to the wind turbine drivetrain. It is essential to ensure that the drivetrain isolates the gearbox in an efficient way or ensure that the gearbox is able to support all the designed loads; otherwise internal gearbox components can become seriously misaligned, which can lead to stress concentrations and thus, failures. Furthermore, drivetrains suffer severe transient loading during start-ups, shut-downs, emergency stops and also during grid connections.

Considering commercially available gearbox configurations in the market place today, the classification showed in **Table 2.2** for the wind turbine gearbox can be done.

Table 2.2 Gearbox configurations

Sketch	Configuration	Key characteristics
	Three-stage gearbox	• Conventional gearbox with either single planetary + two parallel stages or two planetary + single parallel stages • Significant industry experience with failure modes
	Two-stage gearbox	• Two-stage planetary, typically with outputs of 400 rpm, compact drivetrain • Limited weight reduction achieved as low-weight third stage is removed
	One-stage gearbox	• Single-stage gearbox, compact drivetrain • Lighter standalone component, but lower gear ratio, leading to heavier generator • Generator flanged to gearbox or sharing same housing
	Torque split architecture	• Equal load distribution on planetary stage • Demonstrates lower weights compared to other configurations
	Flex pin architecture	• Uniform loading similar to torque split, however with slight variation

		• Lighter configuration  compared to conventional designs
	Compound architecture	
	Mechanical-hydraulic conversion	• Voith's WindDrive addition to driveline • Synchronous generator
	Gearbox with several outputs	• Simple generators ease of replace • Complex gearbox internal topology

Bellow, are described the most common configurations that have been mentioned in the tables above [WIND10]:

1. **BASIC CONFIGURATION:** This configuration consists on a planetary-helicoidally three stages gearbox connected by the High Speed Shaft at 1200 rpm to an induction generator of six pols with wound rotor. It is called basic configuration because it is the most extended one between wind turbine manufacturers.
2. **DIRECT DRIVE CONFIGURATION:** Some wind turbines generate electricity by coupling the main shaft directly to the generator. This configuration requires larger diameter generator and power electronics to allow variable speed operation. The goal of using this configuration is simplifying the nacelle and avoid the problems that gearboxes cause, as the main failures that happen in a wind turbine are regarding the gearbox. The generator is synchronous with permanent magnets. Comparing to the basic configuration, this one is more simple, lighter and cheaper to manufacture. However, direct drive configuration has not yet a sizable market share.
3. **ONE STAGE GEARBOX CONFIGURATION:** This configuration consists on a stage gearbox and a moderated speed (100 rpm) generator based on

permanents magnets. The main advantage of this configuration is that the generator has a smaller size.

4. **MULTIPATH:** Between all the possible multipath configurations, the most extended one is the configuration based on a one stage gearbox and six medium speed (100 rpm) generators of permanent magnets.

2.3.4 Gearbox design and calculation

Each gearbox OEM has its own development procedures, but all methodologies have a common approach from a design and calculation perspective which can be summarized as it is shown in **Figure 2.9**

The design and calculation cycle includes: preparation of loads, dimensioning of the gearbox, optimization of the technical solutions identified, verification of the done calculation, validation of the designs, detailed design and drafting of complete bill of materials and generation of certification reports.

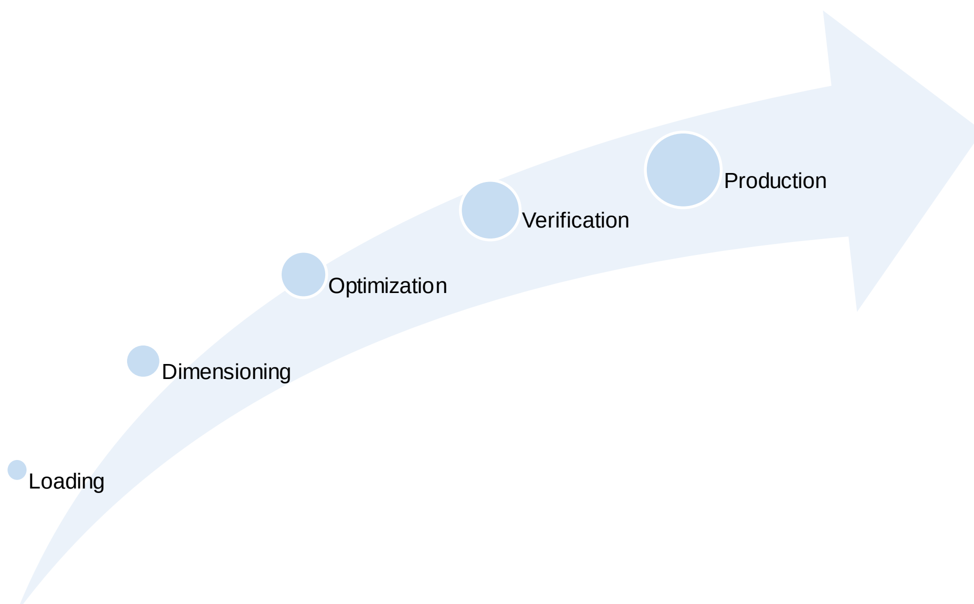


Figure 2.9 Gearbox design cycle

The unique and main function of the gearbox is transmitting the mechanical torque and adapting the speed from the rotor to the generator. To achieve this, four main functional systems can be identified:

- Torque transmission chain system: usually implemented by means of gears, shafts, bearings and structural elements.

- Protection system and gearbox enclosure: seals, housings, covers, coatings.
- Non torque loads distribution to mainframe system: structural elements and connecting devices to the nacelle basement.
- Thermal conditioning: filters, coolant piping, pump elements, heaters and heat dissipation elements.

2.4 Loads and conditions in a life cycle

The loads and stresses to which wind turbines are subjected to are very specific and complex. Because of the nature of the wind, the loads are highly variable and they require special attention in the design, operation and maintenance of the wind turbines. In order to avoid wind turbine's catastrophic failure, and understanding of the loadings on them as well as the turbine's response to them is required. [HAU13]

Varying loads are more difficult to handle than static loads because the material becomes fatigued. Because of the low density of the air flow, a large surface is required in order to capture the energy; and as the dimensions of the rotor increase, the dimensions of other components (tower height, for example) should increase also, delivering into more elastic structures. Moreover, the changing loads induce vibrations and resonances that can produce high dynamic load components. [HAU13]

Three aspects must be taken into account in the structural design of a wind turbine:

1. Components must be designed for the extreme loads that can appear, which means that the turbine must be able to withstand the highest wind speeds which may occur.
2. The fatigue life of the components must be guaranteed for their service life, which should be over 20 or 30 years.
3. The components must have enough stiffness to withstand vibrations and critical deflections.

The starting point for the load spectrum of a wind turbine are the loads acting on the rotor blades, which are passed on to the other components. The loads coming directly from downstream components are less significant compared to the ones acting on the rotor. Therefore, loads acting on a wind turbine can be concentrated on the rotor and work with it as being representative of all parts.

The effects of aerodynamic, gravitational and inertial forces, cause all the forces acting on the rotor. Hau, in his book Wind Turbines, makes a classification of the different loads and stresses according to their effect with time on the rotating rotor [HAU13]:

- Aerodynamic loads with a uniform, steady wind speed and centrifugal forces, generate time-independent steady-state loads, as long as the rotor speed is constant.
- A steady air flow, but spatially non-uniform over the rotor swept area causes cyclic load changes on the rotor. Particularly, this includes the uneven flow towards the rotor due to the increase in wind speed with the height, a cross flow towards the rotor and the interference due to flow around the tower.
- The inertia forces due to the dead weight of the rotor blades cause periodic and unsteady loads. When the rotor yaws, gyroscopic forces are caused, which must be also taken into account.
- Apart from the steady-state and cyclically changing loads, the rotor is subjected to non-periodic, stochastic loads caused by the wind turbulence.

In order to determine structural stresses, the effects of load variations with time must be considered. Moreover, fluctuating and alternating loads must be identified, especially with respect to the fatigue life of the structure.
In **Figure 2.10**, on the left can be seen the effect of aerodynamic, gravitational and inertial loads on the rotor of a horizontal-axis wind turbine while on the right is represented the co-ordinates and technical terms for representing loads and stresses on the rotor.

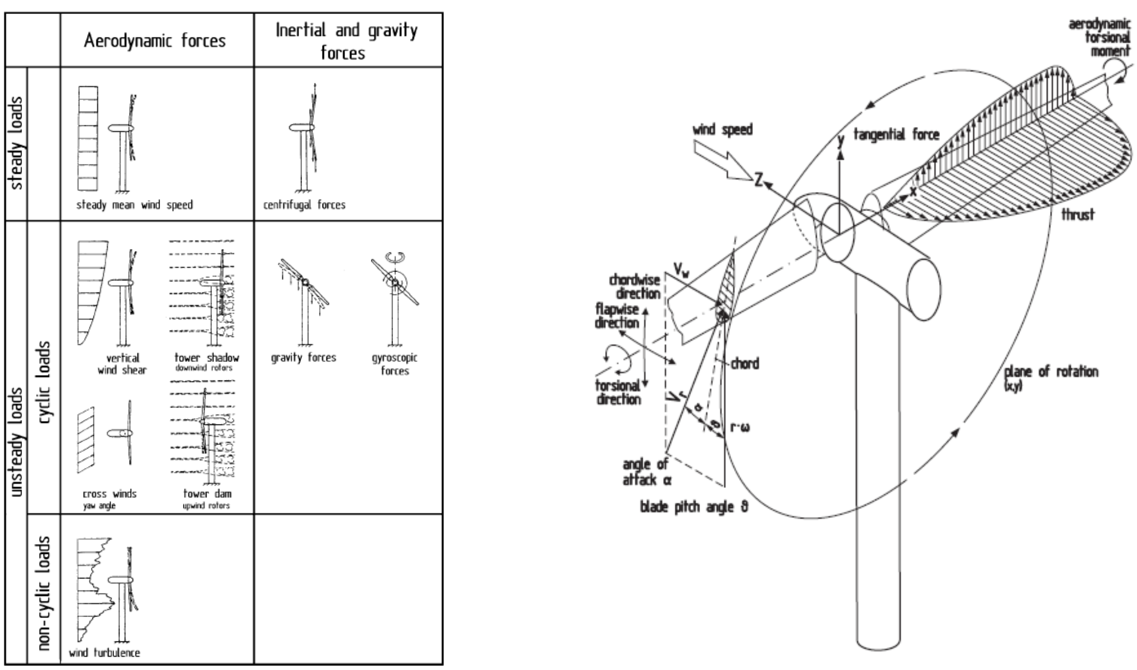


Figure 2.10: Wind turbine loads

In addition, it is not possible to recognize in advance which loads will be dominant within the complete range of loads.

Once the reasons that cause the different loads are known, it is compulsory to identify the conditions during which the wind turbine is subjected to the decisive loads. These conditions are recorded in the form of load-cases. It is important to remark that in the defined load cases the determined loads by calculations are always load assumptions, which deviate from the real loads. Nevertheless, these deviations from the real loads must always be to “the safe side”, which means that the load assumptions made for the design must always be higher than the expected loads in operation. [HAU13]

The conditions for the causes of the load situation, such as wind speed and the operation parameters of the turbine, must be contained in the load cases. [HAU13]

1. **NORMAL OPERATION:** The loads that act in a wind turbine under “normal” operation conditions are mainly relevant to fatigue life. The basis for the definition of load cases are the characteristic wind speeds which are used in the sequence of operational states. The extreme loads appear mostly under extreme wind speeds.
    i. <u>Power production:</u> A load case is formed for each characteristic wind speed (cut-in speed, partial-load wind speed, rated wind speed, full-load wind speed and cut-out wind speed). During power production, the asymmetrical conditions for the rotor and the random wind speed fluctuations must be considered.
    ii. <u>Start-up and shut-down of the rotor:</u> During the start-up and shut-down of the turbine special load cases and load changes take place. These events are frequent during the life of the turbine, so they have an influence on fatigue life. In large wind turbine, the normal shut-down is controlled by means of blade pitch control, as the rotational speed varies, so special loads are not involved. There is one exception, which is the fast braking (emergency shut-down), where the reversed aerodynamic thrust can cause increased loads.
    iii. <u>Parked rotor at extreme wind speeds:</u> It happens when the rotor is parked and the wind turbine has to deal with high wind speeds. In the case of turbines with pitch control, it is assumed that the rotor blades are in the feathered position and that the rotor is aligned with the wind speed. So, in this case the load level is much lower than under cross-wind conditions. For turbines with fixed-blades this does not occur and the strength of the turbine must be verified under cross-wind conditions. Furthermore, some wind turbines do not stop at extreme high wind speeds, as at rotor idling the loads on the turbine are lower compared to fixed parked rotor (using an appropriate blade pitch angle).
2. **TECHNICAL FAULTS:** Technical faults and defect can subject the wind turbine to additional loads that have not been contemplated in the other load cases. Normally,

technical problems lead to an emergency stop of the rotor with a safety system, so they do not result in any extraordinary loads. However, before the shut-down of the rotor, some malfunctions can happen, causing abnormal loads on the structure.

i. Rotor emergency stop: Even if most of technical failures will force the rotor to an emergency stop via the safety circuit, the sudden deceleration of the rotor results in an extraordinary loading situation for the turbine. In turbines with large rotors, this situation can increase the bending stress on the rotor blades up to the strength limit.
ii. Control system fault: In turbines with blade pitch control, a failure in the control system can lead to an inappropriate pitch angle for the operating condition. This is directly related to special aerodynamic loads and indirectly to other consequences, such as rotor over-speed.
iii. Generator short-circuit: A short circuit in the electrical generator causes an extreme load on the drive train. The generator short-circuit torque can be increased up to seven time the rated torque value.
iv. Rotor over-speed: Defects in the blade pitch control or a sudden loss of the electrical load (e.g.: power system shut-down) can lead to an over-speed in the rotor. Therefore, a large safety margin between the permissible operating speed and the maximum speed before fracture is required.

2.5 Drivetrain and gearbox failure types

The development of component test procedures requires the knowledge of the damages itself and the damage mechanisms occurring in practice as well as the damage-relevant loads. One unfortunate characteristic of the wind turbine industry is the lack of clarity about the root causes of the WTG downtimes and failures [TAVN10], and the continuous blame on the drivetrain and gearbox systems poor reliability as the main origin of the electricity generation stoppage.

But as the wind industry is gathering more experience along the years of wind farm exploitation, new data is being collected and analyzed, which shows that the so called gearbox problem is not the real problem but a consequence of a list of causes related to other subsystems. The gearbox is acting as the mechanical fuse of the whole wind turbine, so for instance if wrong wind turbine control loop is performed, the gearbox will undertake extra loads which may cause a fatal damage leading to breakage of critical elements such as gears or bearings.

Reliawind program in EU [TAVN10] and Gearbox Reliability Collaborative [LINK11] in USA were researches activities which concluded that the gearbox was not the system with the higher failure rates. However, it was reported that the gearbox was the subsystem of the WTG that more time requires for replacement, thus, provoking more WTG downtime.

Within the gearbox, the failure rates are distributed in general as follows [TAVN10]:

- Bearings: 66% out of which:
  - 50% high speed stage (HSS)
  - 25% intermediate speed stage (ISS)
  - 15% planet bearings
  - 10% low speed stage (LSS)
- Gears: 15 % mostly the sun gears in planetary stages and Intermediate and High Speed Shafts
- Structural parts: less than 4%

In general lines, gearboxes and bearings can be affected by faults for the following main reasons:

- Underestimated design loads
- Torque overloads
- Wrong material
- Manufacturing errors
- Damage during transportation and assembly
- Misalignment of components in the shaft

The maturation of the gearbox-design process that currently is used un the wind-turbine industry has reduced the gearbox failures originated from gear failures. The main driving failure mechanism in the gearboxes nowadays is surface-fatigue failure of the bearings, and the debris produced by this failure mechanism causes the abrasion of other components of the gearbox. There are characteristic loading conditions that drive this surface-fatigue failure mechanism in the bearing, including misalignment and reverse axial loading. Other parameters that influence the failure of bearings can include the bearing fitting either loosely or tightly with the housing, which can result in fretting.

Nevertheless, gear failures also occur independently of bearing failures, although not as commonly. The most common gear failures in the industry are wear failures, caused by a poor lubrication that normally appear in the planets of the planetary stage because of their low rotatory speed; abrasion and surface fatigue initiated by the debris created from bearing failures. Furthermore, tooth interference, poor tooth load distribution produced by misalignment and changes in center distances due to lack of gearbox housing compliances can also occur in the gears.

2.5.1 Failure modes for gears

Below are listed and explained the main failure modes observed in the gears within a wind turbine gearbox [OYAG09]:

- **Wear:** is a surface phenomenon whereby metal is removed or worn away more or less uniformly from the contacting surfaces of the gear. This phenomenon is highly dependent on lubrication (oil-film thickness and oil cleanliness). Also, the ground surface roughness of the tooth flanks and the sliding velocity play an important role.
- **Abrasion:** is caused by particles suspended in the oil film, which hardness is near to or even higher than the hardness of the gear. This phenomenon appears as small grooves that are carved outwardly from the axis of the gear. It can be prevented by using a filtering system which eliminates larger particles from the system.
- **Surface fatigue:** it is caused by the loading and unloading of the tooth face. This failure appears when the endurance limit of material is exceeded under the surface of the tooth. This type of failure is not normally catastrophic during its early stages, but it typically progresses and ends in catastrophic failure, depending on the loading conditions.
- **Micropitting:** it is characterized by the creation of small pits (0.38 – 0.76 mm) on the surface of the tooth. The failure appears at localized overstressed areas and subsequently is followed by a redistribution of the load. The subsequent operation of the gear generally results in the polishing of the pitted area, improving its appearance. In many cases, if the tooth surface geometry is disrupted by the initial micropitting, then the pitting progresses and evolves into macropitting and tooth breakage.
- **Macropitting:** is a progression of micropitting and results in the destruction of the tooth profile. It is usually a consequence of poor load distribution and high Hertzian stress. Its appearance is similar to micropitting, but it is larger and has a more irregular shape.
- **Spalling:** is similar to macropitting except that the damage is quite extensive, more irregular and quite shallow. Materials of medium hardness are more prominent to this type of failure, but it can occur also in materials with a high hardness that support high loads.
- **Scuffing:** is defined as localized damage caused by the occurrence of solid phase welding between sliding surfaces, so material transfer takes place between the surfaces of the teeth. This failure appears when the existing lubricant film is not enough and permits metal-to-metal contact between gear teeth.
- **Fracture:** It appears as a consequence of fatigue, overload or tooth bending. It usually results in catastrophic failure and it propagation is quicker as the other failure

mechanism mentioned above. This type of failure leads in the immediate loss of serviceability and usually occurs with little or none warning.

- **Fretting corrosion:** is a special wear process which occurs at the contact area between two materials under load and subject to minute relative motion by vibration or some other force.

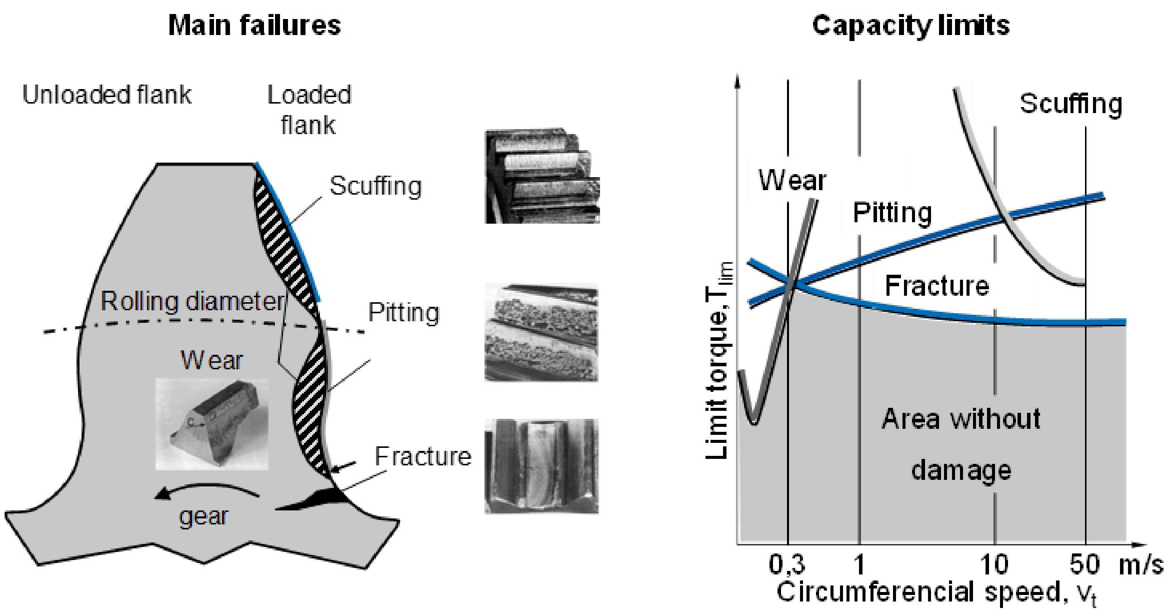


Figure 2.11: Gear failures upon service conditions

2.5.2 Failure modes for bearings

The failure mechanisms of bearings are quite similar to the gear failure mechanism, and it can be classified into two categories: lubrication failures and surface fatigue failures. [OYAG09]

- **Surface fatigue failure:** normally this type of failure is progressive and can be distinguished: micropitting, macropitting and spalling. The raceway is usually the first one that fails comparing to the other components of the bearing under fatigue surface. The bearings give an audible indication of pitting, and moreover, pitting increases the system's vibration. If pitting progresses can lead to spalling or fracture of the rolling elements.

- **Lubrication failure:** is the result of the lack of the required lubricant film thickness that is needed in order to avoid metal-to-metal contact of the rolling elements and the raceways. This, causes an overheating of the bearing. Overheating results in the loss of the bearing's material hardness. The symptom that typically appears is discoloration of the roller elements, rings, and cages. Moreover, high temperatures can also degrade or destroy the lubricant film. In wind turbines, many of the bearings work at low rotatory speeds, which induces wear due to a loss of the lubricant film thickness or complete loss of elastohydrodynamic suspension. However, this type of failure mechanism is not as catastrophic and does not progress as rapidly as overheating.

2.6 Design and validation of wind turbines by IEC 61400

The IEC 61400 is and International Standard regarding wind turbines published by the International Electrotechnical Commission (IEC). The scope of IEC is to promote international cooperation on all questions regarding the standardization in the electrical and electronical fields. The IEC 61400 is a set of design requirements made to ensure that wind turbines are properly engineered against damage within the planned lifetime. The standard contemplates most aspects of the turbine life, from the external conditions to turbine components being tested, assembled and operated [WIKI16].

Regarding wind turbine gearboxes, the most interesting parts of the IEC 61400 are the part 4, which deals with the design and requirements for wind turbine gearboxes and the part 13, which is about loads measurements.

The IEC-61400-4 describes the minimum requirements for specification, design and verification of gearboxes in wind turbines, installed onshore or offshore.

The objective of designing a gearbox is to achieve an adequate high availability and reliability which enables low maintenance and low repair cost throughout the design life, which should be at least the same as for the wind turbine (at least 20 years). [IEC12]

Designing a drivetrain for a wind turbine is an iterative process, in which are involved the turbine manufacturer, the gearbox manufacturer, the bearing manufacturer and the lubricant suppliers. During the design process each of them should be involved in a critical system analysis in order to identify design assumptions which strongly influence the correct design, manufacture and operation of the gearbox. [IEC12]

The first step for de design of a WTG includes the selection of the rotor design, the drivetrain configuration, the nominal power and torque and the definition of the control system. After, all the relevant design load cases (DLC) that affect to the wind turbine must be calculated, which are defined in the IEC 61400-1. Once the different load cases are defined, time series simulations for many conditions must be done. With the loads and operational characteristics defined and specified, the initial gearbox design can be started, which would lead to one or more prototype gearboxes. The prototype gearbox, shall be intensively tested in a test rig. The test specification is based on the design loads as well as a design validation taking into account the assumptions made during the design. The results of the workshop test must be summarized in a report. Followed by a test on a workshop, a field test must be done. If the obtained results during both tests are not the expected ones, the design must be redone and the design process will start again. Otherwise, If the results of both tests are acceptable, a test specification for the standard serial production test of the gearbox type can then be defined. [IEC12]

Furthermore, the IEC 61400-4 defines also the minimum requirements for verification testing of new gearbox designs. Planning the tests shall include the turbine manufacturer, the lubricant supplier and designer and the gearbox manufacturer. The test plan for the gearbox must be formed by a prototype test of the gearbox as a unit, some tests of the gearbox as an integrated part of the wind turbine and requirements for acceptance testing in serial production. Below are mentioned the specific tests that this standard says that must be done in order to validate a gearbox [IEC12]:

- Workshop prototype tests:
    - Component testing
    - Gearbox prototype testing
    - Lubricant system testing
- Field tests:
    - Validation of loads
    - Type test of gearbox in wind turbine
- Production tests:
    - Acceptance testing
    - Sound emission testing
    - Vibration testing
    - Lubrication system considerations
    - System temperatures
- Robustness tests
- Field lubricant temperature ad cleanliness
- Bearing specific validation

The standard says that all phases of verification testing must be largely documented and evaluated by means of reports [IEC12].

On the other hand, the IEC 61400-13 deals with mechanical load measurements on wind turbines. The goal of this chapter of the standard is describing the methodology and techniques for the experimental determination of the mechanical loading on wind turbines, as is the case of a test rig. The measurement program involves collecting both a statistical database and a set of time series, which define the behavior of the turbine in certain specific situations. In the chapter, a system of measurement load cases (MLCs) is defines to determine the wind turbine loads in conditions corresponding to a selection of design load cases (DLCs) of the IEC 61400-1. These MLCs can be used for documentation of the load in relation to the DLCs or can be used as a basis for the validation of calculation models at specific and well defined external conditions. Subsequently, the models can be used to estimate the loads at the design conditions. [IEC12]

Moreover, the IEC 61400-13 says that a capture matrix should be used during the measurement campaign, which is used to organize the measured time series. The capture matrix has two objectives: it can be used as a guideline for programming the data acquisition system or it can be used as a tool to decide when the measurement requirements are fulfilled. [IEC12]

2.7 Test cycles for wind turbines and wind turbine gearboxes in field and on test rigs

Improving reliability and availability of wind turbines is the scope of testing wind turbines as well as testing its subcomponents. In order to succeed, wind manufacturers must produce systems and subcomponents with reliable performance and low maintenance. The load on components, combined with the environmental conditions, require robust requirements for the engineering of main system and sub-systems, as well as their lubrication (particularly the gearbox). A full understanding of the interaction between the turbine and the gearbox is compulsory and must be demonstrated. Multi-body simulations, FE approaches and the validation on test rigs and on field are all part of the gearbox development. The IEC 61400-4 states the recommendations for test criteria, sensors and sensor position. [KRUL14]

Furthermore, testing is essential to understand and predict the system behavior of a wind turbine. As the turbines that are manufactured nowadays have high nominal powers, the lifetime of operating at relevant power levels hardly achieve the required service of 20 years.

The gearbox is the most complex subsystem within the drivetrain, this is why it is crucial to create an accurate simulation model that must be validated. A test rig helps to validate the gearbox simulation models and to get a better understanding of the gearbox's dynamics. [KRUL14]

Tests carried out on test rigs have the advantage of providing a correlation between set loads and the responses of the test objects under reproducible conditions. They are appropriate whenever unknown material properties, the interaction of different materials in a specific design, uncertainties concerning manufacturing techniques or even the verification of calculated results are to be investigated. The loads themselves must, however, be predetermined, thus, the tests must be based on the assumption that the loads are correct [HAU13].

As it can be shown in the test pyramid illustrated in **Figure 2.13** for a wind turbine, there are different test-levels for a wind turbine; starting with a material test for the different components of the turbine and going up to a product test, in which the wind turbine is tested in the field.

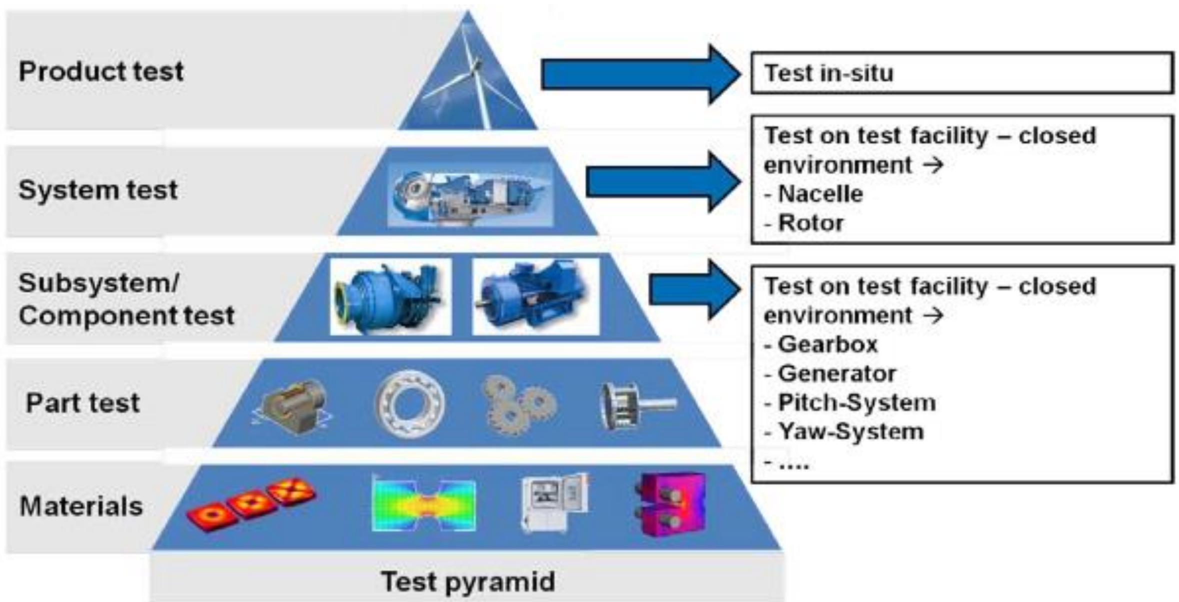
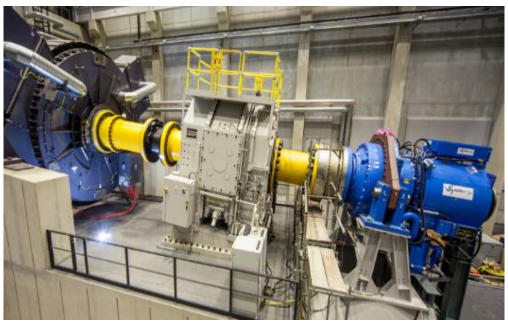
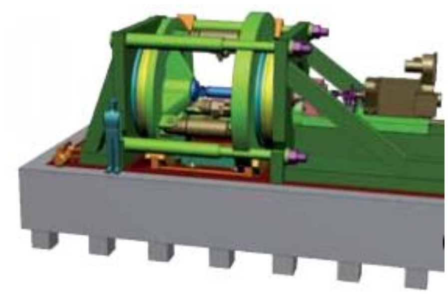


Figure 2.12: Test pyramid

In **Table 2.3** are shown the tests than can be done for the different subsystems of a wind turbine [RENK16].

Table 2.3: Types of test rigs

Sketch	Test type	Specifications
	Drivetrain/nacelle test rig	• Testing of drivetrains and complete nacelles • Reproduction of real dynamic movements of the drivetrain under wind load conditions • Analysis of components within an integrated system • Verification of overall system behavior • Reproduction of events recorded in field measurements for problem solving under laboratory conditions • Optional wind load simulation using a dynamic 6 DOF loading mechanism • Used in development, type approval and production level testing • Standard measurements: temperature, noise, vibrations…

	Main rotor bearing test rig	• Testing of main rotor bearings with dynamic wind load simulation • Used in development and type approval tests • Analysis of rolling element behavior under dynamic bearing loads • Lubricant analysis under extreme conditions • Testing in a back-to-back arrangement • Standard measurements: temperature, noise, vibrations…
	Rotor blade test rig	• Blade fatigue testing • Blade static testing • Blade material testing • Dual axis static or fatigue testing • Lightning protection testing (pending design) • Prototype development and blade repair capabilities
	Generator test rig	• Testing generator-converter combinations • Used in development and approval test • Reproduction of electrical mains pollution • Standard measurements: temperature, noise, vibrations…

	Gearbox test rig	• Testing gearboxes in back-to-back configuration or in a dynamometer test rig • Used in development, type approval and production-level tests • Load application optional (mechanically or electrically) • Wear pattern check after full load and overload • Lubricant analysis under extreme conditions • Standard measurements: temperature, noise, vibrations…

An example of a drivetrain/nacelle test rig can be found in the Center for Wind Power Drives (CWD), in Aachen (Germany). The CWD institute of Aachen (Germany) coordinating its research works on wind energy with other seven institutes from the RWTH Aachen university, has developed a 4 MW nacelle test rig for wind turbines. Before, a 1 MW experimental setup was fully set-up and tested as a proof. The 4 MW test rig has optimized multi-physics Hardware in the Loop and is capable to offer 6 degrees of freedom in the testing. [CWDR16]. Furthermore, The SCE&G Energy Innovation Center of the Clemson University (USA) has developed the most advanced wind turbine drivetrain facility of the world, which is capable of full scale highly accelerated mechanical and electrical testing of wind turbine drivetrains. They have two different test rigs. The first one allows to test complete nacelles up to 15 MW while the other test is capable to handle nacelles and gearboxes up to 7.5 MW. [CLEM16], [BUSH12].

However, the National Renewable Energy Laboratory NREL) placed in the United States, offers research and analysis of wind turbine components and prototypes from 400 W up to 3 MW. They own different types of test facilities: dynamometer tests facilities, field tests facilities and structural testing laboratories. They also have blade testing facilities. [NREL16]

Moreover, wind turbine and gearbox manufacturers, such as Gamesa [GAME16], Vestas [DVOR13], etc., also have their own test facilities.

Testing standards were made to ensure that accredited third-party laboratories are conducting tests consistently. These tests reveal many design and manufacturing deficiencies that cannot be detected by analytical tools. Furthermore, they also provide the final verification that the design process has worked. Thorough full testing shortens the time from when the gearbox exists “on paper” to the day it is assembled in the wind turbine and producing energy.

Different levels of testing are considered for wind turbine gearbox validation:

- **Level 1 - Gearbox subcomponents level:** Laboratory testing and gearbox subsystems validation by means of ad-hoc test rigs and methods. Usually conducted by the components suppliers, not the gearbox manufacturer.
- **Level 2 – Gearbox system level:** The gearbox functionality is tested using full scale test rigs, normally in back-to-back configuration.
- **Level 3 – Drivetrain system level:** For some drivetrain concepts it is not possible to test the gearbox isolated, so a full drivetrain test is to be performed (e.g.: integrated design for main-shaft and gearbox).
- **Level 4 – Nacelle or wind turbine level:** Final validation tests of gearbox features and integration requirements into the wind turbine under real load conditions at field.

For the levels 2 and 3, the most time consuming is the endurance test that age the components of the drivetrain under a combination of high loads and speeds. In some cases, the test can be focused to the gears, in others to the bearings, couplings, frame, and so; the approach is different.

There are two main topologies of gearbox test stands for full load testing, which is required by the mind turbine manufacturers and certification bodies:

- **Dynamometer test rig:** The scope of this type of testing is to verify the performance and reliability of wind turbine drivetrain prototypes. During the test, operating field conditions are simulated in a workshop environment. Normally, a powerful motor replaces the rotor and the blades [NREL16]. Dynamometer testing focuses on the mechanical and electrical power-producing systems (generator, gearbox, frequency converters, bearings, brakes, lubrication and cooling system and control system). There is an effective way to validate new designs, as the wind input into the turbine can be simulated without waiting for nature events to occur. During a steady-state testing, different points along the turbine’s power curve are chosen by the dynamometer for a fixed period of time, in order to evaluate the mechanical and electrical performance. Nevertheless, some points outside the turbine’s normal operating range can be chosen in order to proof the gearbox behavior at extreme conditions or to conduct Highly Accelerated Life Tests (HALT). Overloading intentionally a turbine in a HALT test, a lifetime of wear and fracture is applied during a period of time. Furthermore, in order to get a better understanding of a turbine’s mechanical, electrical, and control system response in real conditions (as if the gearbox was in the field), the called “hardware-in-the-loop” techniques are being used to replace the rotor, tower, pitch and yaw systems with computer simulations operating in real time. The dynamometer test rig in the preferred configuration for test laboratories and wind turbine integrators if the drivetrain configuration is compact [MUSI00].

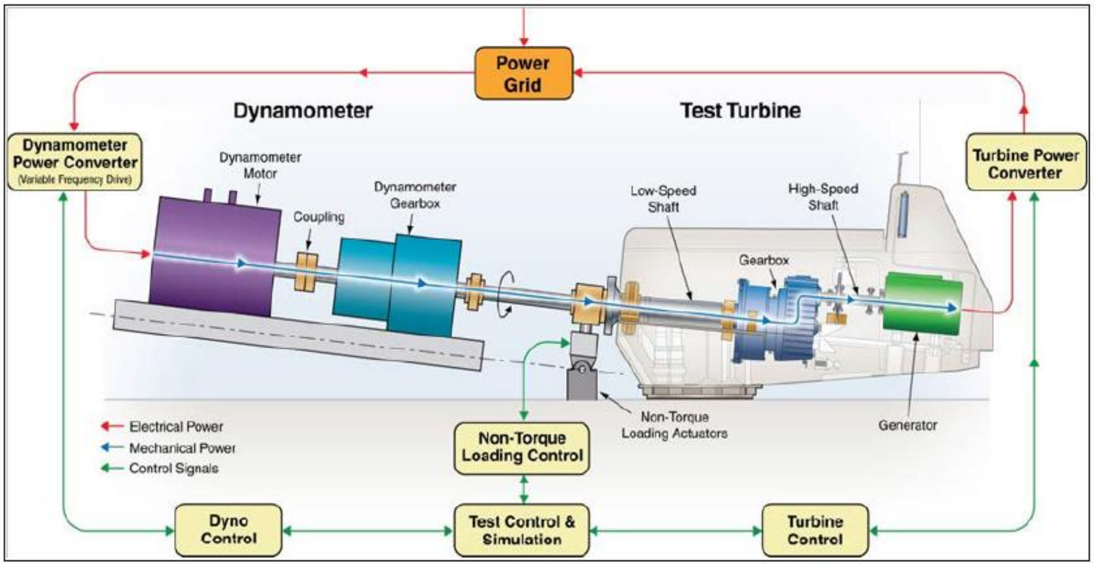


Figure 2.13: Configuration of a dynamometer test rig

- **Back-to-back test rig:** This configuration consists on one motor, who is the one in charge of providing the torque, two identical gearboxes and one generator, in order to recirculate the energy used during the test. One of the gearbox is called the slave, who is the one to be tested while the other gearbox is called the master. In most of the rigs, the torque is applied under static circumstances. The left side, composed by the motor and one of the gearboxes, represents the wind and the rotor; while the right side, composed by the other gearbox and the generator, represents the grid and the generator itself. The back-to-back set up is the typical configuration for gearbox manufacturers and for modular drivetrain concepts [HELS10]. There are three essential types of tests performed in a back-to-back set up:
    - Dynamic tests on load, load distribution and efficiency
    - Structure-borne noise and air borne noise tests and measurements
    - Lubrication tests: oil distribution, leak tightness and climate chamber tests

  Normally, load tests are done in order to check gearbox behavior; the gearbox is therefore typically tested at overload and according to the specification. Then, the gearbox is dismantled and every single part is inspected after all this tests are done. In **Figure 2.14** is shown a photo of a typical back-to-back test rig arrangement.

Figure 2.14: Back to back test rig

In order to obtain an accurate validation of the gearbox model, large analysis and verifications of the gearbox within a test rig environment are necessary. The quality of the gearbox model has a significant impact on the quality of the simulation within the wind turbine model. Closing engineering loops supported by multi-body simulation results will help to enhance the suitability of the gearbox design and its reliability. In a wind turbine, loads and temperatures vary depending on the turbine's speed conditions. Non-stationary loads and speeds occur when loads high control activities take place under turbulent and strong wind conditions. The design and suitability of support systems, bearings and lubrication are relevant for reliable component lubrication. Furthermore, there has to be an agreement between the gearbox functionality and the environmental conditions [KRUL14]:

- High speed, low loads: smearing has to be avoided
- High speed, high loads: scuffing must be avoided
- Low speed, high loads: micropitting and wear have to be avoided
- Stand still: still-standing marks and "false brinelling" must be avoided
- Idling: supply of toothings and bearings above the oil sump are necessary
- No grid: minimum supply of components during parking periods are needed
- Cold climate: is necessary to maintain the lubrication and filtration system and study the pattern-contacts, materials, mechanical fracture and fatigue under these conditions
- Frozen gearbox: the gearbox has to be able to start the pressure-fed lubrication, to avoid coal oil at heating elements, to start in a short period of time and brittle crack propagation has to be studied.

The operating conditions mentioned above are not the unique ones that should be addressed by testing, calculation and simulation.

Regarding the tests that must be done for a wind turbine gearbox, as part of the design process, there can be distinguished eight different test categories [KRUL14]:

1. **Prototype tests on a gearbox test rig:** These tests are made in order to check the assumptions made in design, to obtain important parameters for the execution of series tests during the production of a WTG. These tests also need to be done to cover the confirmation of the functionality of some parts of the gearbox, the fulfillment of noise and vibration criteria, the temperature limits and lubrication criteria.
2. **Robustness tests on a gearbox test rig:** It is related to the fatigue of the gearbox and should expose its weakest link. This test can be done as an accelerated fatigue test and/or as an overload test.
3. **Cold climate tests in climate chamber:** This tests are done in order to confirm the functionality of the lubrication system under extreme low temperatures and also to verify the start-up and warm-up strategies of the gearbox up to the minimum operation temperature.
4. **Component and sub-system test:** Component and sub-system tests confirm design assumptions and verify calculation results. For example, the lubrication system must be tested.
5. **Material tests (strength tests, quality tests, etc**.)**:** These tests qualify properties of brittle cast iron and temperature, depending on the properties of the steels used for the teeth of the gears. Moreover, quality tests are necessary to assure specific material properties with respect to chemical consistency, grain sizes and metal structure.
6. **Field tests of the gearbox:** After prototype tests of the gearbox at a test rig environment, a prototype qualification and validation on-site is required. During a field test, only relevant items for the gearbox as part of the wind turbine are tested. Validation of the gearbox within the actual turbine environment in the field is vital in confirming design assumption. The measurement of the following aspects is essential for the design evaluation:
   - Confirmation of design assumptions: deflections, loads, bending, movements…
   - Acoustic and vibrational behavior
   - Validation of simulation models and results
7. **Field tests of the gearbox oil:** Field tests for lubrications have the aim to check the compatibility of the lubricant with all the components of the gearbox under field environment conditions.
8. **Acceptance tests for series production on the test rig of the gearbox manufacturer:** Series and acceptance tests are required as end-of-line tests for each gearbox that will be delivered to the customer. These tests have to be operated with specific load steps up to nominal load under consideration of requirements to

test criteria, such as temperatures, pressure limits, noise and vibration limits and contact patterns of the teeth.

In **Figure 2.15** is shown a plot which links the development process of the validation of a wind turbine with the costs due to design changes.

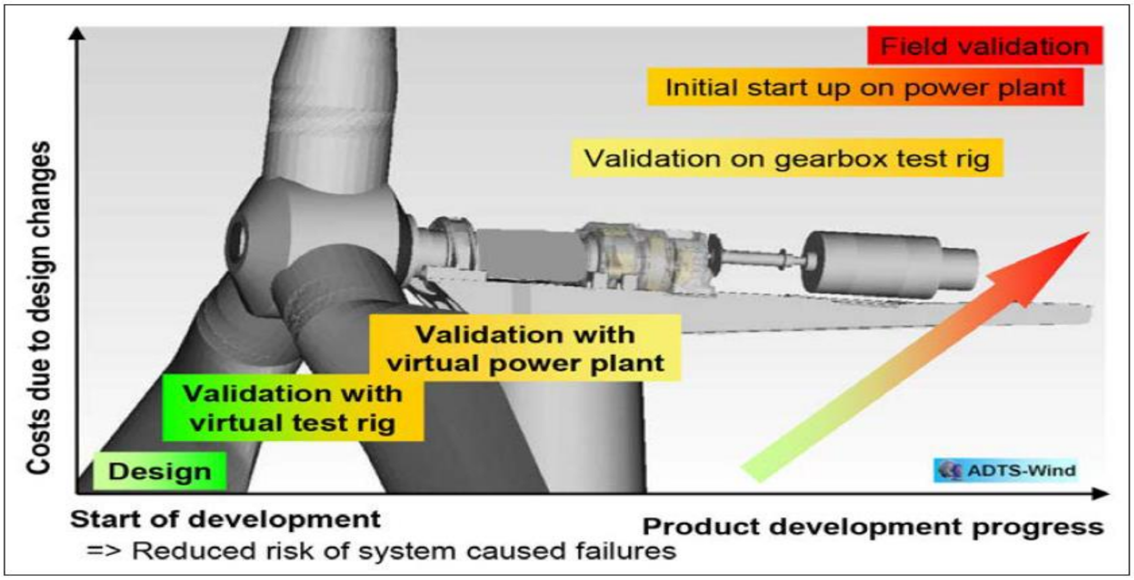


Figure 2.15: Testing vs. costs

2.8 Conclusion

As it has been mentioned along this document, the importance of wind power as a renewable source of energy is rapidly increasing and manufacturers of this industry have to face big challenges because of the need to develop quickly and efficiently save and reliable products. On the other hand, one key factor in the successful deployment of wind turbines is their certification. Testing wind turbines and their subsystems allow the manufactures to know how their products will behave before they are put it in a field, this is, testing allows a better understanding of the load distribution on the wind turbine and it is also necessary in order to validate and certificate new designs of the wind turbine and of its components.

Field testing of wind turbines for their validation, is very time consuming and cost intensive. On the other hand, full size wind turbine test rigs allow a realistic operation of wind turbines in a workshop environment. Due to the independency of wind and grid conditions, the cost and duration of the test program and certification can be reduced. Moreover, as the local load conditions are affected by the wind loads, the grid loads and the operational behavior of the nacelle controller, new test rig infrastructures and test procedures are required for measuring the local load conditions in the drivetrain.

Summarizing, testing a gearbox and so a wind turbine is essential, apart for validation, in order to achieve a high reliability and availability for minimizing the downtime and loss of power generation, which leads to lower maintenances and thus to lower a lower CoE.

3 Objective

In the wind industry, it does not exist a standard methodology to test a wind turbine or any of its components. The IEC 61400 standard is a guideline for the certification of installed wind turbines or their single components. It does not include measurement of entire nacelles or drivetrains on system test benches. Accordingly, there is no standard guideline available these days. To stablish a basis for a general comparability it is desirable to have standardized test procedures that can be operated by independent test laboratories. This is why the aim of this thesis work is to develop a structured methodology which can be implemented to test a wind turbine drivetrain, or more specifically the gearbox, in a nacelle test rig in order to proof the correct operation of it and determine its critical operating conditions, before it is put in a field to produce power. The methodology that will be described along this thesis does not pretend to be the best one or the unique one, but a structured one that can be used whenever a wind turbine drivetrain must be tested in a test rig.

A wind turbine gearbox is normally tested in a back-to-back arrangement. Nevertheless, in the testing methodology that will be described along this document the gearbox will be only tested inside the nacelle, in a nacelle test rig. So, another aim of this thesis work is to be able to test exhaustively the behavior of the gearbox in a nacelle test rig, without being necessary a previous testing in a back-to-back arrangement. So, it could be said that with the possible testing of the gearbox as part of the drivetrain it would not be necessary to test the gearbox in a subcomponent testing level.

Furthermore, as the local load conditions are affected by wind loads and the grid loads, new infrastructure and test procedures are required for measuring the local load conditions in the drivetrain. This is why the worldwide demand for system tests of wind turbine nacelles is steadily increasing. These test benches are used for function tests and for fatigue tests of entire Wind Turbine nacelles [BOSS13]. Actually, there is a lack of experience in nacelle test rigs, so another aim of this thesis work is to get a better understanding in this type of test rigs, as it is believed that a nacelle test rig provides more useful data during the testing of the specimen under test than any other type of test rig, as it is possible to analyze the interferences between the rotor and the generator in the gearbox. Moreover, nowadays the different subsystems that constitute the drivetrain of a WTG are designed and manufactured by different manufacturers, so it is necessary a test rig that integrates all the subsystems to analyze the interactions between them and the influence of each of them in the other ones before they are put in the field.

When a wind turbine or any of its components is tested in a test rig, the main goal is to proof their functionality before they are put into their real service. Besides, in the test cycles that will be defined and described in this thesis, there will be also taken into account the wind and grid behavior while testing the specimen under test, which means that there will be defined some specific tests that will test the drivetrain behavior, and so the gearbox, with real operating conditions. On the other hand, the specimen under test will be tested for different load conditions (up to overload) and also for special load conditions in order to test its functionality and to improve its reliability and availability.

During the testing, different measurement positions must be defined inside the gearbox in order to know what happens inside the gearbox. Each of those positions must have a specific location and a particular aim in order to measure in an efficient way the physical parameters that have to be controlled and measured during the different tests. Once all the measurements positions are defined and stablished, the measured parameters must be obtained with the data acquisition system and consequently analyzed. The evaluated measured parameters will be compared to the different simulation methods done in during the gearbox design conception.

4 Definition of Test Object

4.1 FVA Gondel

The Center for Wind Power Drives (CWD) in Aachen is immersed in the FVA-Gondel research project, which goals are validating simulation methods for wind turbines on a 4 MW system test rig and analyzing and reducing local stresses in a wind turbine drivetrain. It is believed that reliable simulation models and an adequate validation are necessary in order to prevent premature failures and to increase the availability and cost effectiveness of WTG. The major challenges of wind turbine generator concepts, such as extreme torque, complex load changes in all six degree of freedom, permanently unsteady operation, etc... are often insufficiently taken into account in existing simulation models. [RWTH16]

Within the scope of this project, the reliability of the simulation models will increase through the validation by a full size research wind turbine generator in a nacelle test rig. The vital factor of this approach is the systematic and reproducible adjustment of various operating and loading conditions with comprehensive measurement instrumentation of the WTG under unique laboratory conditions. [RWTH16]

The research project is based on the approach to modify a wind turbine nacelle with current range of 2,75 MW so that all technical details of the system and also the results of the research can comprehensively be disclosed. The wind turbine nacelle under test will be tested in a 4 MW nacelle test rig in the Center for Wind Power Drives (CWD) in Aachen, Germany. This test rig can realistically simulate and reproduce wind fields and grid conditions. Moreover, in this test rig can be tested nacelles with a maximal length of 18 m and a rotor hub height of 4,5 m. The implemented multi-physics hardware-in-the-loop (HIL) concept reproduces realistic stresses in all 6 degree of freedom on the device under test as it would experience in the field. The mechanical stress caused by the wind field as well as the electrical interaction of the device under test with the grid are emulated. In addition, critical test scenarios such as Fault Ride Through (FRT) tests can be performed consuming less time and cost. [AVER15]

To emulate the wind loads in all 6 degree of freedom, the test bench consists of a 4 MW direct drive and a Non Torque Load (NTL) unit. The load application in the NTL is realized with six pairs of pre-stressed hydraulic actuators that are located on a spinning journal bearing disc and a radial bushing. With the NTL it is possible to apply the emulated wind load in all 6 degree of freedom and to realize maximum bending moments of about 7.2 MNm and maximum forces of 3.3 MN. The NTL unit is capable of emulating the dynamic thrust, tilt and yaw bending loads and the static rotor weight [AVER15].

On the grid side, a converter system controlled by a Real Time Digital Simulator (RTDS) emulates the behavior of the electrical grid for normal power production and grid faults. The direct drive has a nominal speed of 14 rpm and a maximum speed of 30 rpm. From 4 to 14 rpm it provides a nominal torque of 2.7 MNm while its maximum torque is 3.4 MNm. [AVER15]

The scheme of the 4 MW test rig can be shown in **Figure 4.1**.

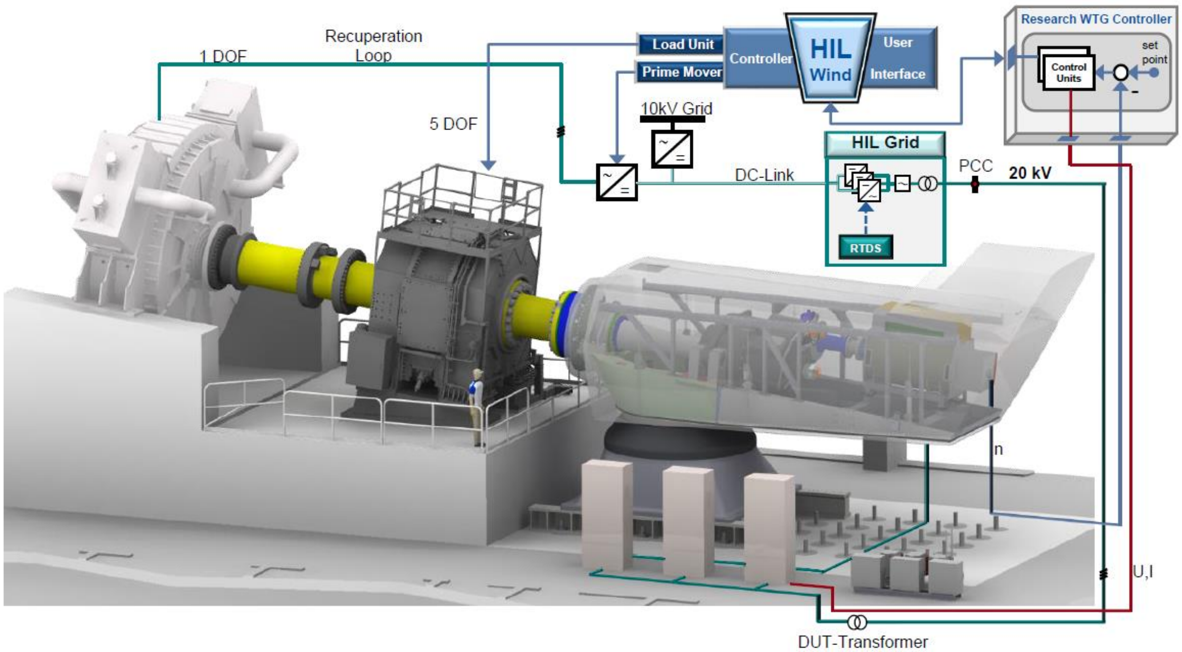


Figure 4.1: Scheme of the 4MW nacelle test rig

4.1.1 Device under test

The device under test for the definition of the test cycles in the 4 MW test rig is the NEG Micon NM80-2750, that has been on service in a field for 10 years. The nominal power of this wind turbine is 2750 kW, with a nominal speed or the rotor of 17,5 rpm. The hub height is 60 m and the rotor diameter is 80 m.

In **Table 4.1** is given the main data of the turbine under test.

Table 4.1: Technical data of the turbine

Technical Data NM80-2750	
Nominal Power	2750 kW
Rotor Diameter	80 m
Hub height	60 m
Nominal rotor speed	17,5 rpm

As it is represented in **Figure 4.2**, some modifications have been done in the gearbox, the generator, the lubricant and cooling systems and in the control of the nacelle in order to achieve a better functionality of the wind turbine.

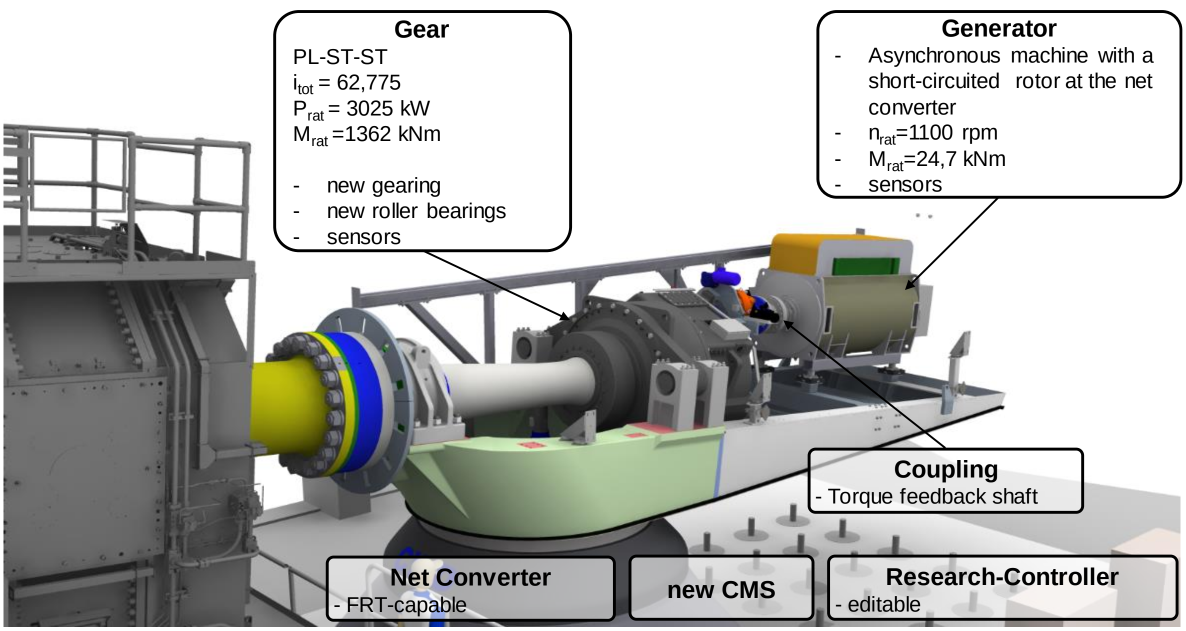


Figure 4.2: Modifications of the nacelle

The original gearbox assembled in the nacelle is a triple stage hybrid-design, consisting of one helical geared planetary stage and two helical geared parallel axis stages. This gearbox was specially design for the loads in a wind turbine with triple-point suspension, which is the configuration of the NM80-2750. The total gearbox ratio, which is 1:63, was selected such that the blade tip speed was optimized at the selected generator reference speed, which is 1100 rpm. This gearbox can provide a nominal power of 2750 kW. The low speed shaft (LSS) in the gearbox was implemented as a hollow shaft. It encloses the rear end of the rotor shaft and the connection is by a strong clip bushing. This special connection between the rotor shaft and gearbox, forming a single unit in a three-point suspension prevents alignment problems. But, as it has been said before, in order to get a higher functionality, the original gearbox has been disassembled form the nacelle and a new one has been designed and afterwards assembled in the nacelle. The new gearbox, also a triple stage design consisting of one helical geared planetary stage and two helical geared parallel axis stages, has a gear ratio of 62.775 and can provide a nominal power of 3025 kW and a nominal torque of 1362 KNm. Furthermore, gears of both stages as well as bearings and sensors have been redesigned and changed and some modifications have been done for the Intermediate Speed Shaft (ISS) and High Speed Shaft (HSS) for the design and manufacture of new gearbox.

Regarding the generator, the original one installed in the nacelle was a 6-poled generator with a nominal speed of 1100 rpm. One of the advantages of a 6 poled generator, compared to a 4-poled generator, is the reduced stress on the bearings. Form wound windings are used in both the rotor and stator and the generator is designed to withstand the electrical stress from frequency converter operation. But the new generator, an asynchronous with 6-poled machine with a short-circuited rotor at the net converter and nominal speed of 1100 rpm, has been developed in order to reduce local stress loads. The nominal power of this new generator is of 2750 kW but with a nominal current of 2564 A instead the original one of 1518 A.

In **Figure 4.3** can be seen the operating range in terms of wind speeds for the nacelle under test, where v_{in} is the cut in speed (speed at which the turbine starts up), v_{ave} is the average speed, v_r is the rated speed (speed referred to nominal conditions of the turbine) and v_{out} is the cut out speed (speed at which the turbine shuts down).

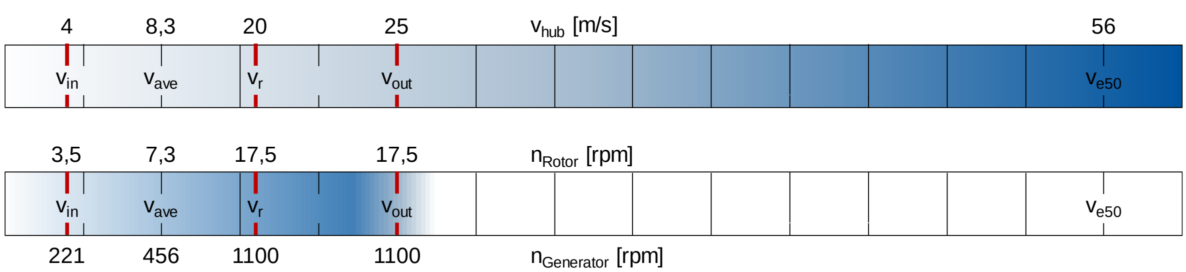


Figure 4.3 Wind speed operating range for the nacelle

4.2 Measurement points in Gearbox and Nacelle

In order to validate the designed model of the gearbox, some measurement techniques must be developed before testing. The main goal of this, is the detection of the dynamic behavior and load-induced deviation of the transmission as a result of rotor loads and feedbacks from the electrical network. Three different categories will be distinguished for the measurement:

- Global measurement of the gearbox
- Gear measurement
- Bearing measurement

4.2.1 Global measurement

The aim of the global measurement is validating the Multi-Body-Simulation model in terms of vibration and displacement. In order to achieve it, structure-borne noise sensors on the housing will be installed and relative deviation to the machine frame, loads of the NTL, torque in the High Speed Shaft, oil sump and bulk temperature, individual oil supply of the Intermediate Speed Shaft and bearings of the High Speed Shaft will be measured.

The relative deviation to the machine frame will be measured in order to investigate the influence of outer loads (loads introduced with the NTL) on the gearbox as well as to check the gearbox behavior without load. On the other hand, oil sump and bulk temperature will be measured to obtain information for the nacelle controller while the aim of measuring the individual oil supply for the ISS and the HSS bearings is obtaining a better understanding of roller bearing slipping by changing the oil volume. In **Figure 4.4** and scheme of what it has been mentioned above can be seen.

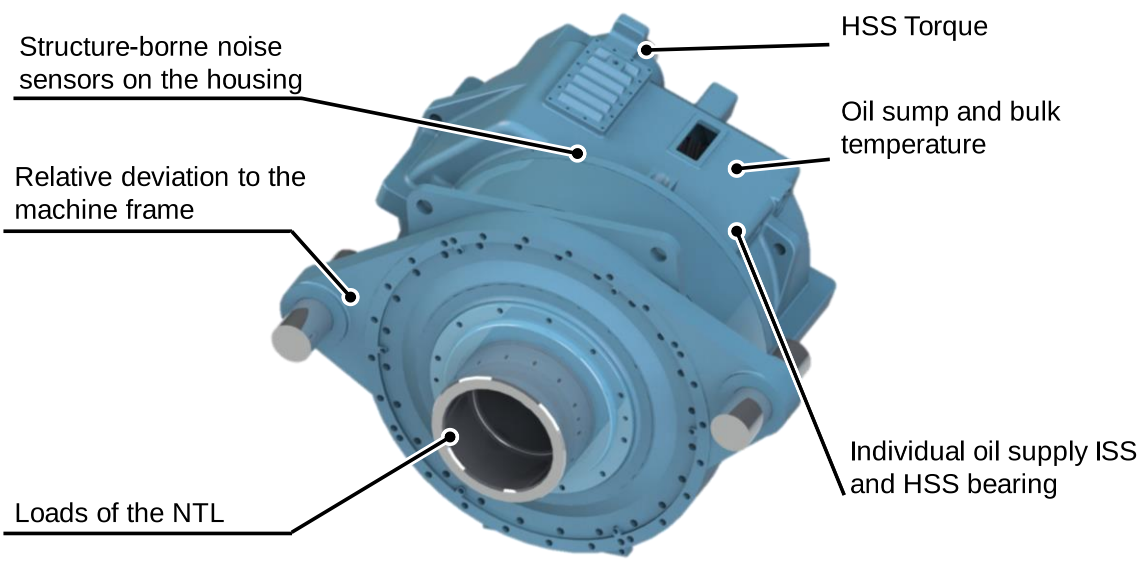


Figure 4.4: Scheme of the global measurement

4.2.2 Gear measurement

The specific purpose of gear measurement is the validation of the tooth contact analysis by evaluating the load distribution and transmission errors. Six strain gauges will be placed in tooth width in four different teeth of the ring gear while six strain gauges will be placed in tooth width in a single tooth of the sun gear in order to measure axial and radial displacement of the planet carrier relative to the housing. The aim of measuring these displacements is to analyze the load distribution caused by both inner and outer loads as well as to check the influence of rotor loads. Moreover, axial acceleration in order to analyze the influence of the slipping as well as tangential acceleration in order to check the dynamic transmission error (for high rotatory speeds) will be measured for the detection of influences of the grid. In addition, rotation angle and rotational speed will be measured too, which aim is the measurement of the transmission error (at low rotatory speeds) for the tooth contact analysis validation. Finally, contact pattern examination will be done also for the validation of the tooth contact analysis.

In **Figure 4.5** a scheme of gear measurement can be seen.

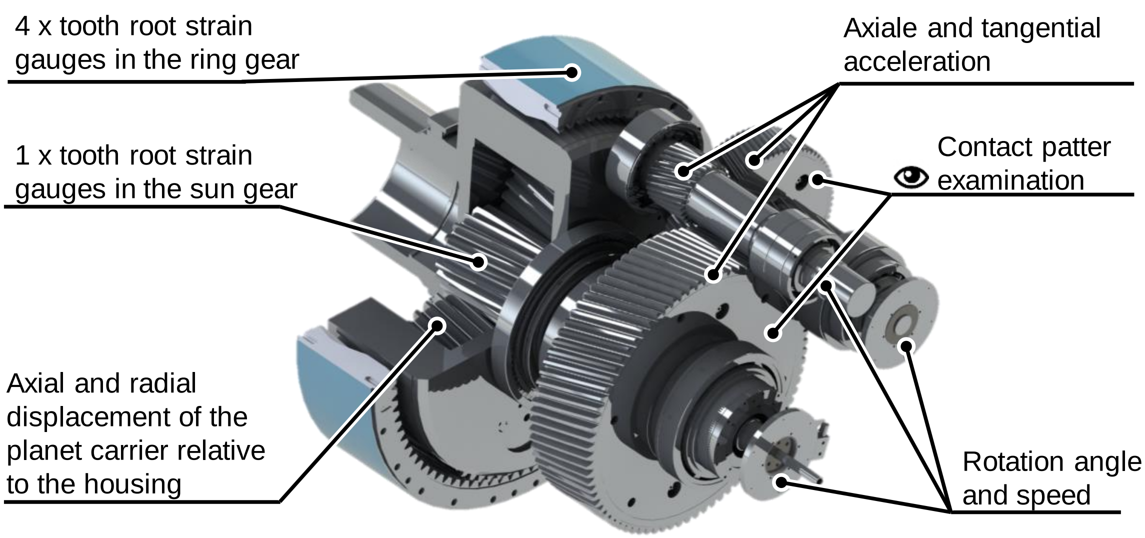


Figure 4.5: Scheme of gear measurement

4.2.3 Bearing measurement

The specific goal of bearing measurement is the validation of simulation methods for bearings, by measurement of load distribution and operational clearance. In order to measure edge contact strain gauges will be placed in the inner ring of one planet bearing. Moreover, strain gauges will be placed in the inner ring of one planet bearing to measure load distribution. During measurement, axial displacement of the ring movements, relative displacement of the planets relative to the planet carrier, slippage measurement in both HSS and ISS and temperature measurement in both inner and outer ring will be evaluated. In **Figure 4.6** and scheme of bearing measurement is shown.

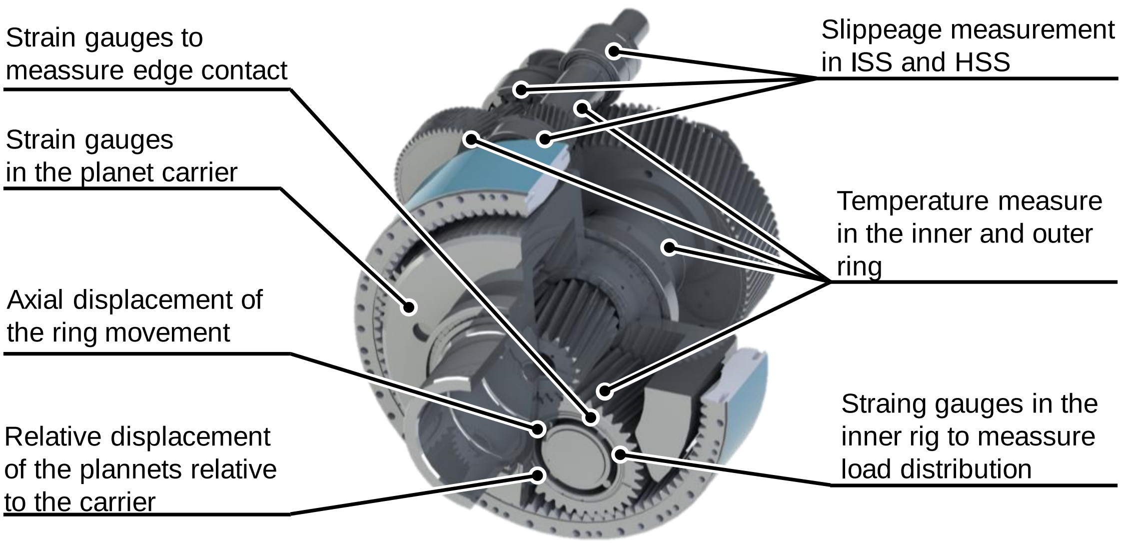


Figure 4.6: Scheme of bearing measurement

The measurement and data acquisition of the complete nacelle can be schematically seen in **Figure 4.7**, in which are shown all the parameters that will be measured during the different test cycles and how the 289 measuring points will be distributed along the different subsystems of the nacelle.

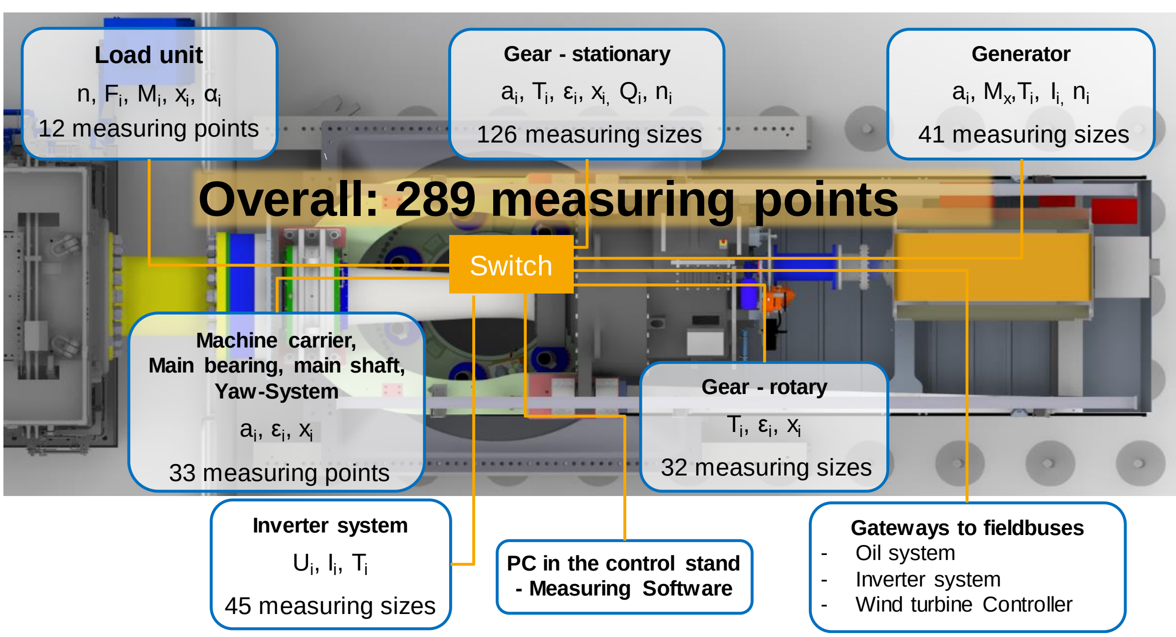


Figure 4.7: Measurement and data acquisition of the nacelle

In **Table 4.2** are shown the operational possibilities of the 4 MW test rig installed at the CWD in Aachen, Germany.

Table 4.2: Operational possibilities of the test rig

User Interface	Possible operational modes
Test rig (GE Motor, NTL Load Unit)	Speed Control: - n, My, Mz, Fx, Fy, Fz can be specified Torque Control: - Mx, My, Mz, Fx, Fy, Fz can be specified
Network	Normal operation: - “normal network” Specification of voltage and frequency: - Voltage and frequency must be specified as times series
Wind	HIL Wind: Specification of: - Average wind speed at hub height: Range: 4-22 m/s - Turbulence intensity: Range: 0.12-0.16 - Angle of incidence: Range: ± 5º
Examinee	Manual mode: manual specification o - Speed control - Torque control Real mode: HIL-mode - The wind turbine operates as in reality

The test rig can be operated under either torque control or speed control, and for both modes all loads are introduced by the NTL unit except the torque, which is introduced by the GE Motor of the test rig. The network of the test rig can be set up for a normal operation mode (as if the wind turbine is in the field producing power) or different values of voltage and frequency can be specified in order to analyze the behavior of the wind turbine under different circumstances, as the Fault Ride Through (FRT) operation mode, which will be explained in this project work.

Furthermore, the test rig can be operated in a manual mode in which the speed, leap and lamp must be stablished by the user or in a real mode with the implemented multi-physics hardware in the loop (HIL) concept, which makes possible a better understanding of the behavior of the wind turbine as if it was on the field producing power, but being tested in a workshop test rig. For this real operation mode, the requested input data is only a wind profile, defined by means of speed, turbulence intensity and direction of the flow. Wind speeds between 4 m/s (cut-in speed of the device under test) and 22 m/s (cut-out speed of the device under test) can be stablished while turbulence intensities between 0.12 and 0.16 can be specified. Moreover, the direction of the wind flow can also be set up in a range of ± 5º in order to test different wind conditions.

5 Design of measurement cycles for a 4 MW test rig

5.1 Speed run ups

This first test, is a very simple one in which the aim is to proof how is the behavior of the gearbox for different speeds of the High Speed Shaft (HSS), maintaining the torque constant. What is more, this test will be done for different torque levels, in order to prove how the gearbox behaves at each of them. On the other hand, another goal of this test is analyzing the dynamic behavior and operational vibrations of the drivetrain at each torque level varying the rotational speed. During this test network influences neither wind influences will not be taken into account and the controller of the test rig will not be necessary to carry out this test. As the acquisition system of the test rig does not have enough channels to control or measure all the needed parameters, this test must be done twice: the test will be done once with the sensor installed in the parallel stage of the gearbox and after, the test will be done with the sensors installed on the planetary stage of the gearbox. Moreover, each of those two tests will be carry out two times in order to verify the obtained results, within dynamic test conditions.

Furthermore, housing vibrations, angular acceleration, load distribution will be measured while testing the nacelle and displacement and temperature behavior will be checked out also.

As it has been said, the aim of this test is check the behavior of the gearbox at different speeds of the High Speed Shaft (HSS), maintaining the torque constant. Therefore, in order to design this test, is necessary to know the speed of the HSS, which is the speed of the gearbox at the generator side, and the nominal torque. The wind turbine manufacturer of the device under test (DUT) in which is focused this project work, the NEG Micon NM80-2750, provided the operational range of the wind turbine in terms of wind speed, the nominal operating point of it, the synchronous speed of the generator but not the speed of the HSS. So, in order to obtain the rotatory speed at the HSS some calculations are required.

Firstly, with the tip speed ratio (TSR or λ), which is ratio between the tangential speed of the tip of a blade and the actual velocity of the wind, the expression of the power can be written in the following way:

$$P_{WTG} = \frac{1}{2}\rho C_p \pi R^2 v^3 \xrightarrow{\lambda=\frac{\omega R}{v}} P_{WTG} = \frac{1}{2\lambda^3}\rho C_p \pi R^5 \omega^3 . \tag{5}$$

ρ = Air density (1.25 kg/m^3)

C_p = Power coefficient

v = Wind speed (m/s)

R = Radius of the rotor (40 m)

λ = Tip speed ratio

ω = angular speed of the rotor (rad/s)

Introducing in (5) the data of the nominal operating point of the wind turbine (P = 2750 kW, C_p = 0.112, ω = 17.5 rpm = 1.83 rad/s), is possible to obtain the value of λ, which is 3.6885. Once the value of λ is kwon, the rotor speed can be obtained:

$$\lambda = \frac{R\omega_{rotor}}{v} \rightarrow \omega_{rotor} = \frac{\lambda v}{R}. \tag{6}$$

λ = Tip speed ratio (3.6885)

R = Radius of the rotor (40 m)

ω_{rotor} = Angular speed of the rotor (rad/s)

v = Wind speed (m/s)

Introducing in (6) the cut-in speed of the turbine, which is 4 m/s, the rotor speed at which the turbine starts producing power can be obtained. The same calculation can be done for the average wind speed, which is 8.26 m/s.

$$\omega_{cut-in} = \frac{3.6885 \cdot 4}{40} = 0.36885 \ rad/s = 3.522 \ rpm. \tag{7}$$

$$\omega_{average} = \frac{3.6885 \cdot 8.26}{40} = 0.7616 \ rad/s = 7.27 \ rpm. \tag{8}$$

Finally, as the gear ratio, which value is 1:62.775, relates the speed at the Low Speed Shaft (LSS) or what is the same the rotor speed, with the speed at the High Speed Shaft (HSS); the speed at the HSS or what is the same the speed of the generator, can be obtained. Mathematically:

$$i = \frac{\omega_{rotor}}{\omega_{generator}} \rightarrow \omega_{generator} = \frac{\omega_{rotor}}{i}. \tag{9}$$

i = Gear ratio (1:62.775)

ω_{rotor} = Angular speed of the rotor

$\omega_{generator}$ = Angular speed of the generator

Introducing in (9) the obtained values of the rotor speed:

$$\omega_{gen\_cut-in} = \frac{3.522}{1:62.775} = 3.552 \cdot 62.775 = 223 \ rpm. \tag{10}$$

$$\omega_{gen\_average} = \frac{7.27}{1:62.775} = 7.27 \cdot 62.775 = 456.37 \ rpm. \tag{11}$$

In **Figure 5.1** is shown a summary with all the mentioned and calculated speeds. The real important ones in order to create the speed run-ups are the cut-in rotatory speed of the HSS and the cut-out rotatory speed of the HSS, as they represent the lower and the upper limits of the range of speeds at which the speed run-ups will be done.

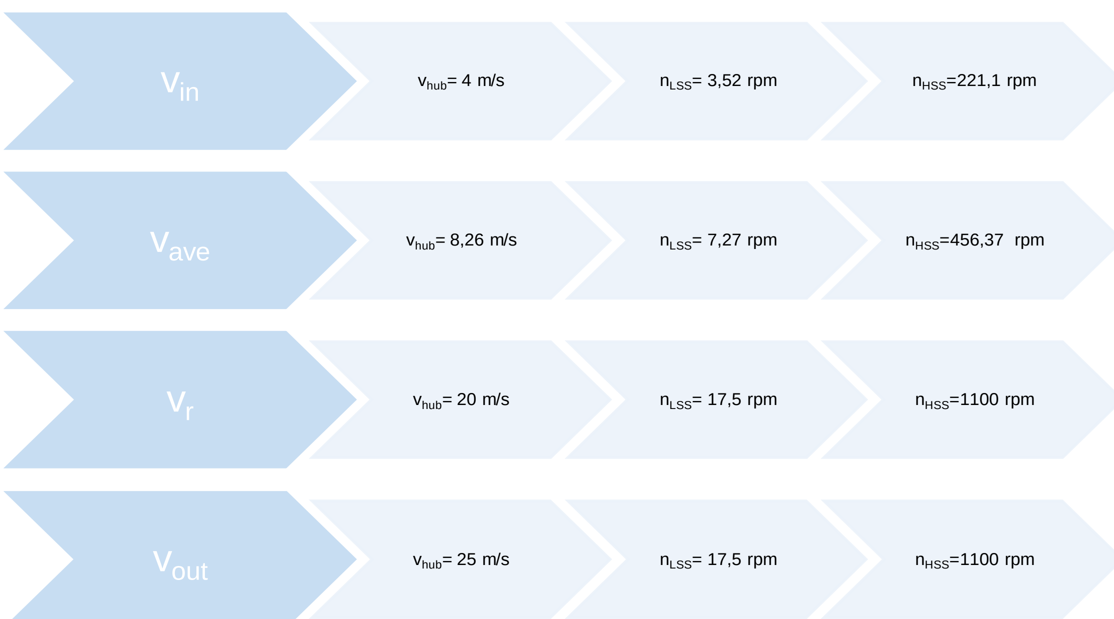


Figure 5.1: Wind speed ranges

On the scheme shown above, v_{in} means the cut-in speed, which is the speed at which the wind turbine starts up and begins producing power; v_{ave} means the average speed; v_r is referred to nominal operation conditions and v_{out} means the cut-out speed, in which the turbine stops producing power and shuts down. These speeds are represented at the power curve of the turbine that will be tested, as it is shown in **Figure 5.2**:

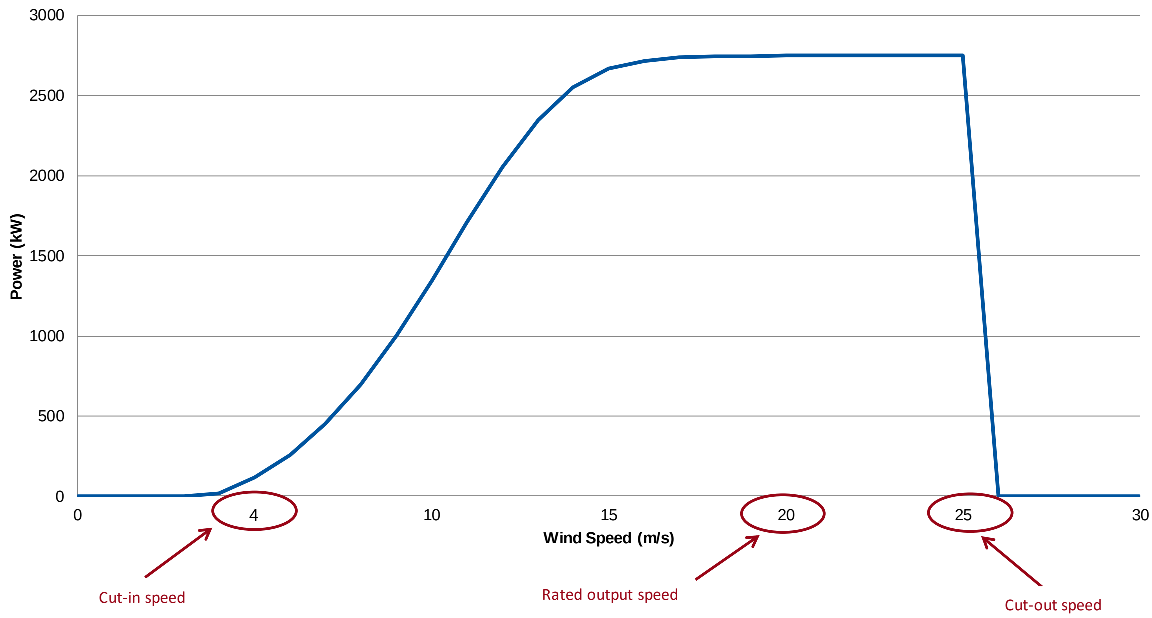


Figure 5.2: Power curve of the NEG Micon NM80

At very low wind speeds, there is insufficient torque exerted by the wind on the turbine blades to make them rotate. However, as the speed increases, the wind turbine will begin to rotate and generate electrical power. The speed at which the turbine first starts to rotate and generate power is called the cut-in speed and in this case is 5 m/s. As the wind speed rises above the cut-in speed, the level of electrical output power rises rapidly as shown. However, at 20 m/s, the power output reaches the limit that the electrical generator is capable of, which is 2750 kW. This limit to the generator output is called the rated power output and the wind speed at which it is reached is called the rated output wind speed. At higher wind speeds, the design of the turbine is arranged to limit the power to this maximum level and there is no further rise in the output power. As the speed increases above the rate output wind speed, the forces on the turbine structure continue to rise and, at some point, there is a risk of damage to the rotor. As a result, a braking system is employed to bring the rotor to a standstill. This is called the cut-out speed and in this case is 25 m/s. [WIND16]

Once all this speeds are known, the next step for the design of the speed run-ups test is determining the upper and the lower limits of the rotor speed. As the lowest limit it is chosen a value next to the cut in speed for the rotor but a bit lower, such as 200 rpm. For the upper limit it is chosen 1200 rpm, a value bit higher than the nominal value of the generator, which is 1100 rpm in order to cover a more width range of speeds.

Regarding the torque, it is known that its nominal value is 21700 Nm. For the testing, there have been set up the following torque steps (evaluated in the same period of time of 2 minutes):

- $T_{25\%}$ = 5425 Nm
- $T_{50\%}$ = 10850 Nm
- $T_{75\%}$ = 16275 Nm
- $T_{100\%}$ = 21700 Nm
- $T_{125\%}$ = 27125 Nm

The transition from one torque level to the following one will be done at the nominal speed (1100 rpm), so the power at each of these torque transitions will be:

- $P_{25\%}$ = 624,915 kW
- $P_{50\%}$ = 1249,83 kW
- $P_{75\%}$ = 1874,745 kW
- $P_{100\%}$ = 2499,66 kW
- $P_{125\%}$ = 3124,575 kW

Before doing the testing at each torque level, temperature in the bearings and in the bulk oil, which is 65 ºC, must be stable. This means that those temperatures only can suffer a deviation of ±2 K in 15 minutes. So, before the speed run-ups start at each of the torque levels a warm up of the oil of the gearbox must be done until it gets stable. Once temperature stability has been achieved, then the testing can be started. This can be better understood with the scheme shown in **Figure 5.3**.

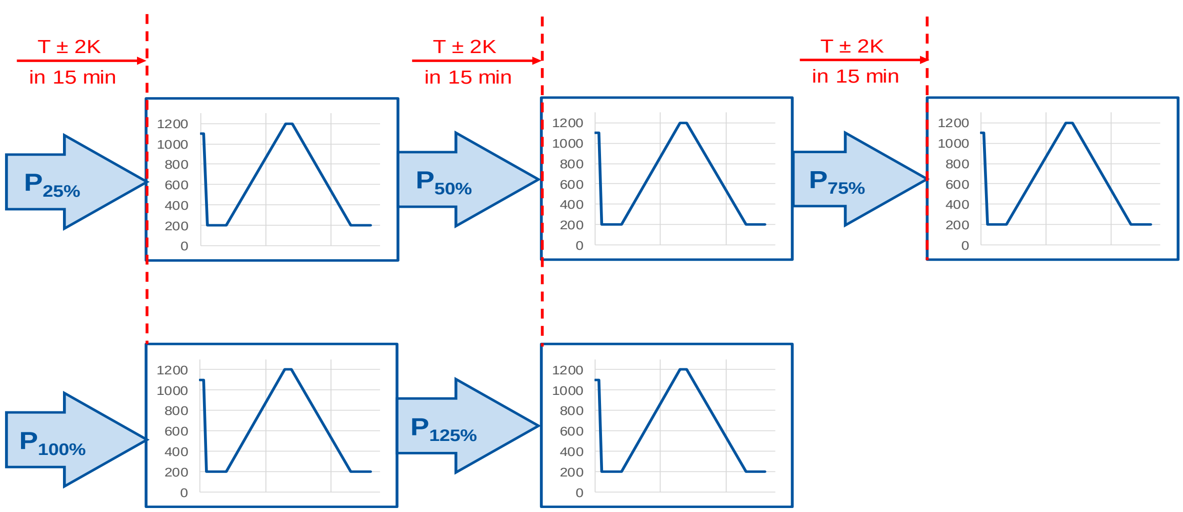


Figure 5.3: Speed run-ups test cycle

The first warm up usually takes longer than the following ones and takes even longer when the torque of the first warm up is low. Because of previous experiences it is believed that the first warm up usually takes around two hours until the temperature of the oil is stable. However, the rest of the warm ups that must be done before the speed run-ups at the other torque levels only take normally around half an hour.

For the testing, the following durations at each of the speeds have been set up:

- Start-up time: 10 minutes, up to the nominal speed (1100 rom) at $T_{25\%}$
- 1 minute to get from 1100 rpm down to 200 rpm
- At 200 rpm the shaft will be rotating for 1 minute
- 3 minutes to get from 200 rpm to 1200 rpm
- At 1200 rpm the shaft will be rotating for 20 seconds
- 20 seconds to go from 1200 rpm down to 1100 rpm

The test at each torque level should last until oil meets the cleanliness limits stated at the ISO 4406/15/12. At our test rig, this would not be possible because the lubrication system will not be equipped with recirculation. So, the oil would be at each tests the same and it will not be cleaned after testing at each torque level.

For the construction of the speed run ups it has been used the data provided at **Table VI.1** in the Annex of this document, in which it has been calculated the power at each torque step for each of the speeds use in the test, in order to verify that the power limit of the generator will not be overpassed at any moment.

In Table VI.1, it can be observed in yellow the points of the test at which the power will be close to the generator's power and in red color are shown the points at which the power of the generator will be overpassed. However, the generator can support this overload, so this two points of the testing will not be a thread for the generator.

To sum up, the result for the Speed ramps test can be seen in **Figure 5.4**, being illustrated in blue color the rotatory speed of the HSS, in yellow color the torque and in red color the warm ups.

So, duration of the speed run ups is of 16920 seconds, or what is the same, 4.7 hours. As it has been mentioned before, the acquisition system has not enough channels, so the test has to be done twice, doing a duration 9.4 hours. Finally, in order to verify the repeatability of the results, the test will be carried out two times, so the total duration of the Speed run-ups test will be 18.8 hours.

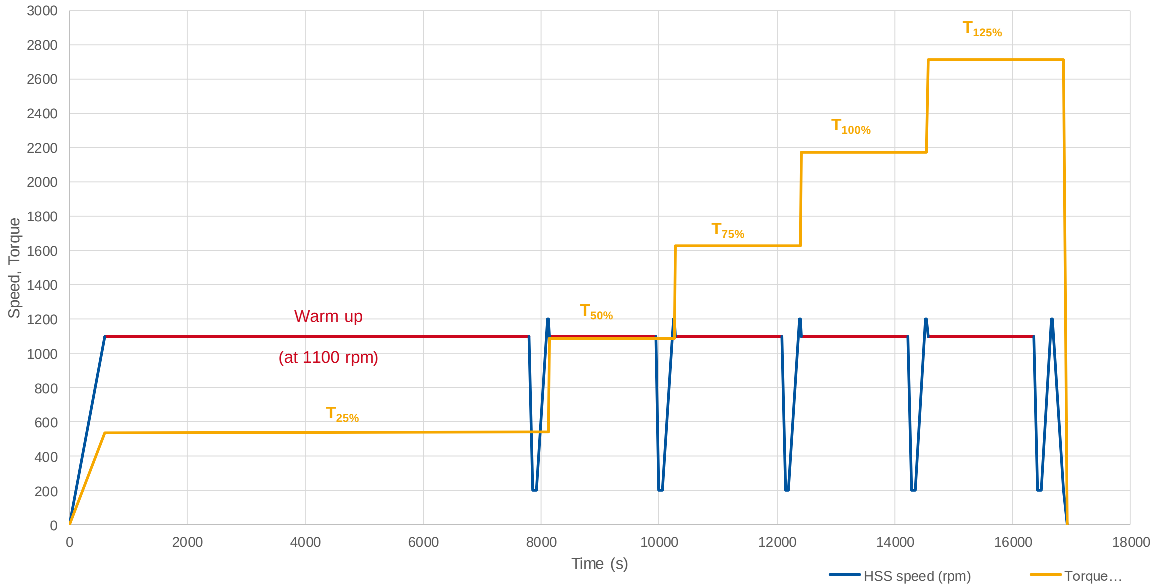


Figure 5.4: Plot of the speed run ups test

5.2 HIL operation

The goal of the HIL operation test is to determine the behavior of the wind turbine under real operating conditions. As nowadays the overall system behavior and the complex interaction between the different subcomponents of the drivetrain are poorly investigated, particular realistic workshop tests of complete nacelles are needed in order to clarify reliability details. The 4 MW nacelle test rig installed on the Center for Wind Power Drives (CWD) of Aachen includes a multi-physics Hardware in the Loop (HIL) concept, which reproduces realistic stresses on the device under test (DUT) as it would experience in the field. The objectives of the HIL operation mode are [FRAN14]:

- Providing reproducible loads and conditions
- Good observability under real conditions
- Focus on comprehensive functionality
- Analysis of several damage mechanisms with high accuracy
- Improvement of existing and future designs

Moreover, the control system of the HIL feature emulates a real six degree of freedom environment including all mechanical stress and so electrical interactions with the grid that the nacelle would experience in the field. So, during the testing of nacelles in test rig with implemented multi physics hardware in the loop (HIL) concept, all subsystems of the nacelle, including the controller, behave as if they were on top of the tower. In addition, critical test scenarios such as fault ride-through (FRT) tests, which will be explained in the next chapter, can also be performed. [FRAN14]

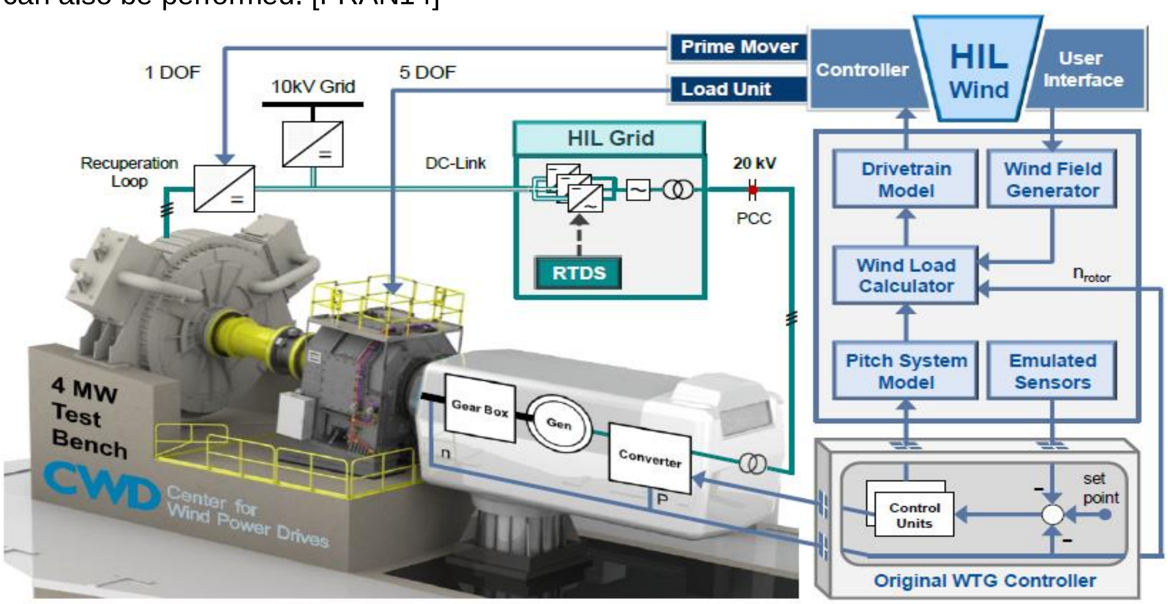


Figure 5.5: Scheme of the HIL operation system

In **Figure 5.5** is illustrated the scheme of the HIL system with all its components. The HIL system can be divided into two different subsystems: HIL wind emulator and HIL grid emulator.

5.2.1 HIL Wind Emulator

The goal of this unit, is to emulate the missing rotor and wind field of the test rig. It includes a real-time simulation of the aerodynamic and mechanical characteristics of the rotor as well as a load application system to apply forces and moments to the device under test. The main module is the wind load calculator, which consists of five parts. The input parameters of the wind field, which are wind speed and turbulence class, are interpreted in the wind field generator which will create afterwards a 3D wind field. Simultaneously, the pitch system model converts and interprets the incoming signals, such as pitch angle and yaw, from the controller. Once all the input data of the wind field has been interpreted, the wind load calculator determines the resulting wind loads in all six degree of freedom, which will be forwarded as reference values to the load application system. However, these loads have to be interpreted by the drivetrain model in order to generate control signals for prime mover and load unit. The prime mover converter receives the torque/speed signal for the rotational movement (one degree of freedom) meanwhile the load unit controller receives actuator signals for radial forces, thrust, yaw and tilt bending moments (five degrees of freedom). The nacelle with its real sensors and with the emulated sensors of the wind load calculator detects all mechanical input value in real time and closes the control loop inside the WTG controller. Apart from the real measured speed and power of the operating nacelle the pitch angle, yaw and anemometer signals must be provided by the HIL Wind system. [FRAN14], [AVER15]

5.2.2 HIL Grid Emulator

Independently from the HIL Wind system, the HIL Grid system is able to emulate all kinds of electrical grid states and grid loads on the grid side of the nacelle. This system is integrated into the electrical recuperation loop of the test bench, which is marked in green color in Figure 5.5. At the point of common coupling (PCC) the nacelle power output is connected to the power electronics of the HIL Grid system at 20 kV AC level. The power electronics consist of several converter systems as well as a power transformer and a filter. Furthermore, a real time digital simulator (RTDS) is part of the HIL Grid system, which function is controlling the emulation of the grid. The DC Link is fed by the power output of the nacelle and supplies the prime mover. In order to avoid undesirable feedback from the grid emulation, the power supply for the test rig is completely decoupled from the HIL Grid.

In addition, all electronical responses of the grid, specially grid faults, are detected by the WTG controller at the PCC. [FRAN14], [AVER15]

In this chapter a test procedure will be developed for the HIL Wind operation system while the HIL Grid one will be developed in the following chapter. As the life of a wind turbine can be represented by a set of design situations covering the most significant conditions that the wind turbine may experience, the HIL Wind test will be designed with a wind profile based on the different IEC 61400-1 Design Load Cases (DLC) for the different working modes of a wind turbine, in which different wind speeds, directions, turbulences and gusts conditions will be considered. So, before the test procedure is described it is necessary to have an overview of the external conditions. Wind turbines are subjected to environmental and electrical conditions that may affect their loading, durability and operation. The environmental conditions are divided into wind conditions, that affect structural integrity, and other conditions while the electrical conditions refer to the electrical power network conditions. Moreover, the external conditions are subdivided into normal and extreme categories. The normal external conditions concern recurrent structural loading conditions while the extreme external conditions refer to rare external conditions. The design load cases shall consist of potentially critical combinations of these external conditions with wind turbine operational modes. Furthermore, the external conditions to be considered are dependent on the intended site or site type for a wind turbine installation. Wind turbine classes are defined in terms of wind speed and turbulence. The wind turbine in which is based this project work is a class III wind turbine with a B turbulence category, which means that a 0.14 turbulence intensity is expected for 15 m/s, as it can be checked on the IEC 61400-1. In this chapter will only be taken into account wind conditions for both normal and extreme situations. A wind turbine must be designed to withstand safely the wind conditions defined by the selected wind turbine class. Below are mentioned the different normal and extreme wind conditions that can occur in a wind field [IEC05]:

1. **Normal wind conditions**
   1.1. Normal wind profile model (NWP)
   1.2. Normal turbulence model (NTM)
2. **Extreme wind conditions**
   2.1. Extreme wind speed model (EWM)
   2.2. Extreme operating gust (EOG)
   2.3. Extreme turbulence model (ETM)
   2.4. Extreme direction change (EDC)
   2.5. Extreme coherent gust with direction change (ECD)
   2.6. Extreme wind shear (EWS)

Based on the different wind situations mentioned above, different design load cases can be established, as it is explained in the IEC 61400-1. The load cases have to be determined from the combination of operational modes or other design situations. All relevant load cases with a reasonable probability of occurrence have to be considered. In **Table 5.1** are represented the different design load cases (DLC) for different design situations that the IEC 61400-1 standard describes and in which will be based the wind profile that will be afterwards designed for the HIL Wind test. [IEC05]

Table 5.1: IEC 61400-1 Design Load Cases

Design Situation / Working mode	DLC	Wind condition	
1. Power production	1.1	NTM	Vin < Vhub < Vout
	1.2	NTM	Vin < Vhub < Vout
	1.3	ETM	Vin < Vhub < Vout
	1.4	ECD	Vhub = Vr – 2 m/s, Vr, Vr + 2 m/s
	1.5	EWS	Vin < Vhub < Vout
2. Power production plus occurrence fault	2.1	NTM	Vin < Vhub < Vout
	2.2	NTM	Vin < Vhub < Vout
	2.3	EOG	Vhub = Vr ± 2 m/s and Vout
	2.4	NTM	Vin < Vhub < Vout
3. Start up	3.1	NWP	Vin < Vhub < Vout
	3.2	EOG	Vhub = Vin, Vr ± 2 m/s and Vout
	3.3	EDC	Vhub = Vin, Vr ± 2 m/s and Vout
4. Normal shut down	4.1	NWP	Vin < Vhub < Vout
	4.2	EOG	Vhub = Vin, Vr ± 2 m/s and Vout
5. Emergency shut down	5.1	NTM	Vhub = Vin, Vr ± 2 m/s and Vout
6. Parked (standing still or idling)	6.1	EWM	50 year recurrence period
	6.2	EWM	50 year recurrence period
	6.3	EWM	1 year recurrence period
	6.4	NTM	Vhub < 0.7 Vref
7. Parked and fault conditions	7.1	EWM	1 year recurrence period

Once the external conditions and the design load cases have been mentioned, the wind profile can be created.

As the acquisition system of the test rig does not have enough channels to control or measure all the needed parameters, this test must be done twice: the test will be done once with the sensor installed in the parallel stage of the gearbox and after, the test will be done with the sensors installed on the planetary stage of the gearbox. Moreover, each of those two tests will be carry out two times in order to verify the obtained results, within dynamic test conditions.

The first approach of the wind profile can be seen in **Figure 5.6** and **Figure 5.7**. In this approach 5% turbulence bins have been set up, from 0% up to 25% and regarding the flow direction, angles from -5º up to 5º have been considered, in steps of 2.5º. It has been selected that range as the HIL system is stable between those limits (-5º-5º). Moreover, 10 minutes´ time series have been stablished as the IEC 61400-13 standard orders and afterwards the same test cycle has been prepared but for 2 minutes´ time series for each event in order to analyze the behavior of the turbine for short wind gusts. To create the wind profile, all the possible design situations mentioned in **Table 5.2** have been considered.

For the first approach of the HIL Wind test, two different tests have been designed. In the first test the aim is to vary the turbulence for each of the wind speeds in order to do a turbulence analysis, this is, the aim is to check how the turbulence affects to the wind turbine and more specifically to the gearbox. On the other hand, for the second test the aim is to vary the flow direction for each of the speeds in order to check how the flow direction affects to the wind turbine and thus, to the gearbox.

In the Annex of this document, In **Table VI.2** can be seen the data used to create the test one (turbulence analysis) while in **Table VI.3** can be seen the data used to build up the second test for the flow direction analysis.

In both **Figure 5.6** and **Figure 5.7** is illustrated the same wind profile, but in Figure 5.6 is shown the wind profile for the first test, this is, for the turbulence analysis test, whereas in Figure 5.7 is shown the wind profile for the flow direction analysis. These two plots are illustrated in a bigger size in the Annex of this document.

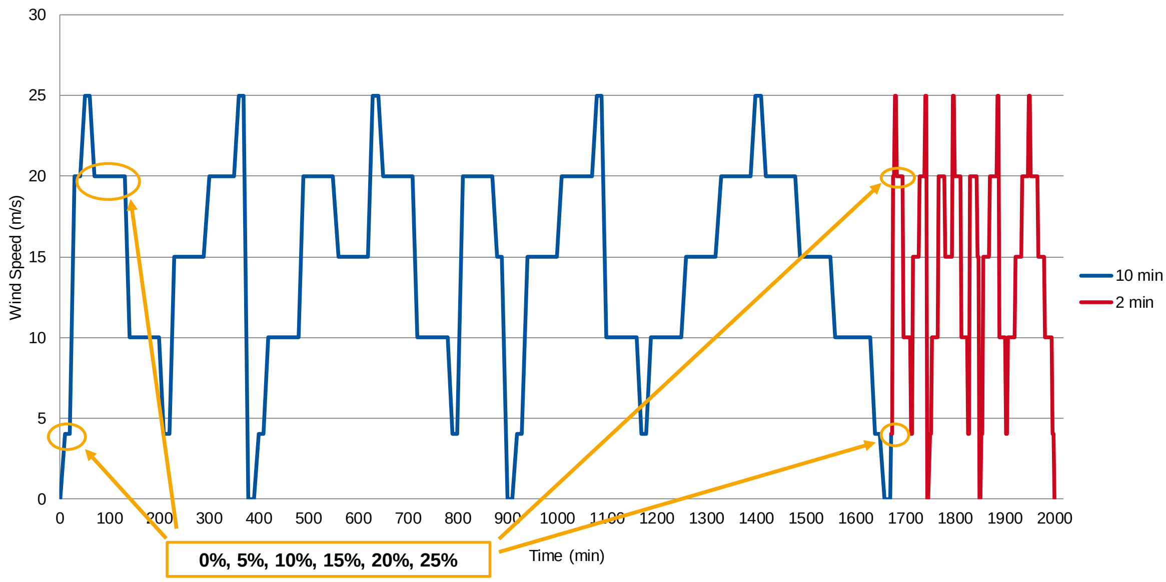


Figure 5.6: TEST 1. Wind profile for a turbulence analysis

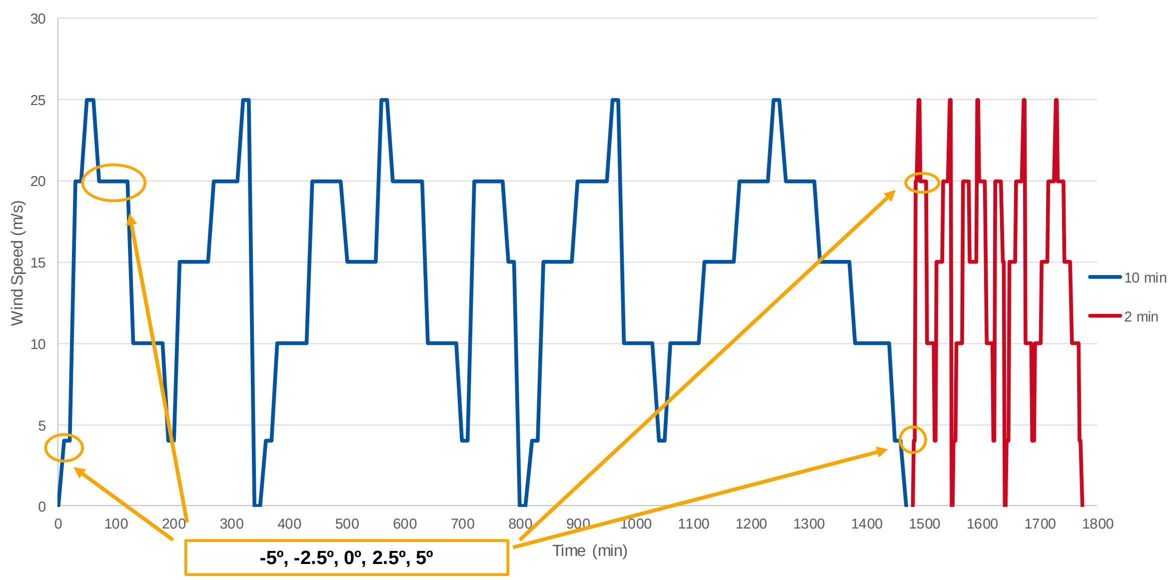


Figure 5.7: TEST 2. Wind profile for a flow direction analysis

The duration of the test within this first approach would be:

- TEST 1. Turbulence analysis: total duration: 2002 minutes.
- TEST 2. Flow direction analysis: total duration: 1774 minutes

So, the total duration of the test would be: 2002 + 1774 = 3776 minutes, which are 62.93 hours. As it is obvious, it has been considered too much time for a unique test. Therefore, it has been concluded that this approach is not affordable to perform it in the test rig. So, this approach has been declined.

As the first approach for the HIL Wind test has been declines, a second one has been designed, a second approach has been designed. For it, time series of 10 minutes have been stablished for each design situation (power production, start-up…). The wind speeds that have been considered are all between the cut-in speed, which is 4 m/s, and the cut out speed, which is 25 m/s. Regarding the turbulence, values between 5%-10% have been considered as normal turbulence while 25% has been considered as extreme turbulence model. Two different situations have been considered regarding the direction of the flow, which are direct flow (0º) or inclined flow (between -5º and 5º). Values higher than 5º or less than -5º have not been taken into account as the HIL system's stability is between these two values.

In the second approach three different tests have been designed, using in all of the same wind profile. As it can be seen in **Table VI.4, Table VI.5** and **Table VI.6** in the Annex of the document, all the design load cases mentioned in Table 5.2 have been considered to create the wind profile and each wind condition is linked in those tables to the corresponding design load case. So, with this approach, the wind turbine will be tested for all the different scenarios that the IEC 61400-1 contemplates.

In **Figure 5.8** is shown the designed test cycle for the second approach. As it can be observed in the plot illustrated in Figure 5.8, the three tests seem to be equal but the difference between them are the values of the turbulence and the flow direction, as it can be noticed in the Tables of the Annex. This is, the goal of this second approach has been combining both turbulence and direction flow analysis mentioned in the first approach of the HIL Wind test.

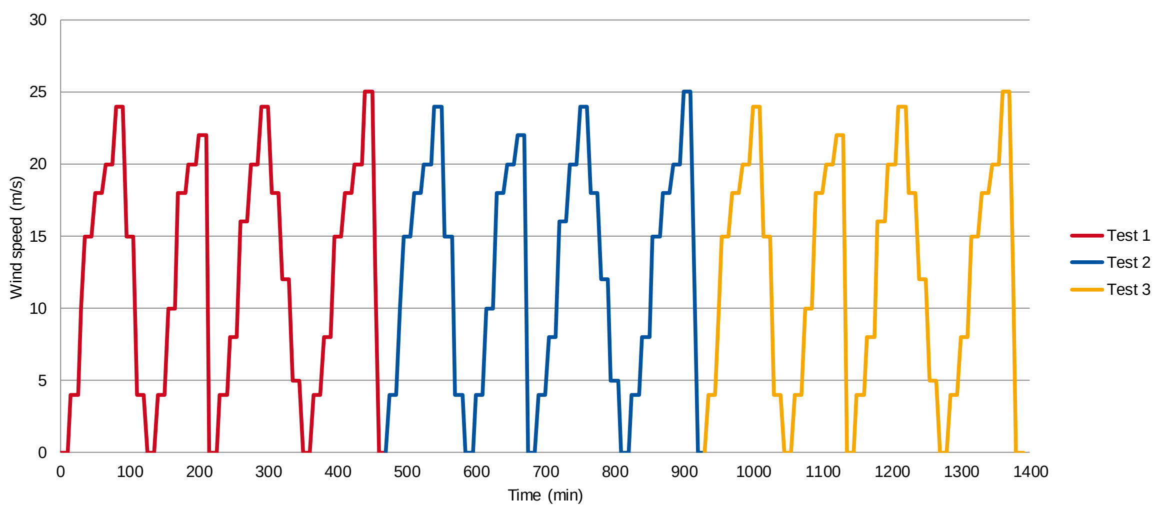


Figure 5.8: Plot of the test cycle of the second approach of the HIL Wind test.

To get a better understanding of this approach, an explanation about the data used for the design of the tests and about the differences between the three tests will be done in the following lines. Between the 10 and 140 minutes of the first test, the wind profile represents a normal power production, DLC 1.1, for different wind speeds and different turbulences. On the first test, the wind has a direct flow (0º) while on the second test (between the 470 and 600 minutes) the wind has a deviation of 5º and of -5º on the third test (between the 930 and 1060 minutes). With this, in this time period an analysis of the direction of the flow can be done, being possible an interpolation of the results for angles between -5º and 5º.

Moving forward in time, between the 140 and 230 minutes of the first test, the wind profile suffers an extreme change of direction of the flow, being representative of the DLC 1.4. On the first test this scenario has been designed for a 5% of turbulence, meanwhile on the second test (between the 600 and 690 minutes) and the third test (between the 1060 and 1150 minutes), this scenario has been designed for a 10% and 15% of turbulence consecutively. So, with this three different situations of turbulence maintaining the same direction of the flow and wind speed, a turbulence analysis can be done for this scenario.

Another wind condition has been created between the 230 and 360 minutes of the first test and accordingly between the 690 and 820 minutes on the second test and between the 1150 and 1280 minutes on the third test. In these periods of time, the created wind profile represents an extreme turbulence model but in the first test a direct flow has been set up (0º) while on the second and third tests deviations of 5º and -5º consecutively have been set up. Therefore, with these three different scenarios an analysis of the influence of the direction of the flow can be done for the extreme turbulence model.

To finish with the explanation of the second approach, between the 360 and 470 minutes of the first test and accordingly between the 820 and 930 minutes on the second test and between the 1280 and 1390 minutes on the third test, the wind profile represents a normal turbulence model for different directions of the flow but for different turbulences in each of the tests. So, once again an analysis of the influence of the turbulence for different wind speeds and angles can be done with these tests.

The duration of the HIL Wind test within this approach would be (as it can be checked in Tables VI.4, Table VI.5 and Table VI.6) 1390 minutes, or what is the same 23.16 hours. This is a more realistic duration for the test so it can be concluded that this second approach will be the selected one. As it has been said, the test must be carried out two times because the acquisition system has not enough channels to measure all the parameters in one unique test and those two tests will be done two times in order to verify the repeatability of the obtained results. All this makes a total duration of the HIL operation test of 23.16 x 2 x 2 = 92.64 hours.

Once the wind profile is created, the next step is to create a capture matrix as the IEC 61400-13 says in order to collect all the data of the test. In a capture matrix, time series of 10 minutes are recorded up to the rated wind speed (20 m/s in this case) and time series of 2 minutes are recorded from the rated wind speed until the cut-out speed, for the different wind speed and turbulences. The complete capture matrix can be shown in **Table VI.7** in the Annex for normal power production. Furthermore, in the Annex of this document can also be found the capture matrixes for the other operating conditions of the wind turbine like power production with a fault, standing-still and idling, start up and others. With the filled in capture matrix for normal power production, a tridimensional plot can be done in order to get a better understanding of it, as it is illustrated in **Figure 5.9**.

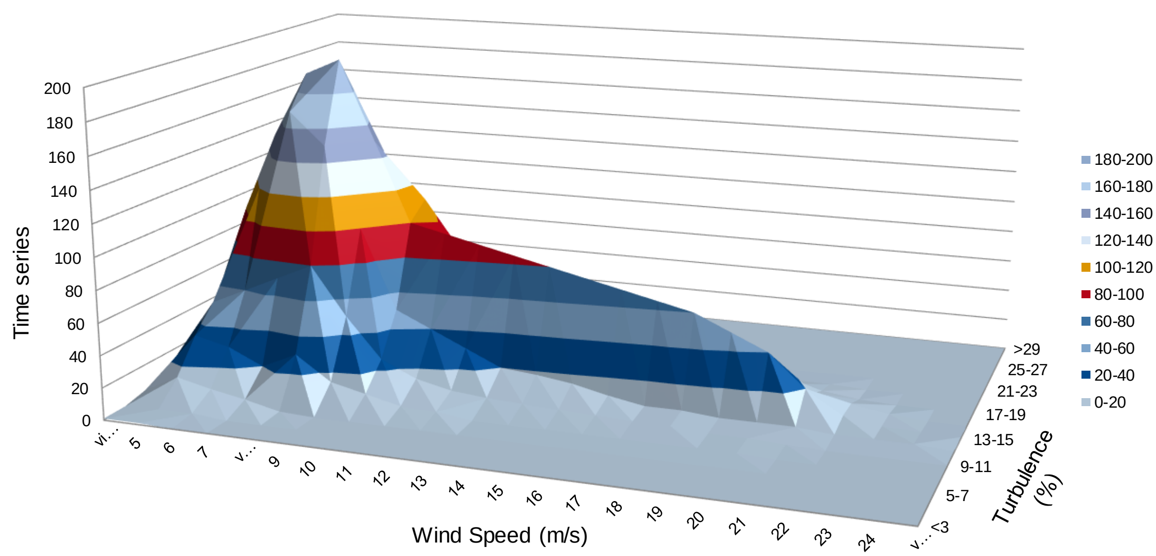


Figure 5.9: Tridimensional plot of the capture matrix for normal power production

In this plot it can be observed that the main measurements should be done around the average speed, which is 8 m/s and for turbulences between 13% and 15%. For the highest wind speeds so much measurements are not required as those speeds only appear occasionally.

5.3 FRT test

Decades ago, the low penetration of wind energy into the electric grids, allowed the disconnection of wind turbines in case of unusual voltage or frequency of the grid. However, with the increasing share of electricity produced by wind power in the last years, the mentioned behavior is not acceptable. If too many wind farms would have to be disconnected because of a grid failure, the complete network could break down, a scenario which is also called a “blackout”. So, today the behavior of wind turbines during grid faults has become of great importance and more and more often grid code specifications require that wind turbines must be able to ride through all kinds of grid faults, including faults with very low remaining voltage levels and unsymmetrical (1 phase and 2 phase) faults. [NEUM12]

The grid codes determine the fault boundaries among the ones a grid connected wind turbine shall remain connected to the network, giving rise to specific voltage profiles that specify the depth and clearance time of the voltage drops that wind turbines must withstand without tripping. Those requirements are known as Fault Ride Through (FRT) or Low Voltage Ride Through (LVRT) and they are described by a voltage versus time characteristic, denoting the minimum required immunity of the wind farm. [LUNA12]

The certification procedures of wind turbines include mechanical, structural and electrical requirements in case of normal and faulty operation. This chapter will focus on the electrical certification procedures, which state that wind turbine operation conforms to the grid codes of network operators. Therefore, a standardized Fault Ride Trough (FRT) test procedure will be developed in the following lines, attending to the IEC 61400-21. The aim of this test is to analyze the gearbox behavior while a grid fault happens, this is, with the FRT test it will be checked out how a grid failure can affect to the gearbox. The FRT operating condition constitutes the main electrical challenge for a wind turbine and the FRT tests consist on checking if the wind turbine fulfills the specified behavior requirements of the grid codes in presence of none nominal conditions or faults at the time of coupling with the power system. Moreover, FRT are usually performed in the field, but it is time consuming and costly due to the dependence of natural factors (occurrence of prescribed wind and resulting load conditions) and location (top of the tower, limited accessibility and sensorization) [RWTH16]. On the other hand, workshop test rigs for FRT testing can address all these limitations. As workshop testing of FRT is not clearly developed in the IEC 61400-21 standard, this chapter pretends to be a guideline for FRT testing in test rigs. For the FRT testing the HIL Grid emulator will be used in order to emulate the sensors that cannot be installed as well as to emulate the grid conditions.

Generators of wind turbines are equipped with power electronics to enhance the static as well as the dynamic behavior of wind turbines. Nowadays, power electronics is increasingly becoming an important part of the electrical grids. Different generator concepts for wind turbines are used these days. On the one hand there is the full converter concept in which the generator is connected to the grid through a full-scale converter system. The full-scale concept can be used with asynchronous as well as synchronous generators. On the other hand, most of the currently installed wind turbines, and so the wind turbine in which is based this project work, are equipped with the so called doubly-fed induction generator (DFIG), which is an asynchronous generator. [NEUM12]

In **Figure 5.10** is shown the scheme of a wind turbine with a DFIG. While the stator of the generator is directly connected to the grid, the rotor windings are linked to the grid through voltage source converters.

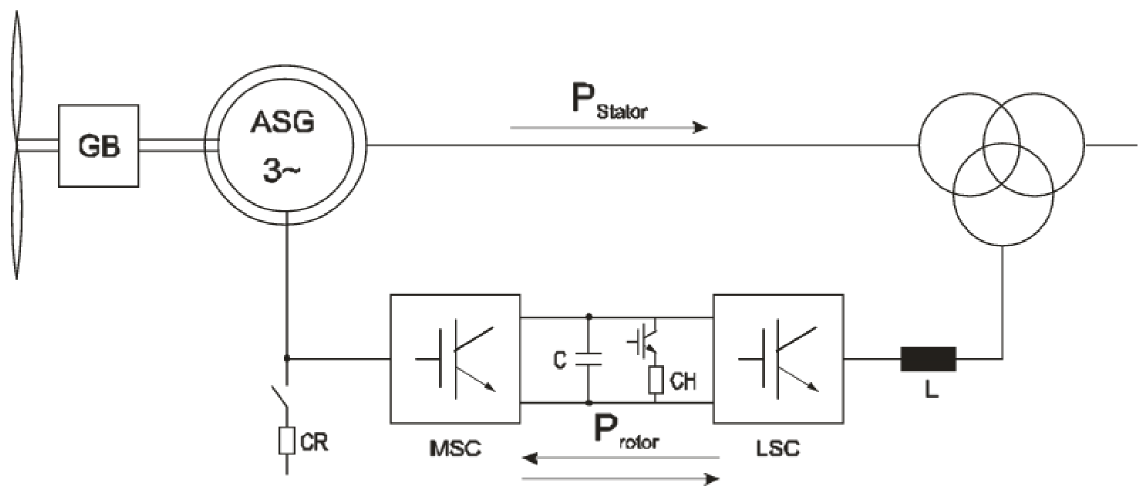


Figure 5.10: Basic concept of DFIG-based wind turbine

The control of the DFIG can be separated into the line side converter (LSC) and the machine side converter (MSC) controls. The main function of the LSC is to maintain the direct current (DC) voltage and provide reactive current support in order to optimize of the reactive power sharing between MSC and LSC in steady state. During grid faults additional short-time reactive power has to be supplied in order to support the grid voltage. On the other hand, the MSC controls active and reactive power of the generator independently from one another and follows a tracking characteristic to adjust the generator speed for optimal power generation depending on the wind speed. By controlling the generator by means of frequency converter, electrical flicker is reduced. [NEUM12]

The converter allows controlling the amplitude, frequency and phase angle of the rotor voltage. This is what enabled the variable speed operation of the generator, which allows to adapt the generator speed according to the wind speed, in order to increase the wind power. The speed range of generator is about ±30% of the synchronous speed, which is 1100 rpm, and thus, the speed of the generator is decoupled from grid frequency. Therefore, the losses in the power electronic converter can be reduced, compared to a system where the converter has to handle the entire power, and the system costs lower due to the partially-rated power electronics. The DFIG operates in both sub-synchronous and super-synchronous modes with a rotor speed range around the synchronous speed. So, with a DFIG the active and reactive power can be controlled independently one from each other, which leads the possibility of choosing the desired power factor. [NEUM12]

Nevertheless, generators with direct grid connection of one generator winding are commonly sensitive to sudden grid voltage variations. This fact proves to be crucial for induction generators due to their direct coupling between excitation and torque production inside the same winding arrangement. As the main flux linkage cannot follow sudden voltage variations, these variations will cause considerable and slowly declining current transients with maximum amplitudes of possibly more than six times the rated current. All this can lead to trips of some protection units of the rotor side frequency converter (MSC) and front-end power switch. Furthermore, when a voltage drop happens in a wind turbine, a serious problem is the torque stress that gears and main shaft suffer during such transients. Besides all this, grid operators demand continuous operation of the generator system during grid disruptions and short circuits with feeding off a defined short circuit current into the fault location in order to enable power system protection devices to trip. [PÖLL14]

All in all, the control strategy of the wind turbine has to enable the following to the wind turbine while a grid fault happens [DITT05]:

1. Maintain the wind turbine connected to the grid while the fault happens and do not consume active power from the grid.
2. Supply reactive power during the fault in order to recover the voltage.
3. Operate under normal conditions once the fault has disappeared.

In addition, the design of a generator system which offers FRT capability has to fulfill the following [DITT05]:

1. Protection of the converter components against overcurrents and overvoltages.
2. Fulfillment of the applicable grid codes, which depend on the country.
3. Applying of minimum possible mechanical stress to gear and main shaft.
4. Reduction of transient short circuit current to enable operation at grids with low short circuit power.

5.3.1 Grid code requirements

Grid code requirements vary according to the locally applicable code and regarding the generator system design those requirements imply [DITT05]:

- Limitation of the transient short circuit current below the natural maximum transient short circuit current i_k'' of the generator.
- Full controllability of the short circuit current during the voltage dip.

Regarding FRT, grid codes define two main issues:

- The properties of the voltage dip during which the generating system has to maintain operability
- Limits or requirements for the short circuit current during the grid fault.

Requirements can be summarized like shown in **Figure 5.11**:

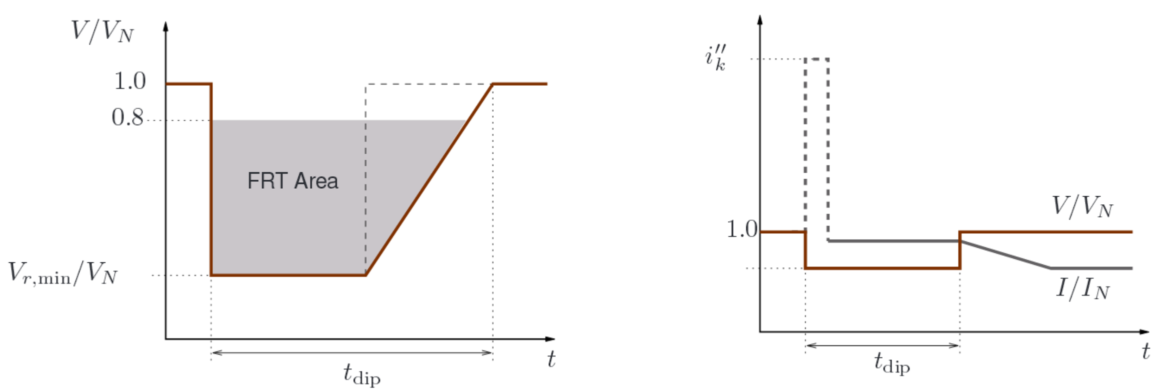


Figure 5.11: Voltage and current definitions of grid codes.

a) Voltage dip duration and retaining voltage

b) Short circuit current during voltage dip

The minimum retaining voltage $V_{r,min}$ defines the minimum voltage and the dip duration, t_{dip}, defines the maximum time at which the turbine has to maintain connected to the grid. Nowadays, grid codes request minimum retaining voltages between 15% and 25% and maximum voltage dip durations between 500 ms and 3000 ms, depending on the retaining voltage. Moreover, short circuit requirements address the maximum transient short circuit current i_k'' and the current to be fed to the grid during the voltage dip. In fact, suppliers which are able to offer systems with low i_k'' (e.g.<2.5 i_N) will gain a clear market advantage. [DITT05]

In **Figure 5.12** can be seen a typical DFIG with all the essential components for the FRT operation, which its basic components are the rotor side crow-bar and the electronic switch on the stator side. The crow-bar circuit on the rotor side protects the frequency converter from overvoltages originating in the rotor windings during grid faults, which potentially could destroy the converter unit.

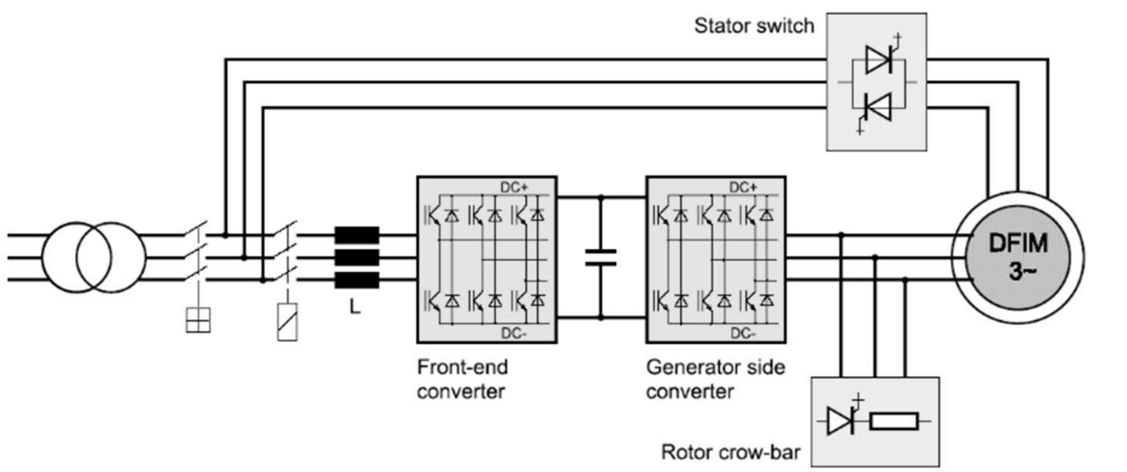
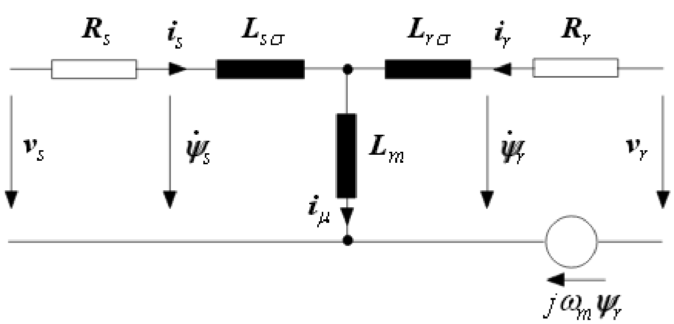


Figure 5.12: DFIM Generator with all the components for FRT operation

5.3.2 System reaction to grid faults

As in a DFIG the stator winding is directly connected to the grid, the system reaction to grid faults is determined by the short circuit behavior of the generator. In **Figure 5.13** is illustrated the equivalent diagram [DITT05].



L_m = Mutual (main) inductance

R_s = Stator resistance

R_r = Roto resistance

$L_r = L_m + L_{r\sigma}$ = Rotor inductance

$L_s = L_m + L_{s\sigma}$ = Stator inductance

Figure 5.13: Equivalent circuit of DFIM

This circuit can be derived from the basic voltage/flux equation system of an induction machine.

$$v_s = R_s i_s + \frac{d\psi_s}{dt} \tag{12}$$

$$v_r = R_r i_r + \frac{d\psi_r}{dt} - j\omega_m \psi \quad . \tag{13}$$

$$\psi_s = L_s i_s + L_m i_r \tag{14}$$

$$\psi_r = L_r i_r + L_m i_s \tag{15}$$

v_s = stator voltage

v_r = rotor voltage

i_s = stator current

i_r = rotor current

ψ_s = stator flux linkage

ψ_r = rotor flux linkage

(all parameters are referred to the stator side)

The outer voltage loop of this system consists of three voltage sources whose balance defines the currents flowing through stator and rotor

$$v_s - v_r - j\omega_m \psi_r = R_s i_s + L_{s\sigma} \frac{di_s}{dt} - R_r i_r - L_{r\sigma} \frac{di_r}{dt} \, . \tag{16}$$

If one of those voltages changes suddenly (like the rotor voltage after a grid fault) and this change is not immediately compensated, a current is driven through both stator and rotor windings, which is only limited by the total leakage inductance of the machine in the first place. Transients of torque, active power and reactive power follow accordingly. [DITT05]

Moreover, assuming complete rotor and stator short circuit (v_s = v_r = 0) and constant speed and stator frequency during the whole process, the current is determined by the generator speed and the initial values of stator and rotor currents. [DITT05]

With the time constants:

$$T_s = \frac{L_s}{R_s} \tag{17}$$

$$T_r = \frac{L_r}{R_r} \tag{18}$$

$$\frac{1}{T_\sigma} = \frac{1}{\sigma T_s} + \frac{1}{\sigma T_r} \tag{19}$$

and the leakage factor:

$$\sigma = 1 - \frac{{L_m}^2}{L_r L_s}. \tag{20}$$

The following expression can be obtained:

$$i_s(t) = i_s(0) e^{\frac{t}{T_\sigma}} e^{j\omega_m t} + \frac{1}{\sigma}\left(i_s(0) + \frac{L_m}{L_s} i_r(0)\right) e^{\frac{t}{T_\sigma}} (1 - e^{j\omega_m t}). \tag{21}$$

This equation shows, that after a current maximum (i_k'') which is determined by the initial currents and the leakage factor, the current declines with the total leakage time constant of the machine T_σ. In addition, the retaining stator voltage and the configuration of the rotor circuit have considerable influence on the short circuit behavior. [DITT05]

5.3.3 FRT Test cycle

As it has been mentioned before, the aim of the FRT test is verifying wind turbine response, and more specifically the response of the gearbox, to voltage drops due to grid failures. This test pretends also to be a basis for wind turbine numerical simulation model validation. Following the IEC 61400-21 standard, the FRT test will be carried out, and thus the response of the wind turbine will be checked, for the voltage drops specified in Table X and for two different operating conditions [IEC08]:

1. Between 0.1 P_n and 0.3 P_n to get the response at the most probable operational mode, depending on the wind conditions. The wind turbine in which is focused this project work has a nominal power of 2750 kW, so the mentioned limits will be: 275 kW – 825 kW

2. Above 0.9 P_n to get the response at tougher conditions. In particular, for the 27500 kW power wind turbine, this limit will be 2475 KW.

Each of the cases listed in **Table 5.2** will be tested twice by time series of active power, reactive power, active current, reactive current and voltage at the wind turbine terminals for the time shortly prior to the voltage drop and until the effect of the voltage drop has abated.

Table 5.2: Specification of voltage drops

Case	Magnitude of voltage phase to phase (fraction of voltage immediately before the drops occurs)	Magnitude of positive sequence voltage (fraction of voltage immediately before the drops occurs)	Duration (s)
VD1. Symmetrical three phase voltage drop	0.90 ± 0.05	0.90 ± 0.05	0.5 ± 0.02
VD2. Symmetrical three phase voltage drop	0.5 ± 0.05	0.5 ± 0.05	0.5 ± 0.02
VD3. Symmetrical three phase voltage drop	0.2 ± 0.05	0.2 ± 0.05	0.5 ± 0.02
VD4. Two-phase voltage drop	0.9 ± 0.05	0.95 ± 0.05	0.5 ± 0.02
VD5. Two-phase voltage drop	0.5 ± 0.05	0.75 ± 0.05	0.5 ± 0.02
VD6. Two-phase voltage drop	0.2 ± 0.05	0.6 ± 0.05	0.5 ± 0.02

It is important to remark that a voltage drop may cause a wind turbine to cut out for many reasons, not only related to the electrical drivetrain but also due to mechanical vibrations or ancillary system low voltage capabilities. It is therefore necessary to do the test on the complete wind turbine, as it can be done in a nacelle test rig. Moreover, it has to be mentioned that tests VD1 and VD4 are basically for testing wind turbines that have no capabilities to ride through any deep voltage drops. Therefore, in this project work will be prepared the VD2, VD3, VD5, VD6 tests. [IEC08]

The test can be carried out using for instance a set-up such as the one showed in **Figure 5.14**, which corresponds to an inductive voltage divider. The voltage drops will be created by the HIL Grid emulator that will connect the three or two phases to ground via an impedance, or connecting the three or two phases together through an impedance. [IEC08]

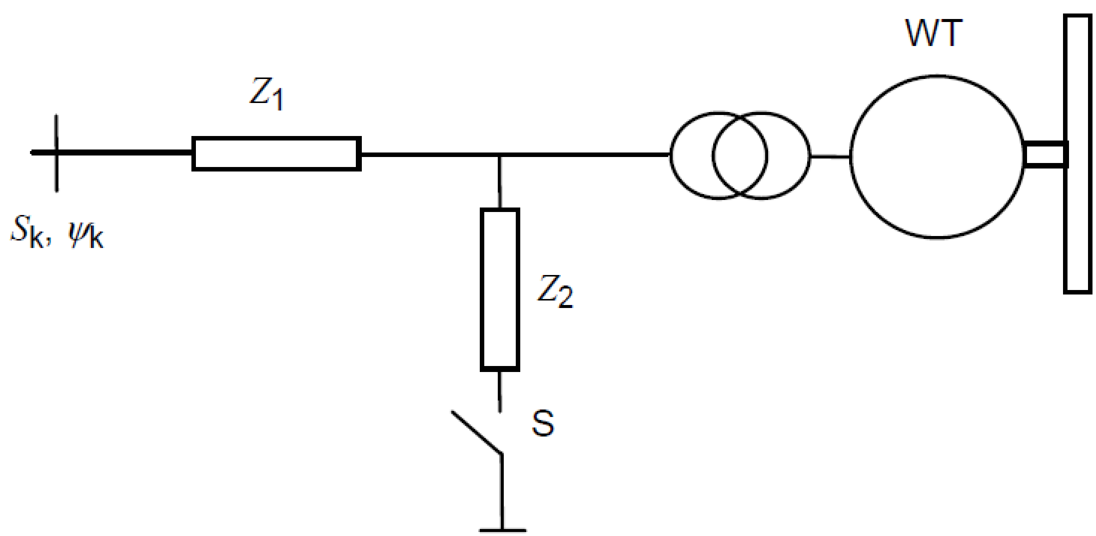


Figure 5.14: System with short circuit emulator for testing wind turbine response to temporary voltage drop

The impedance Z_1 is for limiting the effect of the short-circuit on the up-stream grid. The size of the impedance will be selected so that the voltage drops of the testing do not cause an unacceptable situation at the upstream grid and that at the same time do not affect significantly the transient response of the wind turbine. [IEC08]

The voltage drop will be created by connecting the impedance Z_2 by the switch S. The switch S, which can be a mechanical circuit breaker or a power electronic device, must be able to control in an accurate way the time between connection and disconnection of Z_2, and for all three or two phases. The size of Z_2 will be adjusted to give the voltage magnitudes specified in Table 5.3 when the wind turbine is not connected. [IEC08]

By changing the ratio Z_1 to Z_2 the depth of the voltage drop can be configured. Depending on the respective grid code, different depths of voltage drops have to be simulated. In this project work <5%, 25%, 50% and 75% of the rated voltage (720V) have been chosen as voltage drop depths.

Moreover, without being the turbine connected the voltage drop should have the shape indicated in **Figure 5.15**. The duration of the drop will be measured from closing to opening the switch S, which normally goes from several hundred milliseconds (deep drops) up to several seconds (flat drops). The time tolerance is included as to account for tolerance in operation of the switch S and that the positive sequence voltage will not drop or rise instantly, but with a slope.

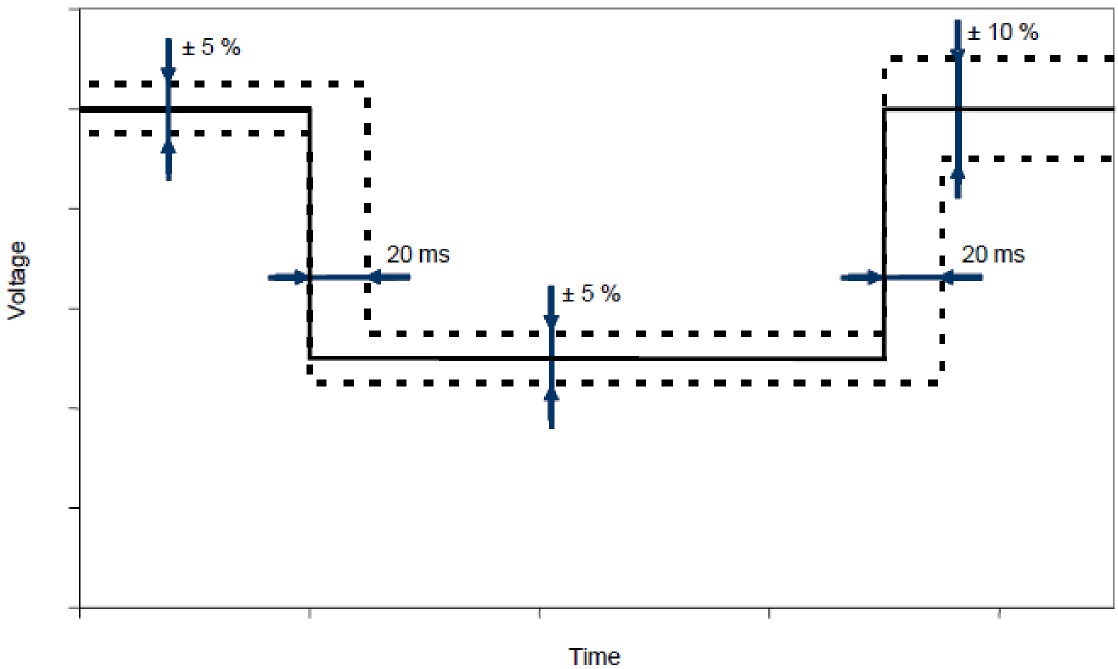


Figure 5.15: Tolerance of voltage drop

5.4 Single flank test under load

The single flank test is a standardized method which aim is to analyze the lack of smoothness of gear motion, known as transmission error. Transmission error is responsible for excitation of gear noise and problems of gear accuracy and sometimes has a relationship to gear failure. Single flank testing allows to evaluate the excitation of a single gear test without disturbing influence of other components and this test makes possible to identify the influences of flank deviations and modifications. [SMIT04]

For this test, network influences neither wind influences will be taken into account and the test will be carried out without the system controller and under quasi-static conditions.

During the single flank testing, gears roll together at their proper center distance and with only one flank in contact. During the measurement, one of the gears is driven while the other one is slightly braked in order to ensure constant contact between the cooperating sides of the gears. The rotational speed for the High Speed Shaft stablished for this test is normally really slow, between 60 and 100 rpm. In this case, it has been set up a rotational speed of 1 rpm for the input shaft of the planetary stage, which means that taking into account the gear ratio of all the gearbox, that is 62.775, the rotational speed for the High Speed Shaft during this test will be 62.775 rpm. Furthermore, the geometrical deviations of the gears will reveal themselves through the non-uniform movement of the passive gear due to the small rotational speed of the gears. [SMIT04]

In order to measure rotational motion (angular displacement error) optical encoders are used during the testing. These encoders are attached to the input and output shaft of the parallel stages. On the other hand, the transmission error will not be measured in the planetary stage during this test [SMIT04]. In **Figure 5.16** is shown the scheme of the gear stages with the three encoders that will be used for this test and its location on the gearbox.

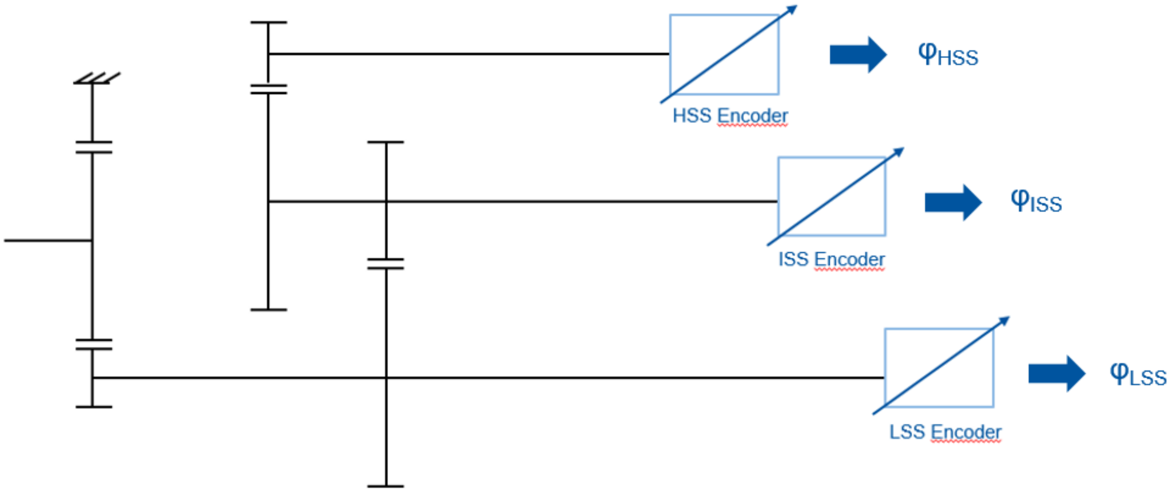


Figure 5.16: Scheme of the gear stages with the encoders

Moreover, in **Figure 5.17** is illustrated the measurement principle during the single flank test for each of the two parallel stages.

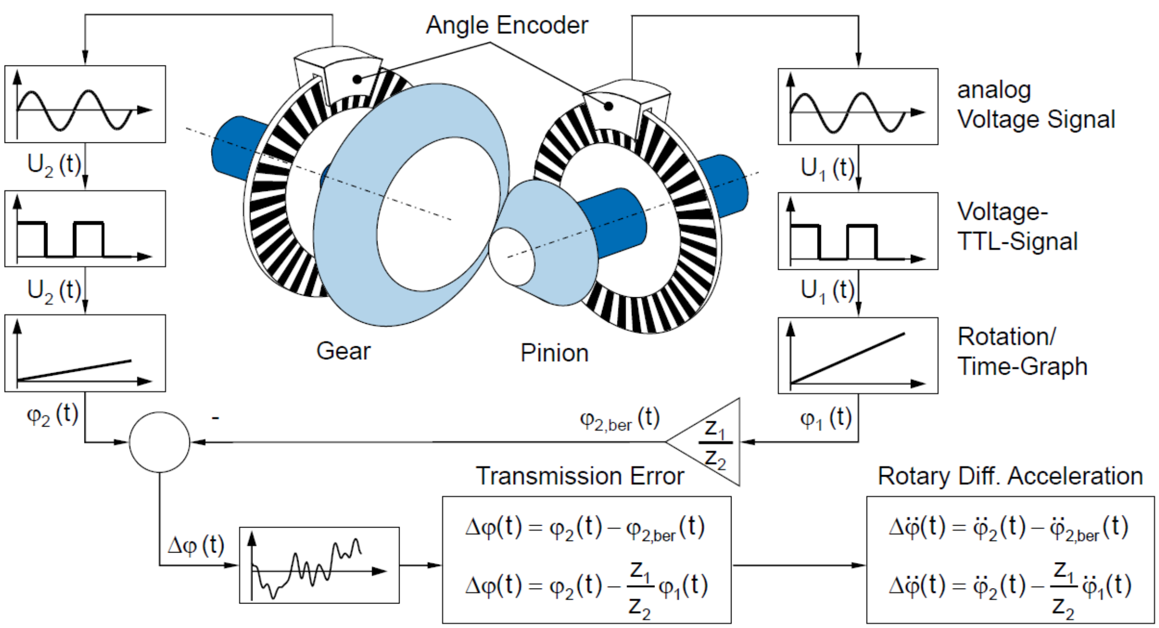


Figure 5.17: Measurement principle for the single flank test

The data that the encoders provide is processed in an instrument that shows the accuracy or smoothness of rotational motion resulting from the meshing of the gears, this is, transmission errors. Below is shown how the transmission error is obtained mathematically for the Intermediate Speed Stage and for the High Speed Stage. [SMIT04]

$$\Delta\varphi_{ISS} = \varphi_{ISS} - \varphi_{LSS} \cdot i_{IS} \quad (22)$$

$$\Delta\varphi_{HSS} = \varphi_{HSS} - \varphi_{ISS} \cdot i_{HS} \quad (23)$$

$\Delta\varphi_{ISS}$ = Transmission error of the Intermediate Speed Shaft

φ_{ISS} = Rotation of the Intermediate Speed Shaft

φ_{LSS} = Rotation of the Low Speed Shaft

i_{IS} = Gear ratio of the Intermediate Stage

$\Delta\varphi_{HSS}$ = Transmission error of the High Speed Shaft

φ_{HSS} = Rotation of the High Speed Shaft

i_{HS} = Gear ratio of the Intermediate Stage

The acquired data can be directly related to profile errors, pitch variation, runout… but it could be said that the most important aspect of single flank testing is that allows measurement of profile conjugacy, which is the closest parameter related to gear noise. However, another aspects of gears such as lead or tooth alignment variation cannot be measured directly in a single flank test. [SMIT04]

The next step to design the single flank test is defining the duration of the test. As it has been mentioned before a rotational speed of 1 rpm has been set up for the input shaft of the planetary stage, leading to a rotatory speed in the output shaft of the gearbox of 62.775 rpm. In order to assure a correct contact pattern between gears it has been stablished that two complete rounds (revolutions) of the gears are needed in both parallel stages. Mathematically it can be written as it is shown below:

$$2 \cdot (z_1 \cdot z_2) \quad (24)$$

z_1 = Teeth number of the pinion

z_2 = Teeth number of the wheel

In both **Table 5.3** and **Table 5.4** is shown the data of the two parallel stages that is needed to calculate the duration of the test.

Table 5.3: Data of the Intermediate Speed Stage

ISS	Wheel	Pinion	i	$z_1 \cdot z_2$
Teeth Number (z)	82	22	3.727	1804
Rotatory Speed (n)	5.68 rpm	21.19 rpm		

Table 5.4: Data of the High Speed Stage

HSS	Wheel	Pinion	i	$z_1 \cdot z_2$
Teeth Number (z)	80	27	2.963	2160
Rotatory Speed (n)	21.19 rpm	62.775 rpm		

In order to calculate the required time for the testing, first it is needed to calculate the number of revolutions that the wheel (or the pinion) makes in two complete rounds, for both stages:

$$rev = \frac{2 \cdot z_1 \cdot z_2}{z_2} = 2 \cdot z_1 . \qquad (25)$$

z_1 = Teeth number of the pinion

z_2 = Teeth number of the wheel

Once this calculation is done, the required time for the testing of each stages can be obtained by dividing the number of revolutions of the wheel into the rotatory speed of the wheel. Mathematically, this means:

$$t = \frac{rev}{n_2} ... \qquad (26)$$

t = Time in minutes

n_2 = Rotatory speed of the wheel in rpm

rev = Number of revolutions of the wheel

In **Table 5.5** are shown the numerical results obtained with the data provided at Tables 5.4 and 5.5.

Table 5.5: Results of the duration of the single flank test

	Z_1	n_2 (rpm)	rev	t (min)
ISS	22	5.68	44	7.75
HSS	27	21.19	54	2.55

Noticing that the Intermediate Speed Stage needs 7.74 minutes to complete two rounds of the gears, comparing to the High Speed Stage that achieves it in 2.55 minutes, it can be concluded that the Intermediate Stage is more restrictive. Taking into account this numbers, a duration of 8 minutes will be established for the single flank test.

Furthermore, as one of the aims of this project work is understand the behavior of the gearbox under different conditions and circumstances, this test will be done for different torque levels; from 10% of torque up to 120% in torque steps of 10%:

- $T_{10\%}$ = 2170 Nm
- $T_{20\%}$ = 47340 Nm
- $T_{30\%}$ = 6510 Nm
- $T_{40\%}$ = 8680 Nm
- $T_{50\%}$ = 10850 Nm
- $T_{60\%}$ = 13020 Nm
- $T_{70\%}$ = 15190 Nm
- $T_{80\%}$ = 17360 Nm
- $T_{90\%}$ = 19530 Nm
- $T_{100\%}$ = 21700 Nm
- $T_{110\%}$ = 23870 Nm
- $T_{120\%}$ = 26040 Nm

So, taking into account that at each torque level the duration of the test will be of 8 minutes, this would make a total duration of the single flank test of 96 minutes. However, another goal of this test is to check the contact patterns of gears so in order to achieve it, gears must be painted before testing at the desired torque levels and subsequently a photo must be taken in order to analyze the contact patterns. So, as this method is very time consuming but efficient, it would be done for the following torque levels: $T_{20\%}$, $T_{40\%}$, $T_{60\%}$, $T_{80\%}$, $T_{100\%}$ and $T_{120\%}$; being the estimated duration to do the contact patterns at each of those torque levels of two hours.

Moreover, as it has been explained in the speed run-ups test, before doing the testing at each of the torque levels, the temperature of bearings and the oil bulk must be stable, meaning that its temperature can only suffer a deviation of ±2K in 15 minutes. Therefore, before testing at each of the torque levels a warm up of the oil must be done. The first warm up, usually takes longer than the others, and even more at low torques, so it has been set up for it a duration of two hours before the testing at $T_{10\%}$. On the other hand, once the temperature has been stabilized once, the following warm ups before the other torque levels do not last so long, so it has been stablished a duration of 30 minutes for all of them.

So, the duration of the single flank test would not be only 96 minutes, taking into account the required time for the warm ups and for the contact pattern analysis. In **Figure 5.18** is illustrated a plot with the testing sequence, in which are distinguished for each torque level in red color the warm ups times, in blue color the single flank test times and in yellow color the contact pattern analysis times. The data used to build up this plot can be found in **Table VI.11** in the Annex of the document.

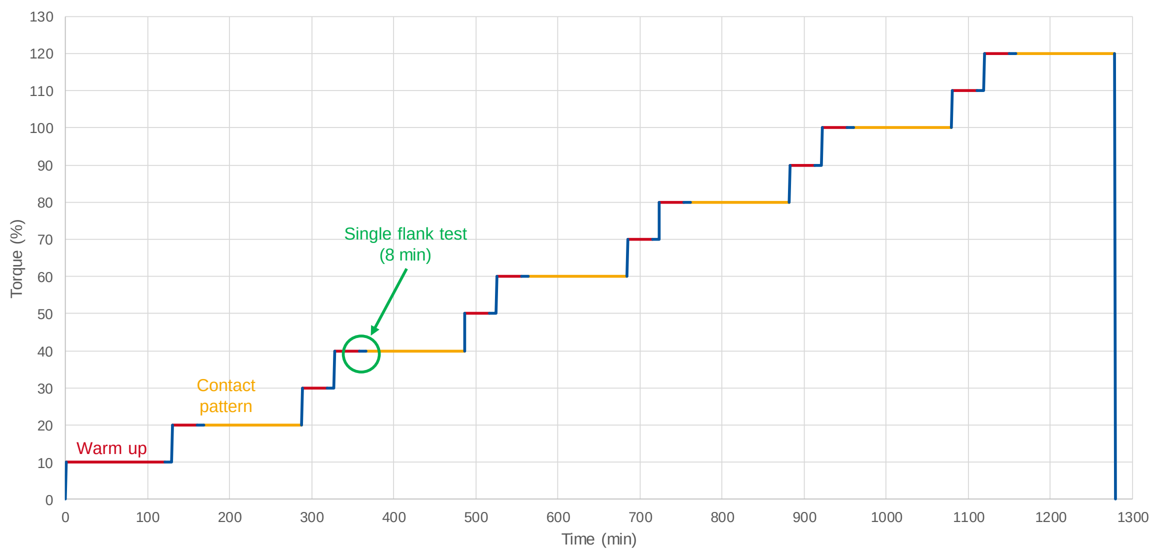


Figure 5.18: Plot with the testing sequence of the single flank test

So, the total duration of the single flank test would be:

- Single flank test: 8 min x 12 torque levels = 96 minutes
- Warm ups:
  - First warm up: 120 minutes
  - Rest of the warm ups: 30 min x 11 torque levels = 330 minutes
- Contact pattern analysis: 120 min x 6 torque levels = 720 minutes
- Start up, shut down and transition times between torque levels:
  - Start up: 1 minute
  - Shut down: 1 minute
  - Transition times between torque levels: 11 minutes

Summing all the durations mentioned above, makes a total duration of the single flank test of: 96 + 120 + 330 + 720 + 1 + 1 + 11 = 1279 minutes = 21.32 hours.

This test will be done also twice in order to verify the repeatability of the obtained results, do the total duration of the test would be 42.64 hours.

In **Figure 5.19** can be shown the aspect of the acquired data as result of the single flank test. In this plot is represented the transmission error along the different pitches of the gear. This plot will be obtained for the High Speed Shaft (HSS) and the Intermediate Speed Shaft (ISS), for the different torque levels. If gears only have profile error, the result will be the red one shown on the graphic, but otherwise if gears also present pitch variations, runouts… the result will be the blue one (in reality will be a curly line).

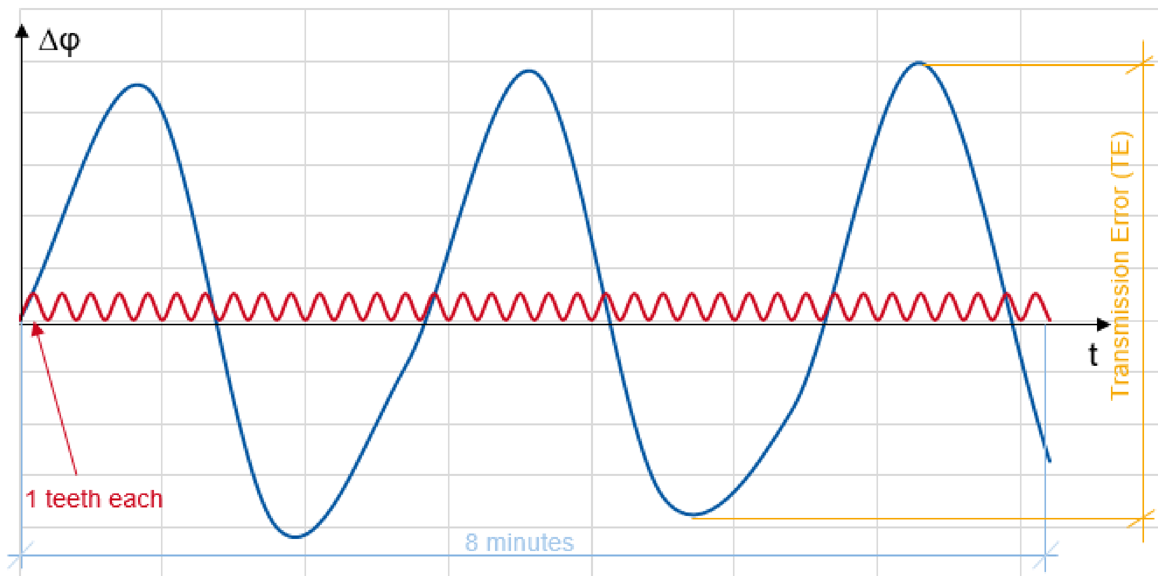


Figure 5.19: Plot with the expected results of the single flank test

The final aim of the single flank test is obtaining similar results to the ones obtained by the simulation done with STIRAK, which are the ones shown in **Figure 5.20**, for both the Intermediate Speed Stage (ISS) and the High Speed Stage (HSS). The two plots shown on the top illustrate the obtained shape of the transmission error in the simulation, varying the torque, while the two plots that are below them illustrate the pressure of the teeth in both stages for the different values of the torque. In blue color is represented the nominal design case, in green color is represented the ideal result that could be obtained in the single flank test and in red color is represented the worst result that could be obtained.

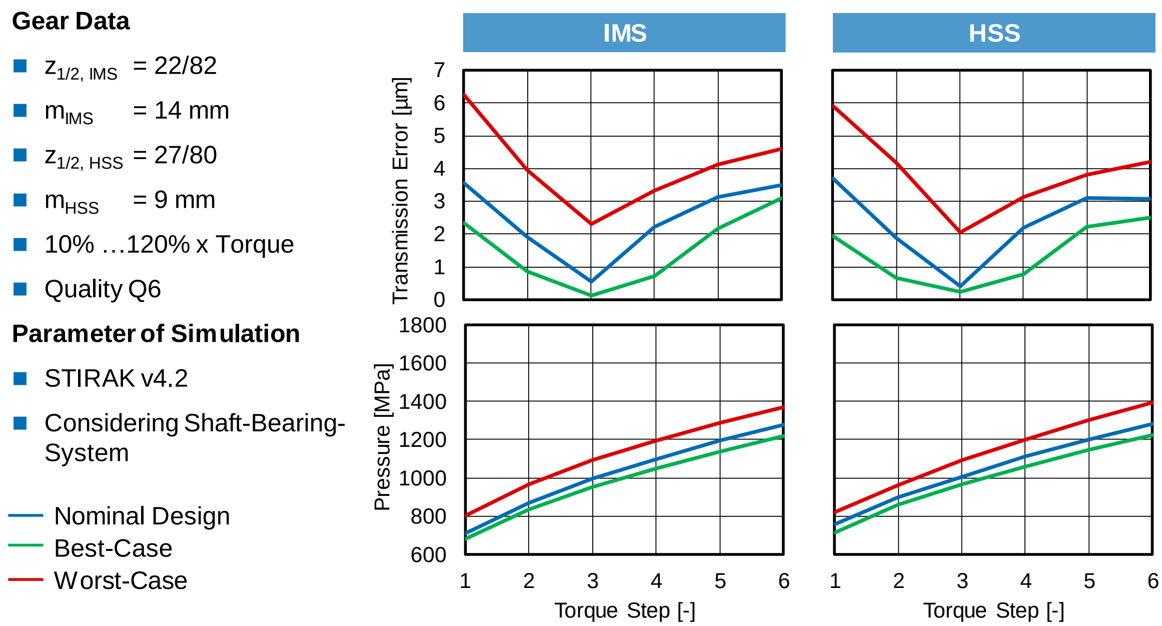


Figure 5.20: Results of the simulation of the single flank test

5.5 Test cycles duration

Finally, once the different tests that will be done in the test rig to test the behavior of the gearbox have been explained, a summary of the durations of all those tests can be done, as it can be seen in **Table 5.6**.

Table 5.6: Tests durations summary

Type of the test	Duration (h)	Duration in working days (h)
Speed run-ups	18.8	2,35
HIL Operation	94.64	11,58
Single flank test	42.6	5,325

Summing up all the durations the measurement campaign of the gearbox will last 156.04 hours. As a normal working day lasts 8 hours, the total duration of the measurement campaign of the gearbox will be 19.505 days.

6 Summary and Outlook

There is no single reason why wind turbine gearboxes fail prematurely, and it continues happening that wind turbine gearboxes tend to fail prematurely more than the gearboxes used in other applications. Furthermore, filed test are very expensive and time consuming, so it is believed that there is a necessity of testing complete nacelles of wind turbines as if they were on the field but within workshop conditions. This can be done in a nacelle test rig equipped with a multi-physics hardware in the loop system in which wind and grid conditions can be simulated, like the 4 MW capacity test rig installed in the CWD (Center for Wind Power Drives) in Aachen, Germany.

The premise of this project work has been defining the test cycles for the gearbox of a wind turbine that will be tested in the just mentioned nacelle test rig. As it does not exist a standard methodology to test a wind turbine or any of its components, the work described in this thesis pretends to be a structured methodology which can be implemented to test a gearbox of a wind turbine, in order to proof the correct operation of it and determine its critical operating conditions, before it is put in a field to produce power. In total, four different tests have been designed making a total duration of the measurement campaign of the gearbox of 19 days. In those tests, different parameters of the gearbox will be measured and analyzed. Firstly, In the speed run-ups tests the aim is getting a better understanding of the behavior if the gearbox for different torque and speed levels. Therefore, in this test only mechanical parameters of the wind turbine will be needed. On the other hand, with the HIL (Hardware In the Loop) operation test, it is believed that it is possible testing the wind turbine as if it was on the field producing power. Therefore, the aim of this second test, is analyzing how the gearbox behaves for different external conditions, thus, wind conditions. So, in this second test mechanical parameters of the gearbox will not be needed but wind parameters such as wind speed, turbulence intensity and direction of the flow will be considered. Moreover, these days the high penetration of wind energy into electrical grids dos not allow the disconnection of wind turbines in case of unusual voltage or frequency of the grid. This is why, grid code specifications require that wind turbines must be able to ride through all kinds of grid faults. In order to test the just mentioned scenario the FRT (Fault Ride Through) test has been designed in this project work, in which the wind turbine will suffer different voltage drops to analyze how the gearbox behaves in case of a grid fault. Finally, the single flank test has been designed to analyze the contact patterns of the gearbox and to measure the transmission error of the gears. Summarizing, it is believed that carrying out these four tests, the behavior of the gearbox of a wind turbine can be analyzed before it is placed in the field and for the different scenarios that can happen in the lifetime of a wind turbine.

The immediate continuation of the work presented in this thesis will consist of testing the wind turbine in which this project work is based in the nacelle test rig installed in the CWD in Aachen and comparing the obtained results with the once obtained with the simulation. Moreover, the next step would be obtaining a real wind profile for the HIL operation test instead of using a DLC based wind profile. In this way, more realistic results would be obtained and the wind turbine would be tested as if it was on a real and specific place. Nowadays, an increasing problem in wind turbines is the formation of ice in the rotor blades. Cold weather presents special problems for wind turbines. Inside the nacelle, low-viscosity lubricants keep the gearbox turning and enclosure seals to keep moisture and ice off electronic components. But outside the nacelle ice easily forms on the blades, adding hundreds of kilograms, which degrades performance and shortens working life of the wind turbine. The problem with ice on a working turbine is that it can be thrown, and when not, it causes additional drive train loads often in excess of design loads. For this reason, many turbines are shut down when ice buildup threatens and restart of the wind turbine only comes after an inspection confirms that ice is gone [HARP11]. Therefore, the continuation of this project work would be taking into account scenarios with ice conditions in the HIL operation mode and analyze the extra loads that ice introduce in the drivetrain, especially in the gearbox.

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VI Annex

Table VI.6.1: Data of the Speed run-ups test

Time (s)	HSS Speed (rpm)	HSS Speed (rad/s)	Torque (Nm*10)	Power (W)	Power (kW)
0	0	0	0	0	0
600	1100	115,192	533,6	614663,07	614,66
7800	1100	115,192	542,5	624915,14	624,92
7860	200	20,944	542,5	113620,93	113,62
7920	200	20,944	542,5	113620,93	113,62
8100	1200	125,664	542,5	681725,61	681,73
8120	1200	125,664	542,5	681725,61	681,73
8140	1100	115,192	1085	1249830,28	1249,83
9940	1100	115,192	1085	1249830,28	1249,83
10000	200	20,944	1085	227241,87	227,24
10060	200	20,944	1085	227241,87	227,24
10240	1200	125,664	1085	1363451,21	1363,45
10260	1200	125,664	1085	1363451,21	1363,45
10280	1100	115,192	1627,5	1874745,42	1874,75
12080	1100	115,192	1627,5	1874745,42	1874,75
12140	200	20,944	1627,5	340862,80	340,86
12200	200	20,944	1627,5	340862,80	340,86
12380	1200	125,664	1627,5	2045176,82	2045,18

12400	1200	125,664	1627,5	2045176,82	2045,18
12420	1100	115,192	2170	2499660,55	2499,66
14220	1100	115,192	2170	2499660,55	2499,66
14280	200	20,944	2170	454483,74	454,48
14340	200	20,944	2170	454483,74	454,48
14520	1200	125,664	2170	2726902,42	2726,90
14540	1200	125,664	2170	2726902,42	2726,90
14560	1100	115,192	2712,5	3124575,69	3124,58
16360	1100	115,192	2712,5	3124575,69	3124,58
16420	200	20,944	2712,5	568104,67	568,10
16480	200	20,944	2712,5	568104,67	568,10
16660	1200	125,664	2712,5	3408628,03	3408,63
16680	1200	125,664	2712,5	3408628,03	3408,63
16860	200	20,944	2712,5	568104,67	568,10
16920	0	0	0	0	0

Table VI.6.2: Data for the turbulence analysis of the first approach of the HIL operation test

Time (min)	Wind speed (m/s)	Turbulence bin
0	0	
10	4	0 % TURB
20	4	
30	20	
40	20	
50	25	
60	25	
70	20	0% TURB
80	20	5 % TURB
90	20	10 % TURB
100	20	15 % TURB
110	20	20 % TURB
120	20	25 % TURB
130	20	
140	10	0 % TURB
150	10	5 % TURB
160	10	10 % TURB
170	10	15 % TURB
180	10	20 % TURB
190	10	25 % TURB
200	10	
210	4	5 % TURB
220	4	
230	15	0 % TURB
240	15	5 % TURB
250	15	10 % TURB
260	15	15 % TURB
270	15	20 % TURB
280	15	25 % TURB
290	15	
300	20	0 % TURB
310	20	5 % TURB
320	20	10 % TURB
330	20	15 % TURB
340	20	20 % TURB
350	20	25 % TURB
360	25	
370	25	
380	0	
390	0	
400	4	10 % TURB
410	4	
420	10	0 % TURB
430	10	5 % TURB
440	10	10 % TURB
450	10	15 % TURB
460	10	20 % TURB
470	10	25 % TURB
480	10	
490	20	0 % TURB
500	20	5 % TURB
510	20	10 % TURB
520	20	15 % TURB
530	20	20 % TURB
540	20	25 % TURB

550	20	
560	15	0 % TURB
570	15	5 % TURB
580	15	10 % TURB
590	15	15 % TURB
600	15	20 % TURB
610	15	25 % TURB
620	15	
630	25	
640	25	
650	20	0 % TURB
660	20	5 % TURB
670	20	10 % TURB
680	20	15 % TURB
690	20	20 % TURB
700	20	25 % TURB
710	20	
720	10	0 % TURB
730	10	5 % TURB
740	10	10 % TURB
750	10	15 % TURB
760	10	20 % TURB
770	10	25 % TURB
780	10	
790	4	15 % TURB
800	4	
810	20	0 % TURB
820	20	5 % TURB
830	20	10 % TURB
840	20	15 % TURB

850	20	20 % TURB
860	20	25 % TURB
870	20	
880	15	
890	15	
900	0	
910	0	
920	4	20 % TURB
930	4	
940	15	0 % TURB
950	15	5 % TURB
960	15	10 % TURB
970	15	15 % TURB
980	15	20 % TURB
990	15	25 % TURB
1000	15	
1010	20	0 % TURB
1020	20	5 % TURB
1030	20	10 % TURB
1040	20	15 % TURB
1050	20	20 % TURB
1060	20	25 % TURB
1070	20	
1080	25	
1090	25	
1100	10	0 % TURB
1110	10	5 % TURB
1120	10	10 % TURB
1130	10	15 % TURB
1140	10	20 % TURB

1150	10	25 % TURB
1160	10	
1170	4	25 % TURB
1180	4	
1190	10	0 % TURB
1200	10	5 % TURB
1210	10	10 % TURB
1220	10	15 % TURB
1230	10	20 % TURB
1240	10	25 % TURB
1250	10	
1260	15	0 % TURB
1270	15	5 % TURB
1280	15	10 % TURB
1290	15	15 % TURB
1300	15	20 % TURB
1310	15	25 % TURB
1320	15	
1330	20	0 % TURB
1340	20	5 % TURB
1350	20	10 % TURB
1360	20	15 % TURB
1370	20	20 % TURB
1380	20	25 % TURB
1390	20	
1400	25	
1410	25	
1420	20	0 % TURB
1430	20	5 % TURB
1440	20	10 % TURB

1450	20	15 % TURB
1460	20	20 % TURB
1470	20	25 % TURB
1480	20	
1490	15	0 % TURB
1500	15	5 % TURB
1510	15	10 % TURB
1520	15	15 % TURB
1530	15	20 % TURB
1540	15	25 % TURB
1550	15	
1560	10	0 % TURB
1570	10	5 % TURB
1580	10	10 % TURB
1590	10	15 % TURB
1600	10	20 % TURB
1610	10	25 % TURB
1620	10	
1630	10	
1640	4	0 % TURB
1650	4	
1660	0	
1670	0	
1672	4	0 % TURB
1674	4	
1676	20	
1678	20	
1680	25	
1682	25	
1684	20	0% TURB

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<tr><td>1914</td><td>10</td><td>15 % TURB</td></tr>
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1926	15	10 % TURB
1928	15	15 % TURB
1930	15	20 % TURB
1932	15	25 % TURB
1934	15	
1936	20	0 % TURB
1938	20	5 % TURB
1940	20	10 % TURB
1942	20	15 % TURB
1944	20	20 % TURB
1946	20	25 % TURB
1948	20	
1950	25	
1952	25	
1954	20	0 % TURB
1956	20	5 % TURB
1958	20	10 % TURB
1960	20	15 % TURB
1962	20	20 % TURB
1964	20	25 % TURB
1966	20	
1968	15	0 % TURB
1970	15	5 % TURB
1972	15	10 % TURB
1974	15	15 % TURB
1976	15	20 % TURB
1978	15	25 % TURB
1980	15	
1982	10	0 % TURB
1984	10	5 % TURB
1986	10	10 % TURB
1988	10	15 % TURB
1990	10	20 % TURB
1992	10	25 % TURB
1994	10	
1996	10	
1998	4	0 % TURB
2000	4	
2002	0	

Table VI.6.3: Data for the flow direction analysis of the first approach of the HIL operati

Time (min)	Wind speed (m/s)	Flow direction analysis
0	0	
10	4	-5º
20	4	
30	20	
40	20	
50	25	
60	25	
70	20	-5º
80	20	-2.5º
90	20	0º

100	20	2.5º
110	20	5º
120	20	
130	10	-5º
140	10	-2.5º
150	10	0º
160	10	2.5º
170	10	5º
180	10	
190	4	-2.5º
200	4	
210	15	-5º
220	15	-2.5º
230	15	0º
240	15	2.5º
250	15	5º
260	15	
270	20	-5º
280	20	-2.5º
290	20	0º
300	20	2.5º
310	20	5º
320	25	
330	25	
340	0	
350	0	
360	4	0º
370	4	
380	10	-5º
390	10	-2.5º

400	10	0º
410	10	2.5º
420	10	5º
430	10	
440	20	-5º
450	20	-2.5º
460	20	0º
470	20	2.5º
480	20	5º
490	20	
500	15	-5º
510	15	-2.5º
520	15	0º
530	15	2.5º
540	15	5º
550	15	
560	25	
570	25	
580	20	-5º
590	20	-2.5º
600	20	0º
610	20	2.5º
620	20	5º
630	20	
640	10	-5º
650	10	-2.5º
660	10	0º
670	10	2.5º
680	10	5º
690	10	

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1614	10	2.5º
1616	10	5º
1618	10	
1620	4	2.5º
1622	4	

1624	20	-5º
1626	20	-2.5º
1628	20	0º
1630	20	2.5º
1632	20	5º
1634	20	
1636	15	
1638	15	
1640	0	
1642	0	
1644	4	5º
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1648	15	-5º
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1682	10	2.5º

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Table VI.6.4: Data of the TEST 1 of the second approach of the HIL operation test

Time (min)	Wind speed (m/s)	DLC	Direction	Turbulence
0	0	6.4	0°	0%
5	0			
10	0			
15	4	3.1		
20	4			
25	4			
30	10	1.1		
35	15			
40	15			
45	15			
50	18			5%
55	18			
60	18			
65	20			
70	20			
75	20			
80	24	1.5		
85	24			
90	24			
95	15	1.1		10%
100	15			
105	15			
110	4			
115	4			
120	4			
125	0	4.1 + 6.1		

130	0			
135	0			
140	4	3.2	-5°	5%
145	4			
150	4			
155	10	1.4	5°	
160	10			
165	10			
170	18		2.5°	
175	18			
180	18			
185	20		-2,5°	
190	20			
195	20			
200	22	2.3	5°	
205	22			
210	22			
215	0	5.1 + 6.2		
220	0			
225	0			
230	4	3.1	0°	25%
235	4			
240	4			
245	8	1.3		
250	8			
255	8			
260	16			
265	16			
270	16			
275	20			

280	20			
285	20			
290	24			
295	24			
300	24			
305	18			
310	18			
315	18			
320	12			
325	12			
330	12			
335	5			
340	5			
345	5			
350	0			
355	0	4.2 + 7.1		
360	0			
365	4			
370	4	3.3	2.5°	
375	4			
380	8			
385	8		5°	
390	8			
395	15			0%
400	15	1.2	-2.5°	
405	15			
410	18			
415	18		0°	
420	18			
425	20		5°	

430	20			
435	20			
440	25		2.5°	
445	25			
450	25			
455	12,5	4.1+6.3		
460	0			
465	0			
470	0			

Table VI.6.5: Data of the TEST 2 of the second approach of the HIL operation test

Time (min)	Wind speed (m/s)	DLC	Direction	Turbulence
470	0	3.1	5°	0%
475	4			
480	4			
485	4			
490	10	1.1		
495	15			
500	15			
505	15			
510	18			
515	18			
520	18			5%
525	20			
530	20			
535	20			
540	24	1.5		

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<tr><td>555</td><td>15</td><td rowspan="6">1.1</td><td rowspan="9">10%</td></tr>
<tr><td>560</td><td>15</td></tr>
<tr><td>565</td><td>15</td></tr>
<tr><td>570</td><td>4</td></tr>
<tr><td>575</td><td>4</td></tr>
<tr><td>580</td><td>4</td></tr>
<tr><td>585</td><td>0</td><td rowspan="3">4.1 + 6.1</td></tr>
<tr><td>590</td><td>0</td></tr>
<tr><td>595</td><td>0</td></tr>
<tr><td>600</td><td>4</td><td rowspan="3">3.2</td><td rowspan="3">-5°</td><td rowspan="18">10%</td></tr>
<tr><td>605</td><td>4</td></tr>
<tr><td>610</td><td>4</td></tr>
<tr><td>615</td><td>10</td><td rowspan="9">1.4</td><td rowspan="3">5°</td></tr>
<tr><td>620</td><td>10</td></tr>
<tr><td>625</td><td>10</td></tr>
<tr><td>630</td><td>18</td><td rowspan="3">2.5°</td></tr>
<tr><td>635</td><td>18</td></tr>
<tr><td>640</td><td>18</td></tr>
<tr><td>645</td><td>20</td><td rowspan="3">-2,5°</td></tr>
<tr><td>650</td><td>20</td></tr>
<tr><td>655</td><td>20</td></tr>
<tr><td>660</td><td>22</td><td rowspan="3">2.3</td><td rowspan="6">5°</td></tr>
<tr><td>665</td><td>22</td></tr>
<tr><td>670</td><td>22</td></tr>
<tr><td>675</td><td>0</td><td rowspan="3">5.1 + 6.2</td></tr>
<tr><td>680</td><td>0</td></tr>
<tr><td>685</td><td>0</td></tr>
<tr><td>690</td><td>4</td><td>3.1</td><td>5°</td><td>25%</td></tr>
</table>

<table>
<tr><td>695</td><td>4</td><td rowspan="2"></td><td rowspan="26"></td><td rowspan="26"></td></tr>
<tr><td>700</td><td>4</td></tr>
<tr><td>705</td><td>8</td><td rowspan="21">1.3</td></tr>
<tr><td>710</td><td>8</td></tr>
<tr><td>715</td><td>8</td></tr>
<tr><td>720</td><td>16</td></tr>
<tr><td>725</td><td>16</td></tr>
<tr><td>730</td><td>16</td></tr>
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<tr><td>745</td><td>20</td></tr>
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<tr><td>770</td><td>18</td></tr>
<tr><td>775</td><td>18</td></tr>
<tr><td>780</td><td>12</td></tr>
<tr><td>785</td><td>12</td></tr>
<tr><td>790</td><td>12</td></tr>
<tr><td>795</td><td>5</td></tr>
<tr><td>800</td><td>5</td></tr>
<tr><td>805</td><td>5</td></tr>
<tr><td>810</td><td>0</td><td rowspan="3">4.2 + 7.1</td></tr>
<tr><td>815</td><td>0</td></tr>
<tr><td>820</td><td>0</td></tr>
<tr><td>825</td><td>4</td><td rowspan="3">3.3</td><td rowspan="3">2.5°</td><td rowspan="4">5%</td></tr>
<tr><td>830</td><td>4</td></tr>
<tr><td>835</td><td>4</td></tr>
<tr><td>840</td><td>8</td><td>1.2</td><td>5°</td></tr>
</table>

845	8			
850	8			
855	15			
860	15		-2.5°	
865	15			
870	18			
875	18		0°	
880	18			
885	20			
890	20		5°	
895	20			
900	25			
905	25			
910	25			
915	12,5		2.5°	
920	0	4.1+6.3		
925	0			
930	0			

Table VI.6.6: Data of the TEST 3 of the second approach of the HIL operation test

<table>
<tr><th>Time (min)</th><th>Wind speed (m/s)</th><th>DLC</th><th>Direction</th><th>Turbulence</th></tr>
<tr><td>930</td><td>0</td><td rowspan="4">3.1</td><td rowspan="26">-5°</td><td rowspan="10">0%</td></tr>
<tr><td>935</td><td>4</td></tr>
<tr><td>940</td><td>4</td></tr>
<tr><td>945</td><td>4</td></tr>
<tr><td>950</td><td>10</td><td rowspan="10">1.1</td></tr>
<tr><td>955</td><td>15</td></tr>
<tr><td>960</td><td>15</td></tr>
<tr><td>965</td><td>15</td></tr>
<tr><td>970</td><td>18</td></tr>
<tr><td>975</td><td>18</td></tr>
<tr><td>980</td><td>18</td><td rowspan="7">5%</td></tr>
<tr><td>985</td><td>20</td></tr>
<tr><td>990</td><td>20</td></tr>
<tr><td>995</td><td>20</td></tr>
<tr><td>1000</td><td>24</td><td rowspan="3">1.5</td></tr>
<tr><td>1005</td><td>24</td></tr>
<tr><td>1010</td><td>24</td></tr>
<tr><td>1015</td><td>15</td><td rowspan="6">1.1</td><td rowspan="9">10%</td></tr>
<tr><td>1020</td><td>15</td></tr>
<tr><td>1025</td><td>15</td></tr>
<tr><td>1030</td><td>4</td></tr>
<tr><td>1035</td><td>4</td></tr>
<tr><td>1040</td><td>4</td></tr>
<tr><td>1045</td><td>0</td><td rowspan="3">4.1 + 6.1</td></tr>
<tr><td>1050</td><td>0</td></tr>
<tr><td>1055</td><td>0</td></tr>
</table>

1060	4	3.2	-5°	15%
1065	4			
1070	4			
1075	10	1.4	5°	
1080	10			
1085	10			
1090	18		2.5°	
1095	18			
1100	18			
1105	20		-2,5°	
1110	20			
1115	20			
1120	22	2.3	5°	
1125	22			
1130	22			
1135	0	5.1 + 6.2		
1140	0			
1145	0			
1150	4	3.1	-5°	25%
1155	4			
1160	4			
1165	8	1.3		
1170	8			
1175	8			
1180	16			
1185	16			
1190	16			
1195	20			
1200	20			
1205	20			

<table>
<tr><td>1210</td><td>24</td><td rowspan="12"></td><td rowspan="15"></td><td rowspan="15"></td></tr>
<tr><td>1215</td><td>24</td></tr>
<tr><td>1220</td><td>24</td></tr>
<tr><td>1225</td><td>18</td></tr>
<tr><td>1230</td><td>18</td></tr>
<tr><td>1235</td><td>18</td></tr>
<tr><td>1240</td><td>12</td></tr>
<tr><td>1245</td><td>12</td></tr>
<tr><td>1250</td><td>12</td></tr>
<tr><td>1255</td><td>5</td></tr>
<tr><td>1260</td><td>5</td></tr>
<tr><td>1265</td><td>5</td></tr>
<tr><td>1270</td><td>0</td><td rowspan="3">4.2 + 7.1</td></tr>
<tr><td>1275</td><td>0</td></tr>
<tr><td>1280</td><td>0</td></tr>
<tr><td>1285</td><td>4</td><td rowspan="3">3.3</td><td rowspan="3">2.5°</td><td rowspan="15">10%</td></tr>
<tr><td>1290</td><td>4</td></tr>
<tr><td>1295</td><td>4</td></tr>
<tr><td>1300</td><td>8</td><td rowspan="12">1.2</td><td rowspan="3">5°</td></tr>
<tr><td>1305</td><td>8</td></tr>
<tr><td>1310</td><td>8</td></tr>
<tr><td>1315</td><td>15</td><td rowspan="3">-2.5°</td></tr>
<tr><td>1320</td><td>15</td></tr>
<tr><td>1325</td><td>15</td></tr>
<tr><td>1330</td><td>18</td><td rowspan="3">0°</td></tr>
<tr><td>1335</td><td>18</td></tr>
<tr><td>1340</td><td>18</td></tr>
<tr><td>1345</td><td>20</td><td rowspan="3">5°</td></tr>
<tr><td>1350</td><td>20</td></tr>
<tr><td>1355</td><td>20</td></tr>
</table>

<table>
<tr><td>1360</td><td>25</td><td rowspan="3"></td><td rowspan="7">2.5°</td><td rowspan="7"></td></tr>
<tr><td>1365</td><td>25</td></tr>
<tr><td>1370</td><td>25</td></tr>
<tr><td>1375</td><td>12,5</td><td rowspan="4">4.1+6.3</td></tr>
<tr><td>1380</td><td>0</td></tr>
<tr><td>1385</td><td>0</td></tr>
<tr><td>1390</td><td>0</td></tr>
</table>

Table VI.6.7: Capture Matrix for normal power production for the HII operation test

NORMAL POWER PRODUCTION **Wind Speed bin size: 1 m/s** **Turbulence bin size: 2 %**																						
Time series length	10 min																2 min					
Wind (m/s) → I (%)↓	v_{in} 4	5	6	7	v_{ave} 8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	v_{out} 25
<3	1	1	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
3-5	1	2	2	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
5-7	4	15	10	6	1	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
7-9	8	35	30	20	30	15	10	5	4	1	1	1	1	1	1	1	1	0	0	0	0	0
9-11	12	50	60	70	80	50	25	15	10	8	7	6	5	4	4	2	2	1	0	0	0	0
11-13	15	60	120	170	150	100	50	40	30	20	18	15	10	6	6	4	4	4	2	2	2	0
13-15	20	100	150	190	200	150	100	95	90	85	80	75	70	65	60	50	40	20	6	2	1	1
15-17	18	80	100	140	150	130	110	80	70	65	55	50	45	40	30	25	20	10	6	2	0	0
17-19	15	50	80	90	100	70	60	50	40	30	20	15	10	10	8	6	6	4	1	0	0	0
19-21	10	20	40	50	50	30	20	10	6	5	5	4	4	2	2	1	1	1	0	0	0	0

21-23	5	6	10	20	25	15	10	5	1	1	1	1	1	1	1	0	0	0	0	0	0	0
23-25	3	3	6	9	10	5	1	1	0	0	0	0	0	0	0	0	0	0	0	0	0	0
25-27	1	1	1	1	1	1	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
27-29	1	1	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
>29	1	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
Total number	115	424	609	766	797	566	386	301	251	215	187	167	146	129	112	89	74	40	15	6	3	1

Table VI.6.8: Capture Matrix for power production plus occurrence fault for the HIL test

POWER PRODUCTION PLUS OCCURRENCE FAULT			
Time series record length	2 min	2 min	2 min
Wind (m/s) → Fault condition↓	4 < v < 18	18 < v < 22	22 < v
Fault No. 1	2	2	2
Fault No. 2	2	2	2
….			
Fault No. N	2	2	2

Table VI.6.9: Capture Matrix for the start-up and shut-down operations for the HIL test

NORMAL START-UP AND SHUT DOWN EVENTS				
Event		4 < v < 18	18 < v < 22	22 < v
Start - up	Recommended number	3	-	3
	Actual wind Speed (m/s)			
Normal shut-down	Recommended number	3	3	3
	Actual wind Speed (m/s)			

Table VI.10: Capture Matrix other transient events for the HIL test

OTHER TRANSIENT EVENTS				
Event		Most critical wind speed		
Grid Failure	Recommended number	3		
	Actual measured wind speed (m/s)			
Emergency shut down	Recommended number	3		
	Actual measured wind speed (m/s)			
Overspeed combinations	Recommended number	3		
	Actual measured wind speed (m/s)			
Other design critical transients	Recommended number	3		
	Actual measured wind speed (m/s)			

Table VI.11: Data for the test cycle of the Single Flank test

Time (min)	Torque (%)	Operation
0	0	
1	10	
121	10	warm up
129	10	test
130	20	
160	20	warm up
168	20	test
288	20	contact pattern
289	30	
319	30	warm up
327	30	test
328	40	
358	40	warm up

366	40	test
486	40	contact pattern
487	50	
517	50	warm up
525	50	test
526	60	
556	60	warm up
564	60	test
684	60	contact pattern
685	70	
715	70	warm up
723	70	test
724	80	
754	80	warm up
762	80	test
882	80	contact pattern
883	90	
913	90	warm up
921	90	test
922	100	
952	100	warm up
960	100	test
1080	100	contact pattern
1081	110	
1111	110	warm up
1119	110	test
1120	120	

1150	120	warm up
1158	120	test
1278	120	contact pattern
1279	0	