Final report

1. Project details

Project title	DMDS – Demonstration of Modular Drive System
File no.	4395-900-DOC
Name of the funding scheme	EUDP
Project managing company / institution	R&D test systems
CVR number (central business register)	37844136
Project partners	DTU Mechanical Engineering
Submission date	27 February 2023

2. Introduction

In order to validate the design, performance and safety of a wind turbine before entering service, a test of the wind turbine components must be conducted. In addition to full scale blade testing, the test of a complete wind turbine nacelle is the most critical in the validation scheme for a new wind turbine design. A full-scale test of a nacelle requires a test bench capable of replicating both the drive torque and the appertaining wind loads from the turbine blades. Test benches capable of replicating realistic operation conditions are today very limited with less than a handful in operation worldwide. A large portion of the cost for building a full-scale test rig, are driven by the cost of the large capacity direct drive motors which provide the drive torque of the test bench.

The rapid development of new and continuously larger wind turbines is driving industry demand for lowering CoE through larger and more efficient turbine design, while achieving minimal design risk. Focus on wind turbine system and component cost optimisation, reliability increase, energy output optimisation and risk mitigation all calls for verification by extensive test programmes. A full-scale wind load simulator for test of complete wind turbine drivetrains, which meets the demands of the wind turbine industry both today and in the future and which is economically and technically feasible fulfils this need, and that is the focus for this project.

Current test benches for full scale nacelle testing has been available to the wind turbine industry to a limited extent, and presently only a limited number of facilities exist worldwide, which can perform wind load application test of wind turbine nacelles above 8 MW. These drive systems rely on very large direct drive motors to supply the needed drive torque of the test bench. The large drive capacity demand that the direct drive motors used in the test bench motors have capacity of more than 5MW per motor. The limited supply of these motors, consequently drive price and delivery times, which heavily influence the overall cost and lead time of full scale nacelle test benches.

2.1 Executive summary (English):

The project focus is to make large scale nacelle tests technologically and economically feasible for the industry, by developing and demonstrating a new modular drive technology. A Modular Drive System (MDS) utilizing several drive motors connected to a large ring gear is investigated alongside a Direct Drive (DD) which relies on several stator segments joined to form a single stator to drive a rotor. A demonstrator concept description and a requirement specification for both the MDS and DD system is delivered.

For the MDS focus is put on identifying suitable gear materials, bearings, and lubrication system. A rolling contact fatigue test concludes that the selected material 42CrMo4 in an induction hardened condition is very well suited for application in the MDS. Additionally it is concluded that gear properties and service life is greatly improved when having a thick and hard surface layer, a wide transition to the soft matrix and proper annealing.

For the DD system focus is put on stator segment design and verification of cooling system and electrical performance. Therefore, a demonstrator concept consisting of a full-scale stator segment with a replica of the machine cooling paths is manufactured and tested. Both the electrical and cooling tests confirms that the full-scale DD coil segment design performs as intended and can be cooled during operation with a combination of air cooling and water cooling. The DD design is therefore considered feasible for full scale implementation in a DD driveline solution for a 25 MW Nacelle test bench. The DD design was however not developed further within the EUDP project but was realized in a separate commercially funded project with LORC.

The main commercial result for R&D from the project is that the verification of the full-scale DD coil segment provided the confidence required to utilize it in a 25MW DD driveline design for a nacelle test bench at the Lindø Offshore renewables Center (LORC). Main commercial result from DTU is the submission for a paper for the 42nd Risø International Symposium as well as an additional spin off project.

2.2 Executive summary (Danish):

Projektets fokus er at gøre nacelle test i stor skala både teknologisk og økonomisk mulige for industrien ved at udvikle og demonstrere en ny modulær drivteknologi. Et modulært drivsystem (MDS), der anvender flere drivmotorer forbundet til et stort ringgear, undersøges sammen med et Direct Drive (DD), som opbygges af flere stator segmenter, der tilsammen danner en enkelt stator til at drive en rotor. Der leveres en konceptbeskrivelse og en kravspecifikation til en demonstrator for både MDS og DD systemet.

Fokus for MDS-systemet lægges på at identificere egnede gearmaterialer, lejer og smøresystemer. En ”rolling contact fatigue test” konkluderer, at induktionshærdet 42CrMo4 er meget velegnet til anvendelse i MDS-systemet. Derudover konkluderes det, at gearegenskaber og levetid forbedres væsentligt, når man har et tykt og hårdt overfladelag, en bred overgang til den bløde matrix og korrekt udglødning.

For DD-systemet fokuseres der på design af stator segment og verifikation af det tilhørende kølesystem og stator segmentets elektrisk ydeevne. Derfor fremstilles og testes en demonstrator bestående af et stator segment i fuld skala indeholdende passende kølekanaler. Både de elektriske test og køle test bekræfter, at DD-stator segmentet fungerer efter hensigten og at dette kan køles under drift med en kombination af luftkøling og vandkøling. DD-designet anses derfor for velegnet til implementering i en DD-drivlinje til en 25 MW Nacelle-testbænk. DD-designet blev imidlertid ikke videreudviklet i dette EUDP projekt men blev realiseret gennem et seperat kommercielt projekt med LORC

Det vigtigste kommercielle resultat for R&D fra projektet er, at verifikationen af DD-spolesegmentet i fuld skala muliggjorde at bruge et 25MW DD-drivlinje design til en nacelletestbænk ved Lindø Offshore Renewables Center (LORC). Det primære kommercielle resultat fra DTU er indsendelsen af et paper til det 42. Risø Internationale Symposium samt yderligere et spin off-projekt.

3. Project objectives

The project objective has been to make large scale nacelle test facilities technologically and economically feasible for the industry, by developing and demonstrating a new modular drive technology. The incentive of the project was to expand market potential for test facility business and lowering the Cost of Energy (LCoE) by facilitating current and future test needs of the wind industry.

The main deliverable of the project was to demonstrate a drive technology which mitigates the main disadvantages of the current drive systems used on full scale nacelle test benches. Two different Drive technologies were chosen to pursue: The Modular Drive System (MDS) and the Direct Drive System (DD).

3.1 Objectives from Work Packages

WP0: Project management

The objective of WP0 is to ensure a successful execution of the project, with emphasis on achieving the projected milestones and deliverables within the allocated timeframe and budget.

WP1: Demonstrator concept and specification

The objective of WP1 is to develop the concept and specification of a MDS and DD demonstrator which can experimentally validate the MDS concepts for full scale application. During the work package the main technical challenges will be identified and provide input for the data validation conducted in WP2 for the MDS concept and the detailed design for both the MDS and DD concept in WP3.

WP2: Material testing and data validation

The objective of WP2 is to validate material and tribological data on which the design of the demonstrator is based. This is done through both literature studies and through experimental work using different types of methods for characterizing material and tribological behavior of the gear connections in the MDS concept.

WP3: Demonstrator design, manufacturing and installation

The objective of WP3 is twofold. For the MDS the objective is to detail the design of the MDS concept using the requirement specification of WP1 and the findings of WP2. For the DD system the objective is to finalize the design of the demonstrator based on the findings of WP2. The final design will then be validated and followingly issued for manufacturing, installation and testing.

WP4: Demonstrator testing

The overall objective of WP4 is twofold: For the MDS the objective is to perform an analysis of the loaded gear components. For the DD system the objective is to realize and test the DD demonstrator.

WP5: Data and material analysis and evaluation

The objective of WP5 concerns investigation of material failure modes and performance through comparison of components after characterization. Combined with the analysis and simulation results from WP3, these findings form the basis for an overall assessment of the MDS concept validity and recommendations for further concept development and maturing. Similarly, the results from the DD demonstrator testing are considered to perform an overall assessment of the DD concept validity and recommendations for further concept development and maturing.

The final objective of the project is to compare the MDS concept and DD concepts for the purpose of evaluating the project results and the application of the concepts in full scale nacelle testbenches.

3.2 Modular Drive System (MDS)

The Modular Drive System (MDS), relies on several drive motors connected to a large ring gear which drives the main shaft of the test bench as illustrated in Figure 1-1. Here a rendering of a possible concept for MDS implementation in a full scale test bench driveline is shown.

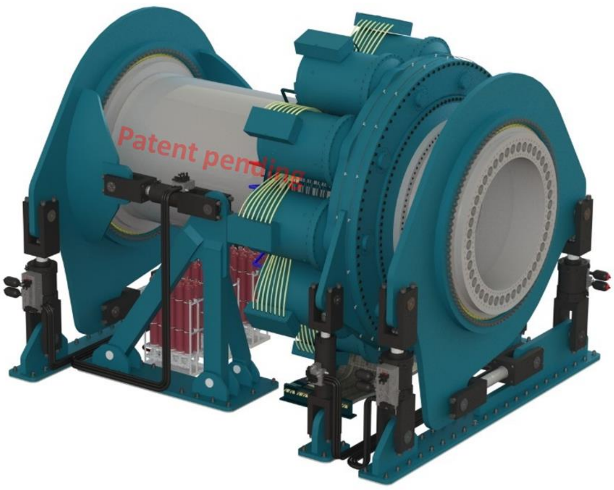

Figure 1-1: Rendering of possible concept for MDS implementation in full scale test bench driveline.

3.3 Direct Drive System (DD)

The Direct Drive system relies on several stator segments joined to form a single stator to drive a rotor connected to the main shaft of the test bench as illustrated in Figure 1-2, which shows a rendering of a possible concept for DD implementation in a full-scale test bench driveline.

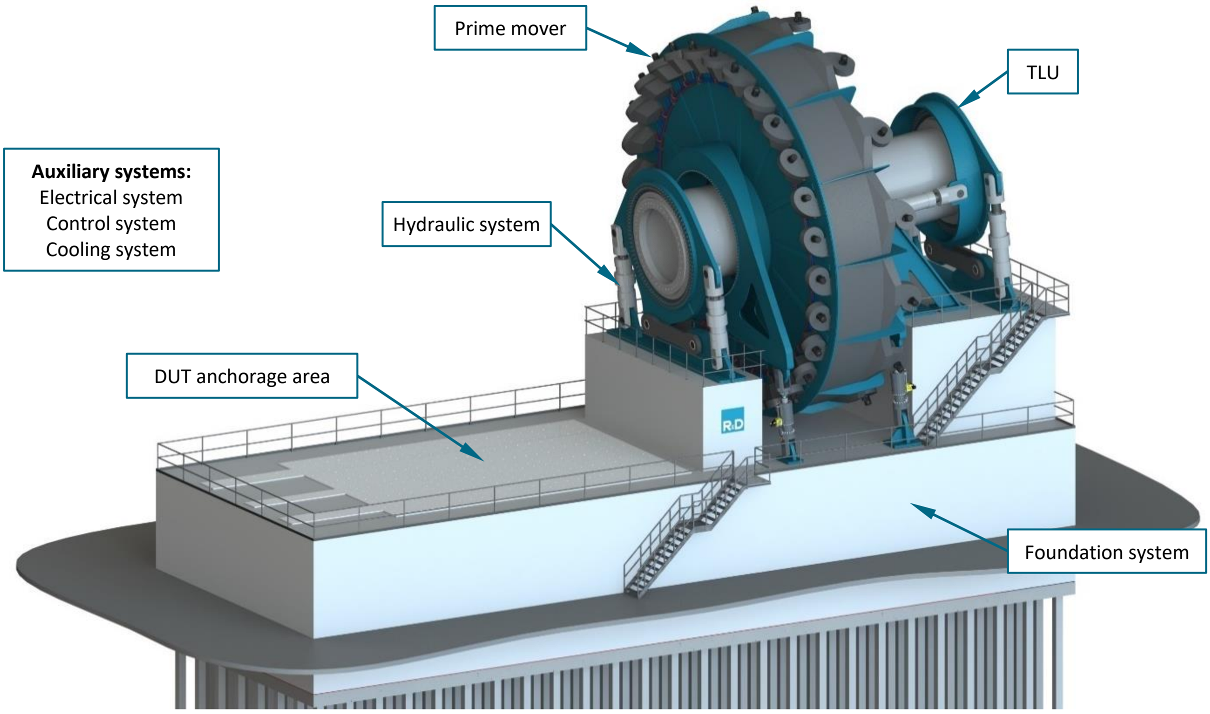

Figure 1-2: Rendering of possible concept for DD implementation in full scale test bench driveline.

4. Project implementation

The DMDS project originally aimed to develop and demonstrate a Modular Drive System as the drive motor for large scale nacelle test benches. The project did however change two times during the duration of the project due to increased knowledge obtained through the project work.

- In March 2019 a concept for designing a comparable more cost-effective large-scale DD electric motor with a very high capacity was identified and added to the scope of the project. The DD activities added to the scope included design and manufacture of a single segment for the DD motor to validate both thermal- and electrical characteristics as well as manufacturing processes and quality.
- In March 2021 it was identified that the scaled demonstrator of the MDS originally would not yield the results expected.
  - The reason for this is that the failure mechanisms that were identified in the gear-interfaces are highly dependant on geometry and loads and therefore a scaled demonstrator would not be representative to the failure mechanisms present in a full scale MDS solution.
  - Additionally the large technical and economical risks with the MDS technology caused a shift of focus to the DD technology.
  - Therefore, a more detailed theoretical investigation of the technical risks associated with the MDS concept is performed by R&D with focus on bearing design, system dynamics and lubrication design. In addition to this DTU performs additional lab-tests to investigate the failure mechanisms present in materials used for pinions and main gear.

5. Project results

In this section the obtained technological and commercial results are described. As described in the previous section the scope of the project was change two times and the work packages covered in section 3.1 reflects these changes. The following sections 5.1 through 5.6 covers each of these work packages and describes the results in detail.

5.1 WP0: Project management

The results from WP0 should be a successful project execution. From R&D perspective the project has been managed with great care. It is due to the effort of the project manager that the scope of the project has been modified based on new knowledge for optimized commercial and technological relevance. The modifications of scope to add the DD technology to scope and reorder funds from the MDS demonstrator into further theoretical investigations has ensured commercial and technological relevance and optimal usage of funds.

As a result all milestones and deliverables in the revised project plan were met as summarized in Table 1-1 below.

Milestones	**Status**
M1: Requirement specification for demonstrators completed	Completed
M2: Design description of demonstrators completed	Completed
M3: Final design review of MDS concept	Completed
M4: Conclusion on technical design risks	Completed
M5: Completion of demonstrator tests	Completed
M6: Component analysis completed	Completed
M7: Design evaluation of demonstrators and concepts completed	Completed
M8: Final design review of DD demonstrator	Completed
M9: FAT/SAT of DD demonstrator	Completed
Commercial milestones	
CM1: Final revision of marketing plan	Completed
CM2: Global "Release for marketing"	Completed
CM3: Global "Release for sale"	Completed
Deliverables	
D1a: Requirement specification for demonstrators	Completed
D1b: MDS - Demonstrator concept description and load specification	Completed
D1c: MDS - Specification of gear design	Completed
D1d: MDS - Specification of lubrication system	Completed
D1b: DD - Demonstrator concept description	Completed
D1f: DD - Specification of cooling system	Completed
D2a: Specification for gear material and treatment	Completed
D2b: Report on data validation findings (DTU)	Completed
D3a: Commissioned DD demonstrator	Completed
D4a: DD Demonstrator test results	Completed
D5a: Report on component characterization (DTU)	Completed
D5b: Report on design evaluation and overall project findings	Completed

Table 1-1: Overview and status on all milestones and deliverables in the project.

5.2 WP1: Demonstrator concept and specification

The following sections contain an abstract of the contents of deliverable D1a through D1f.

5.2.1 D1a: Requirement specification for demonstrators

5.2.1.1 MDS overview

The main components of the MDS are displayed in Figure 1-3. The MDS acts as the Test Bench Prime mover, applying torque (and rotational movement) to the Dynamic Wind Load Module System (rotational DOF around the x-axis). In the following sections the primary MDS components and functions are explained together with the torque pathway.

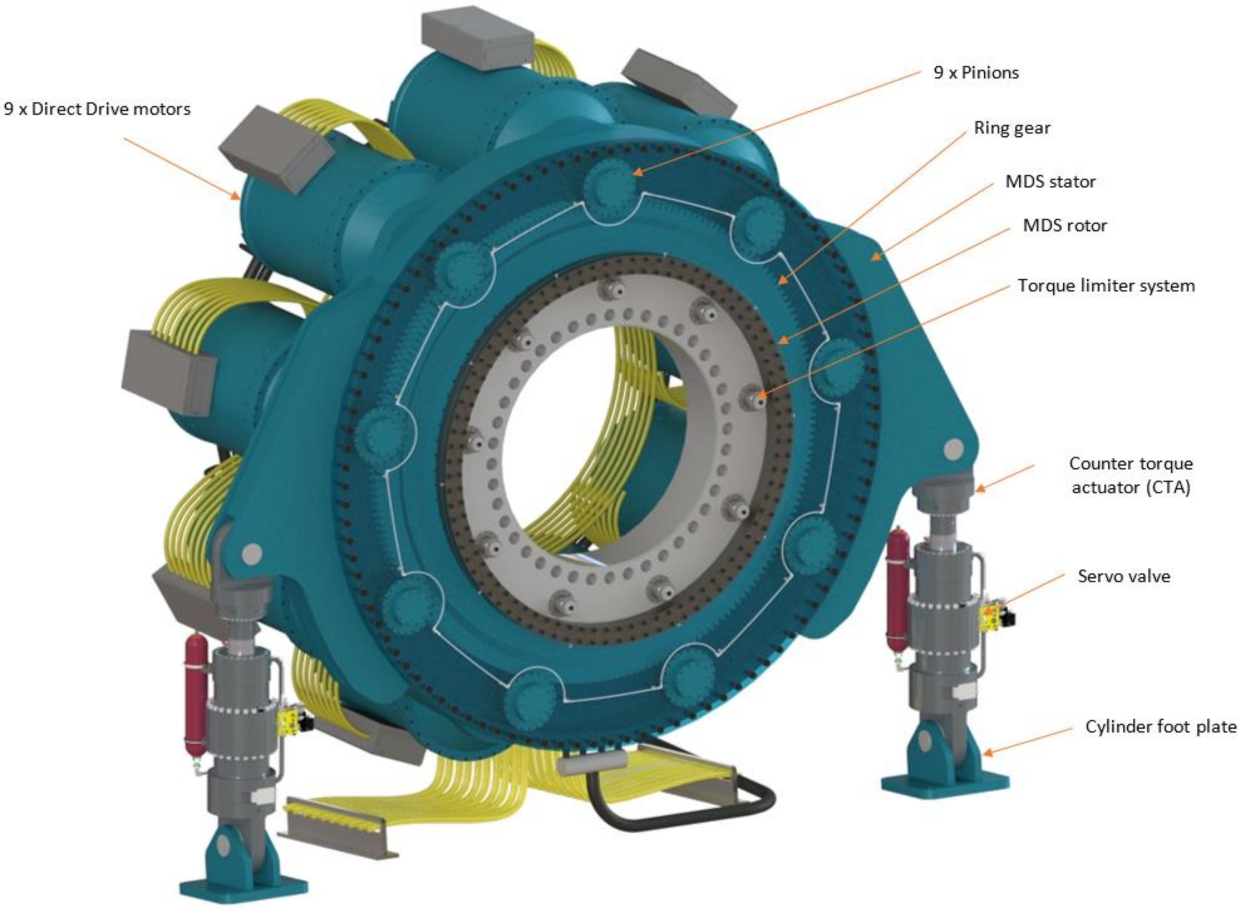

Figure 1-3 MDS drive system concept

Stator and rotor connection

The Motor and the pinion are connected by a shaft with a female (motor side) and a male (pinion side) spline connection, which transfers the torque to a gear connection. The design of the spline connection is performed, such that minor misalignments and deformations can be taken without introducing restoring forces to the system. The pinion and the stator/rotor are supported by bearings. For further details please refer to deliverable D1a.

MDS demonstrator requirement specification

The requirement specification of the MDS demonstrator shown in Table 1-2 is based on the risks identified and assessed in the design FMEA performed in D1a. For each of the risks a recommended action / mitigation has been listed. The risks and the appertaining mitigations form the requirement specification of the DD motor demonstrator:

Failure mode	**Causes of failure**	**Effects of failure on syst. / part / operation**	**Recommended action / mitigation**	**Demonstrator requirement**
Material failure	Insufficient material quality	Gear drive failure PM breakdown	Prototype to test for specific material failure mode	Test of relevant material failure modes Inspection during testing
Structural failure from vibrations levels and dynamics	Vibrations level too high	Structural failure PM breakdown	Prototype to test for system structural dynamics	Validation/measurement of vibration levels Test of failure modes Inspection during testing
Controller, electric system, converter fails	Wear out, short circuit, temperature	Test rig standstill	Prototype test of motor controls	Validation of control system Measurement of motor performance
Distortion/deformation of gear drive housing structure	Vibrations level too high	Influence on gear mesh Increased wear Component replacement Test rig standstill	Prototype to test for structural deformation influence on gear mesh	Variable meshing of gear drive Inspection during testing
Insufficient or poor lubrication	Wear	Increased wear PM breakdown	Prototype testing of lubrication system design	Validation of lubrication system design Wear monitoring of gear drive components.

Table 1-2: Results from the Design FMEA and demonstrator requirements for the MDS

5.2.1.2 DD system overview

The DD motor delivers a drive torque and rotation to a DUT (Device Under Test) on a full-scale test rig. The prime mover system including the DD motor is illustrated in Figure 1-4 and consists of the following main components.

Prime mover system main components:

- Rotor structure with magnets
- Segmented stator structure with coils
- Internal cooling circuit with water/air heat exchanger system
- External ventilation system
- Torque arm structure
- Hydraulic torque actuators

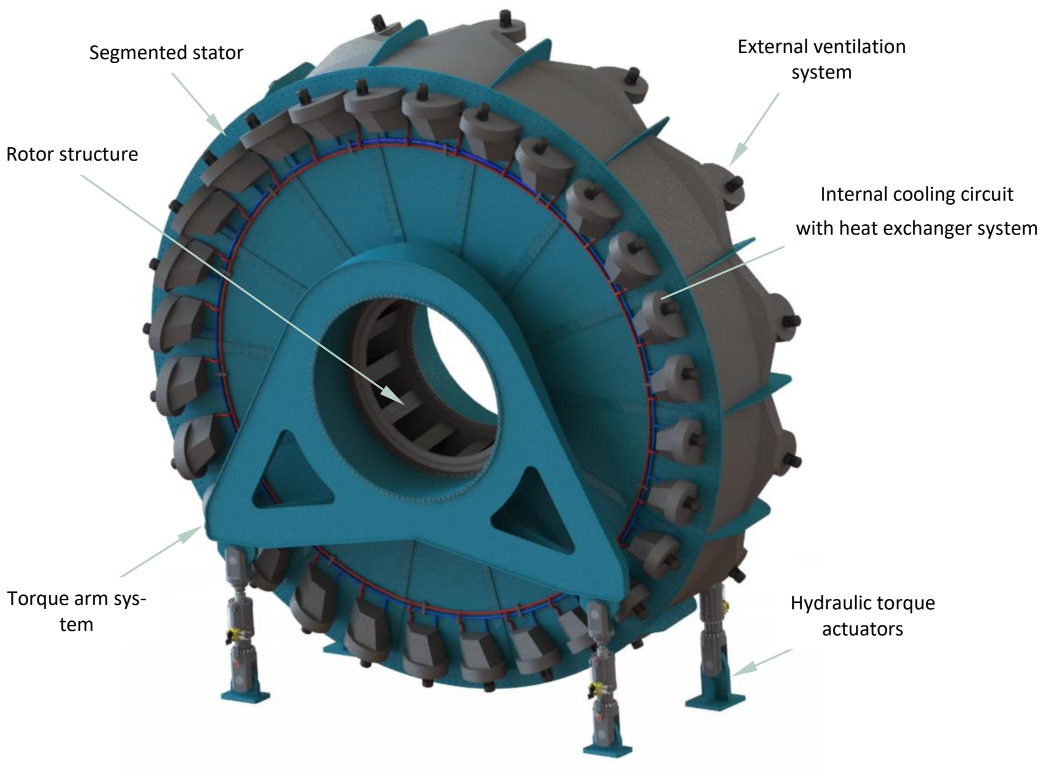

Figure 1-4 DD motor concept

The DD motor has an inside rotor like the MDS, which has the advantage of a direct symmetric connection to the TLU main shaft.

DD demonstrator requirement specification

The requirement specification of the DD motor demonstrator is based on the risks identified and assessed in the design FMEA performed in D1a. For each of the risks a recommended action / mitigation has been listed. The risks and the appertaining mitigations form the requirement specification of the DD motor demonstrator:

Failure mode	**Causes of failure**	**Effects of failure on syst. / part / operation**	**Recommended action / mitigation**	**Demonstrator requirement**
Loss of bonding strength	Wrong bond method	Loose magnets over time	Prototype	Testing of bonding strength
Cracking of resin	Wrong material Ageing Excessive temperatures Wrong tolerances	Coil vibration Debris Noise	Prototype testing of bonding tensile strength of resin at different thicknesses	Test of resin bonding strength
Cracking of resin	Thermal expansion differences between coil and lamination	Coil vibration Debris Noise	Prototype testing of bonding shearing strength of resin at different thicknesses	Test of resin shearing strength before and after thermal loading
Excessive clearance between coil and laminate	Tooth and coil tolerance mismatch	Coils become loose during operation	Prototype to validate that impregnation will secure coils, and that coils and laminate can be assembled with given tolerances.	Validation/measurement of clearance between coil and laminate

Dislocation of coils	Wrong tolerancing Wrong consideration of thermal expansion Insufficient retention from impregnation Shrinkage Ageing and oxidation	Insulation damage Coil damage Coil fatigue	Prototype	Validation/measurement of coil location before and after thermal loading
Resin cracking	Different thermal expansion between bonded materials	Insulation damage Coil damage Rotor touching and abrasions	Prototype heat testing	Verification/test of resin strength after thermal loading
Tooth yielding	Wrong tolerancing Excessive forces Insufficient material strength Wrong structural calculations Wrong material properties	Damage to insulation system Coil mechanical failure	Prototype	Test of tooth yielding strength
Insufficient cooling	CFD analysis not consistent with reality	DDM overheating	Prototype used to validate CFD parameters - mesh density etc.	Validation/test of cooling capacity during thermal loading

Table 1-3: Results from the Design FMEA and demonstrator requirements for the DD system

5.2.2 D1b: MDS - Demonstrator concept description and load specification

Through D1b the MDS concept is refined with regards to both mechanical, electrical, and hydraulic design, as well as system architecture. An initial major finding was that it would be beneficial to use a larger number of smaller (355 kW) Induction Machines (IM) for driving the ring gear since the direct-drive motors constitute a large economic risk. This eases the loads on each gear interface significantly and the cost of this even with a middle-stage gear have been estimated to give reasonable sizes on the full-scale gear and pinions, which furthermore drives the cost down.

This introduces new design challenges in making the entire mechanical structure supporting all these motors and gears. With the addition of many-fold more motors the overall gear is now has widened and changed diameter significantly. This means that we need around 74 motors which are mounted on a total of 3 gear rings. These gears are coupled by stiff flange connections to a common shaft and the entire assembly is carried and actuated by two bearings mounted on a rig that is actuated hydraulically. Based on experience, large deformations of the main shaft will occur and therefore, each motor module (13 motors) will be suspended on pairs of actuators. This evens out the load better and also ensures that all modules can follow the shaft independently, at the cost of additional I/O's. Concept is visualized in Figure 1-5.

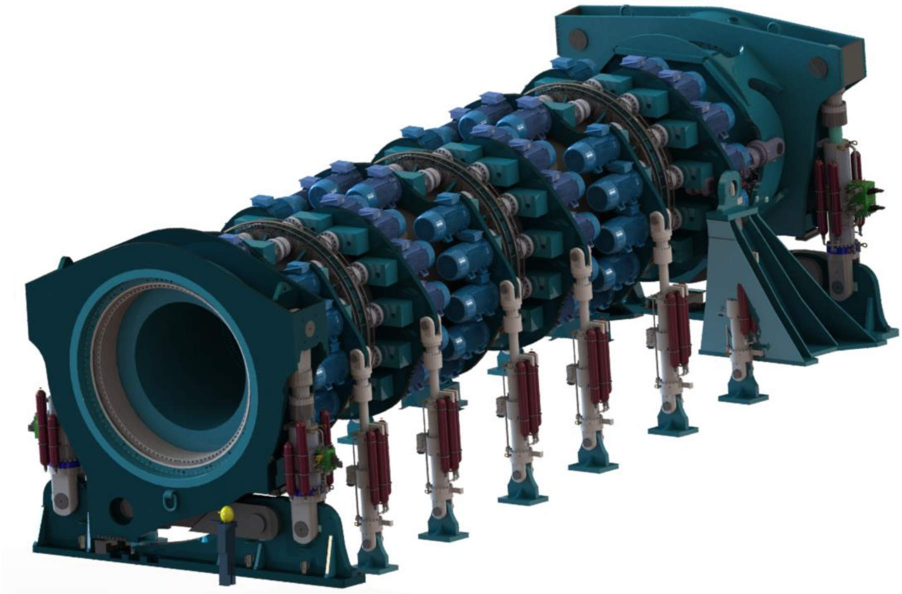

Figure 1-5: Matured 32 MNm MDS with gear concept I and HDO130 gearboxes

The Induction Machines (IM's) are mounted through an $i = 13.6$ gearbox. This has its own lubrication system and the shaft going out of this is connected via a coupling to the pinion which has an inbuild shaft with spline. The various reaction forces are distributed through the two tapered roller bearings which are over-dimensioned purposely. This makes it possible to install the entire pinion assembly with bearings from only one side of the lubrication housing. Each driveline is represented utilizing an orthogonal gear box as seen in Figure 1-6. In Figure 1-5 a straight gearbox is used elongating the main shaft but reducing the diameter.

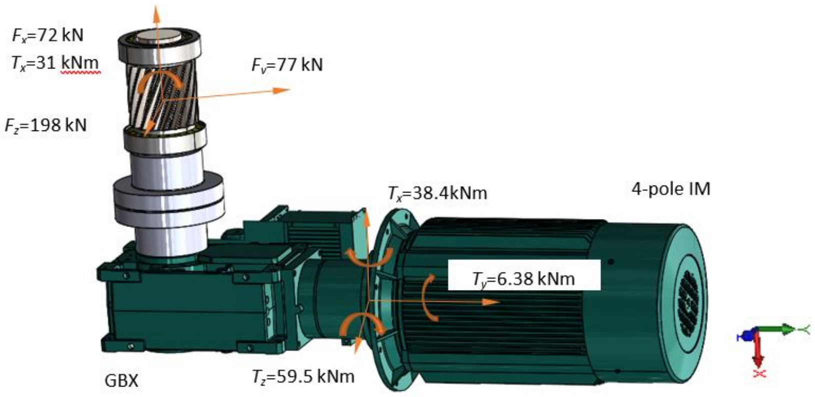

Figure 1-6: Example of an IM with HDO120 gear, coupling, bearings and pinion

Each module is carried on two ball bearings to transfer any misalignment between CTA movement and FLU/RLU movement. The entire shaft is moving in all 6 DoFs and the worst case deceleration situation will cause around 4G on the motor, which results in reaction forces and torques to be dissipated to the GBX and the motor/GBX interface shown in Figure 1-7.

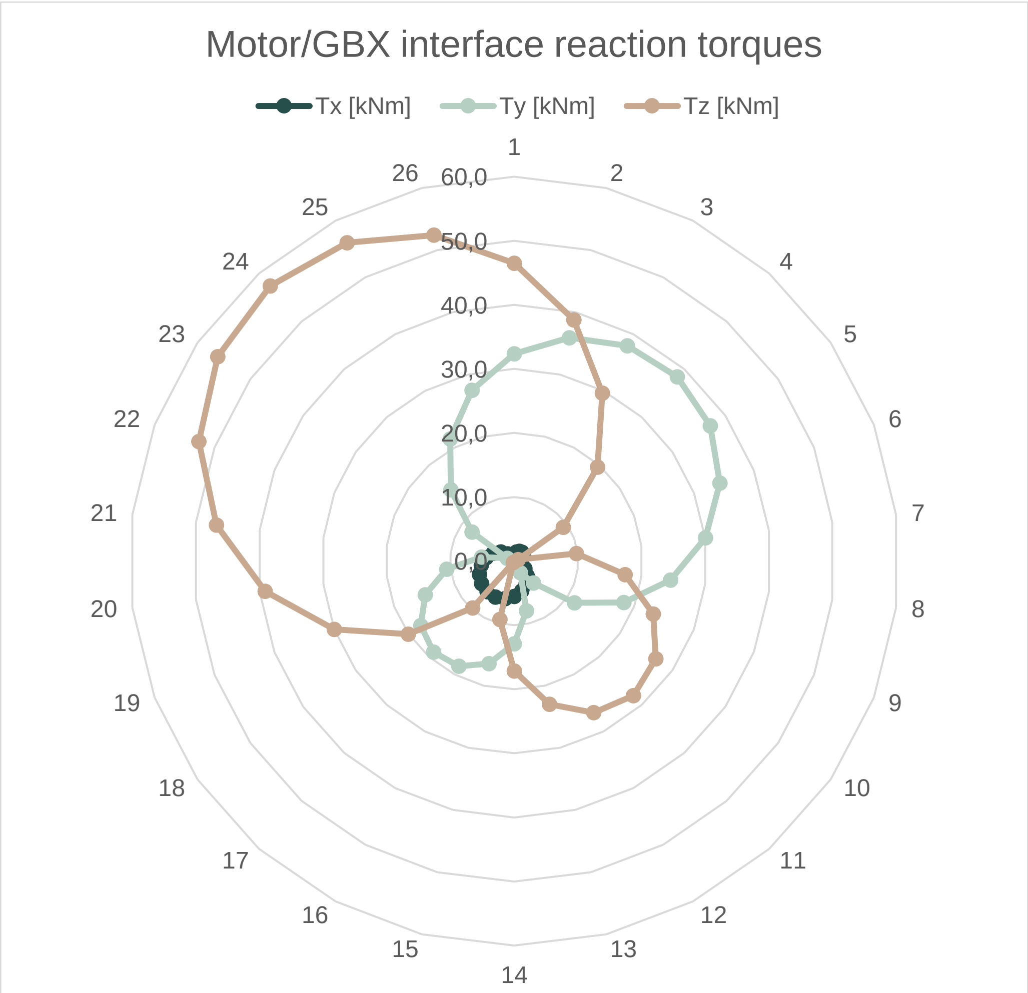

Figure 1-7 Torque loads for each motor in one module. All motors have their own reference coordinates.

Currently the wider and shorter design utilizing an orthogonal gearbox HDO120 shown in Figure 1-6 is preferable if the loads acting on each drive line with motor and GBX can be supported.

5.2.2.1 Motor and gear performance

The Squirrel Cage Induction Machine is known for its low cost and relatively high efficiency. The performance is debatable, especially at speeds outside the nominal speed. Too low speeds means that the rotor MMF wave is leading the flux density wave hence lowering the achievable torque. This should however not be an issue above 5-10% of rated speed. The efficiency at high speed could drop.

- Expected speed range for continuous operation is pm 5 % of 50 Hz
- Over torque at short circuit of the stator will at maximum be between 2.3-2.5 of the rated torque.

The period at which the motor can maintain in the max speed range is unclear. Issues could be heat or magnetic field weakening due to the fast rotation. According to data available for the HDO120 the torque-speed curve for the MDS is assumed to be as shown in Figure 1-8.

Figure 1-8: Torque-speed curve with a slight drop in output power as the speed crosses 80 Hz where field weakening will approximately start to occur.

5.2.2.2 General system analysis

A faulty rotor will cause a backward current flowing in the rotor. This current will have frequency of (1 ± 2s/f Hz where s = slip. This is a possible feature to use for diagnosis of the system. The noise emitted from a 355 kW ABB machine is normally around 83dB at 1 meter away.

HDO120 datasheets suggests max speed limit to be 5000 rpm. For the HDO120 the lifetime is 28000 h of bearings.

Water cooling of IM is quite expensive and does not make sense. If we wish to run the motor at low speeds and high torque we must install additional cooling fans. The gear can be cooled by either water or air, air will be to the room and most likely a higher surface area can be achieved. Lubrication of the Bonfiglioli gear must be ensured since the position with respect to gravity will influence the way bearing of the gear is lubricated. Alternatively, the entire lubrication and cooling system can be interconnected instead of having a sub-system for each gear.

Running the setup at increased speeds will increase safety factors of the GBX, the bearings could be questionable but is expected to have eigenfrequencies a lot faster than this. The worst part of fast rotation will be electrical and whether or not the machine can deliver the total power at the expected efficiency. If not the PE must be over-dimensioned which could become expensive.

5.2.2.3 Coupling design

Shaft diameter is ø160 at the coupling side and ø200 for the pinion (pinion size must fit with inner diameter of tapered bearing). Power is 355 kW and speed is 105 RPM, which is why torque is 32.3 kNm. Shaft is driven by an induction machine that has a middle stage gear, the shaft drives a pinion for a large gear, the diameter of the coupling can be allowed up to ø800, but both length and diameter is desired as low as possible. The required torque shock loads are found from DIN740 to be T_kmax=60 kNm.

5.2.2.4 Shaft design

The preliminary design is dimensioned using MITCalc (MITCalc, 2020) and standards. This does so far not consider the bending occurring from the RLU.

The overall diameter and thickness of the shaft requires knowledge about the torsional load and bending load on the main shaft. The current main diameter is ø2800 with thickness of 200mm.

At the highest load interface bolts of ø80 are implemented, the next load interface needs ø64 and the last interface only requires ø48 bolts where 120 bolts are used in total.

5.2.2.5 Electrical supply design

In order to get power from the external grid into each converter and thereby controlled to each drive a few steps are required for both safety and overall functionality. Step-down transformer must be installed to go from 33kV to ~500-690V. Alternatively a secondary step-down transformer must be included. A switch gear for high voltage is installed to control the power flow of the overall systems like DUT and converter stations. For each converter we must also consider if we require input filters and line choke in case of high harmonic components that must be reduced. In LORC XL they have installed a DC-chopper in case of grid failure, this will cause the back-emf of the rotor (since PM are used) to potentially blow-up the DC-link capacitor. An electrical setup similar to the one shown in Figure 1-9 and Figure 1-10 is expected for the MDS setup.

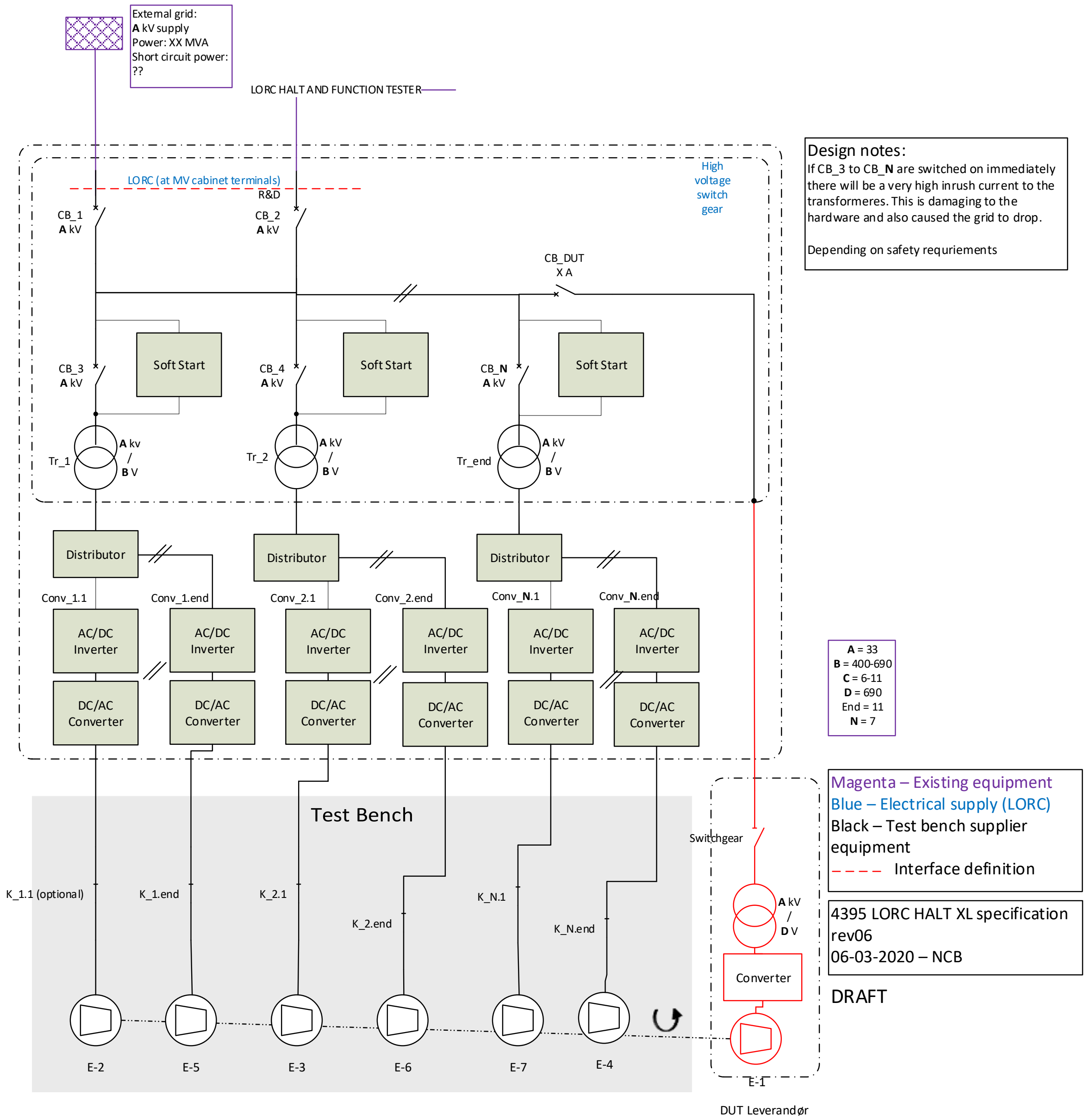

Figure 1-9: Electrical setup of the HALT XL and assumed to be similar to what is required for the MDS

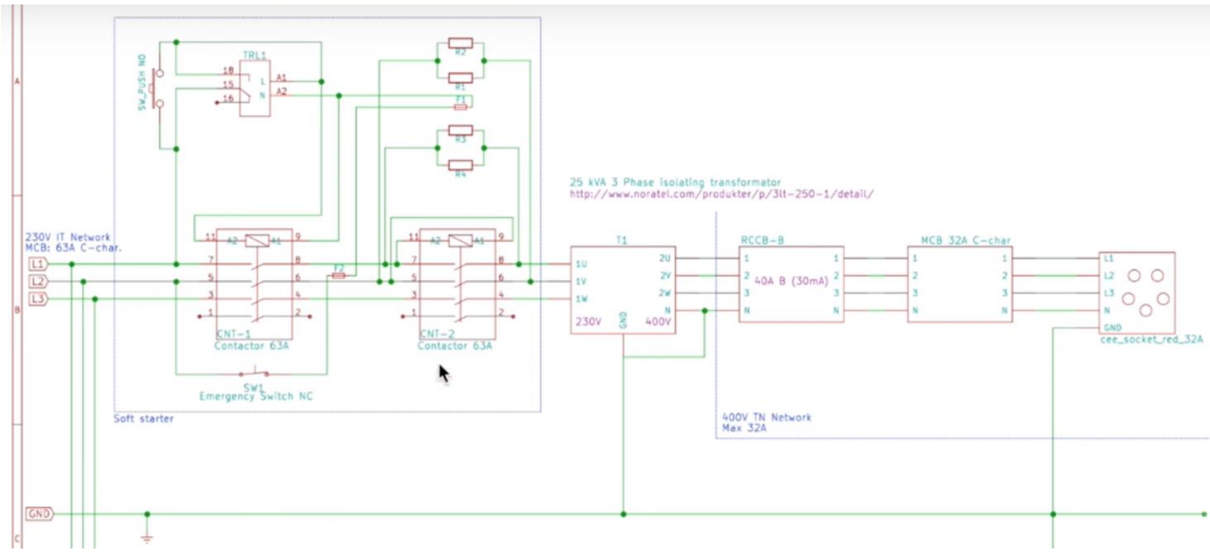

Figure 1-10: Electrical setup of the HALT XL and assumed to be similar to what is required for the MDS

5.2.2.6 Cooling considerations

The losses in converters (water cooled), induction motor (air cooled) and gear box (either water or air cooled, both through the lubrication) will be significant. The HPU is as dimensioned in the HALT XL. Bearings and pinions are cooled by lubrication and a ΔT = 15 °C. Approximately 1.1 MW in the IM must be distributed out of the test room and this will be done via a piping ventilation system. The cooling fan of each IM is reversed so that it is generating a suction force. This mass flow must be re-feeded into the room. The DD will lose around 1.1 MW and out of this 0.5 MW is to the building from the rotor.

5.2.2.7 Sealing considerations

Normally seals go to ø1651 mm and the surface speed up to ~16 m/s. We think a customized o-ring and square ring will be enough at the specified pressures and viscosities. A few leakage drops is tolerable. A manufacturer of such large rubber parts have not been located yet, Eaton and Parker does supply rotating seals but these are limited and must be installed directly around the shaft on each side of the bearings.

5.2.2.8 Failure and risk assessment of the design

Several components can be source of a failure and even though an FMEA have been done it makes sense to give an overview of the various components that may fail with somewhat stochastic failure rates.

5.2.2.9 Concept evaluation

The 32 MW concept visualized in Figure 1-5 is quite long, since it needs to accommodate 3 main gear rings with 74 IM units mounted. This takes up a lot of space and makes the drivetrain significantly longer than the DD drivetrain concept. Therefore, the concept is expected to be most suitable for solutions within 10-12,5 MN range since only 1 main gear ring is required. This greatly simplifies the system architecture and reduces the risk related to introducing additional gear rings which needs to be synchronized in order to reduce wear and vibrations and to achieve a stable torque output.

5.2.3 D1c: MDS - Specification of gear design

This report contains the specification for material, geometry and power loss of the ring gear and pinion combination.

The specification is based on a single pinion/ring-gear connection entered into a MitCalc calculation sheet. The design values are based on work done on the full-scale concept and inputs regarding material quality and manufacturing processes from the company Niebuhr who has extensive experience in manufacturing large ring gears for the wind turbine industry.

5.2.3.1 Material choice

- Material of the pinion: 18CrNioMo7-6
- Material of the gear:42CrMo4
- Accuracy grade: ISO1328 8

The material choices for pinion and gear are selected based on input from Niebuhr Gears. The chosen material types are common materials for these types of application having a sufficient material strength while being relative economically accessible. Material types with higher strength values are available but with a large premium on the cost level.

5.2.3.2 Lubrication

- Oil type: Mineral
- Viscosity: 395 cSt

The calculations on a mineral oil with 395 cSt viscosity, which is a common oil type for this application. Selection of additives and further specifications will be decided based on results from WP2 work.

5.2.3.3 Dimensions and Manufacturing

- Pinion diameter: 289,46mm
- Ring gear diameter: 4375,90mm

The dimensions of ring gear and pinion has been chosen based on input from Niebuhr regarding material quality and manufacturing capabilities. The choice of diameters is within the limits of what is relatively common with regards to manufacturing and therefore also relatively chapter compared to diameters largen than 5m.

5.2.3.4 Summary

Based on the safety factors listed in the calculations of chapter 2, along with the considerations listed above regarding material choice, dimensions and manufacturing capabilities the design is assessed to be valid for its application in a MDS drive system. The choices made also consider the cost of the gear components as this is a critical part of designing a competitive MDS driveline solution.

5.2.4 D1d: MDS - Specification of lubrication system

The design values in this section are based on work done on the full-scale concept shown in Figure 1-11 with 3 ring gears connected to a total of 77 pinions.

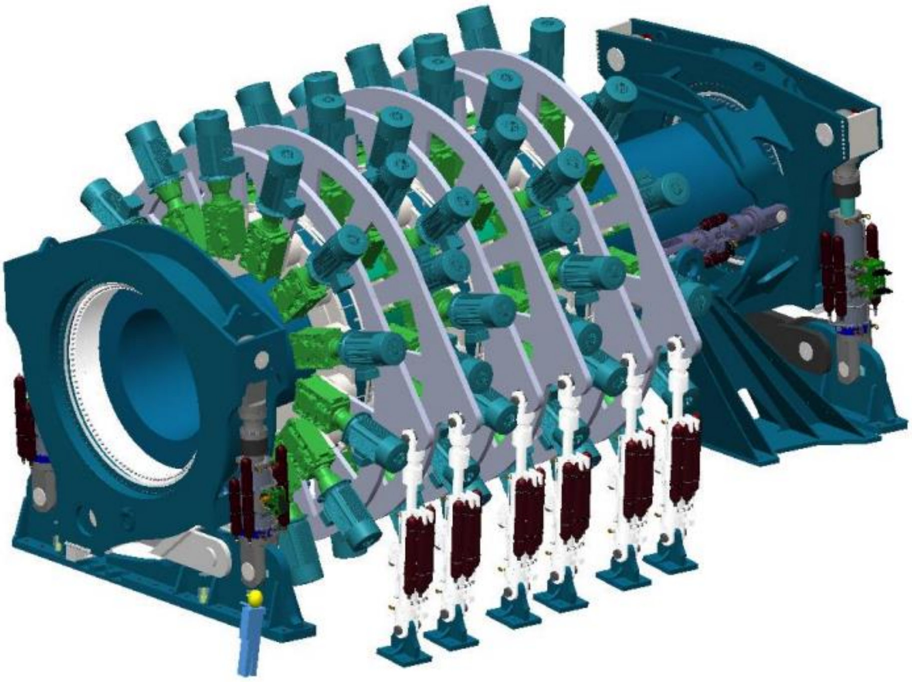

Figure 1-11: MDS design with three ring gears and 77 pinions

The requirement for each set of SKF bearing located at each pinion is 14 L/min and each pinion must get between 18-23 L/min to remove the heat. A tank, pump, tubing, filter, and heat exchanger system is required for each motor module. Delta T is 15 C for calculations. The total desired flow per pinion is 32-37 L/min including 2 SKF bearings. The accuracy of this flow is not critical, and no sensors are needed, except fluid temperature measurements. One main pump will be used for each ring gear. This means it should supply 800 L/min. Three separate lubrication stations should be used with a total of three 2000 L tanks. The pressure must be around 4-5 bar.

5.2.4.1 Line Diagram

The line diagram shown in Figure 1-12 shows the lubrication requirements for the different components in the full-scale MDS concept.

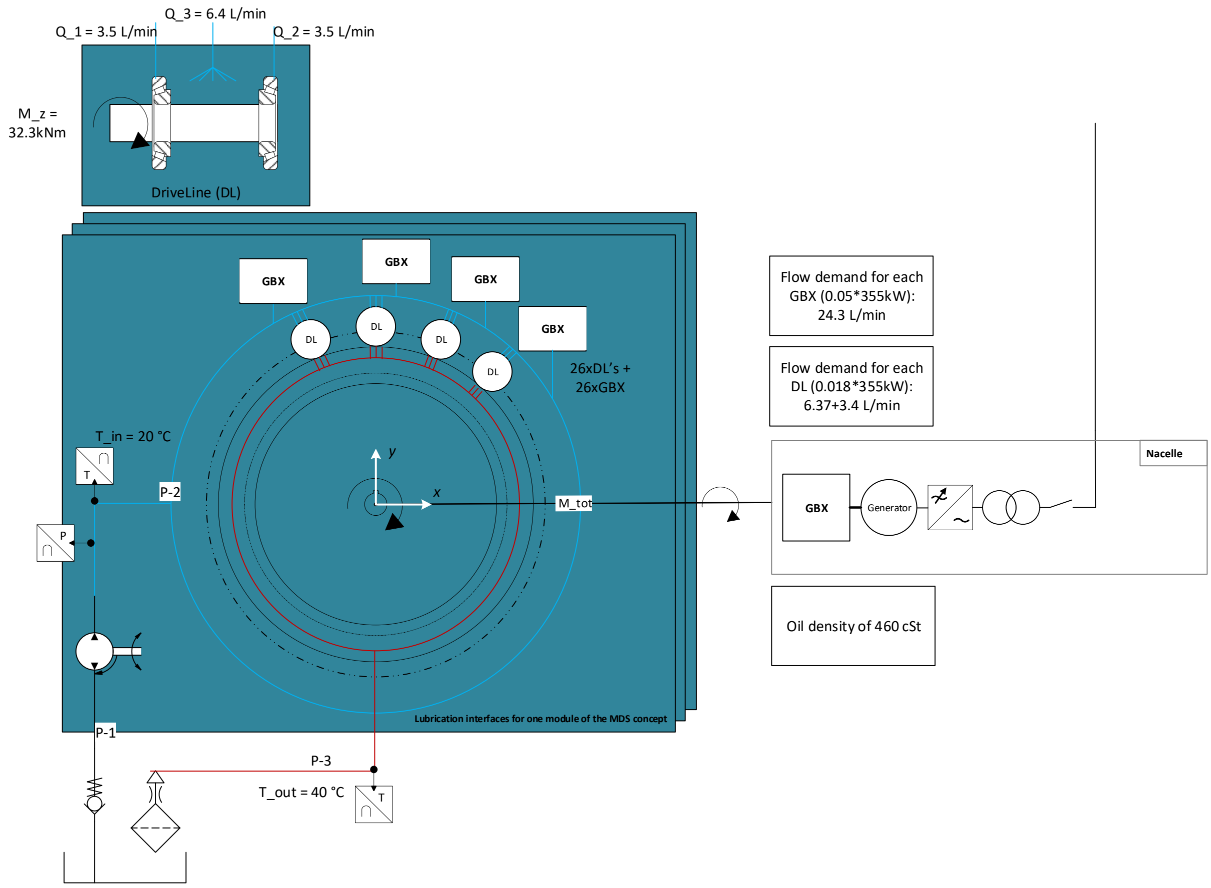

Figure 1-12: Line diagram for MDS lubrication setup

5.2.4.2 Component design

The lubrication concept is based on the following design specifications:

- Direct lubrication of the pinions
- Separate lubrication supply for the pinion bearings
- Main ball bearings for each lubrication housing
- Plan to reuse lubrication from other components
- Coupling for pinion-GBX and GBX-motor interfaces (two places)

Design notes:

- The primary function of the lubricant is to cool down gear and bearings
- Relatively large quantities of lubricant is needed for GBX
- Either flow or pressure should be monitored in order to ensure proper cooling
- Temp measurement can also be used to control the pump
- One separate lubrication system for each module with around 800 L/min is required

For further details please refer to deliverable D1d.

5.2.4.3 Cooling requirements

Achieving proper lubrication to avoid overheating/wear is critical to the functionality and longevity of the MDS concept. In Table 1-4 the calculated heat losses are listed which should be dissipated though the gearboxes, pinion to gear contact and bearing losses. As seen in the table, a total of almost 1.7 MW is required in cooling capacity for the full-scale system, when testing a 16MW DUT during HALT testing.

Parameter	**Large DUT – 16MW (HALT testing)**
GBX losses (~89.4%)	1.14MW
Expected pinion to gear loss (~98.68%)	330kW
Expected SKF + MLU bearing losses (~99.3%)	112kW
Expected TLU bearing losses (~99.3%)	112kW
Sum	**1.694MW**

Table 1-4: Table with calculated heat losses from MDS concept.

5.2.5 D1e + D1f: DD - Demonstrator concept description and Specification of cooling system tests

This section concerns the concept work of the full-scale DD concept with the purpose of providing the best possible basis for designing the DD demonstrator. The main purpose of the DD demonstrator is to demonstrate the technology while mitigating the major technological risks identified in the full-scale concept, to the degree possible within the boundaries of the project.

The section contains a description of the DD full-scale concept, the demonstrator concept and test specifications for both cooling flow tests and electrical tests.

The prime mover system is an ‘inside-rotor’ DDM, which has the advantage of a direct symmetric connection to the TLU main shaft. The rotor and TLU main shaft are connected on two cylindrical flange interfaces almost symmetrically distributed about both DDM COG and magnet/coil positions. This means that both gravitational/inertia loads as well as torque will be equally transferred from the rotor to main shaft, significantly increasing stability in the large structure.

Two bearings positioned almost symmetrically to the DDM centerline connects the rotor to the stator structure. The bearings serve to maintain the airgap between the rotor magnets and stator coils and allows relative rotation between the rotating rotor structure and the stationary stator structure.

The cooling unit ventilates the magnets, coils and external ribs and transfers part of the heat to a water cooling system through water/air heat exchangers. The remaining heat will be delivered to the surroundings.

The winding system of the DDM is designed according to an IEC temperature class F rating and approved for converter operation. Imbedded temperature sensors will give precise temperature surveillance.

For further details please refer to deliverable D1e and D1f.

5.2.5.1 Demonstrator design

The demonstrator concept consists of making a replica of the machine cooling paths’ dimensions on 1/5 of a full-scale prototype segment. as shown in Figure 1-13. This involves the airgap, which needs to be in the right dimensions. Further it is possible to block flow paths partly to replicate a full-scale concept.

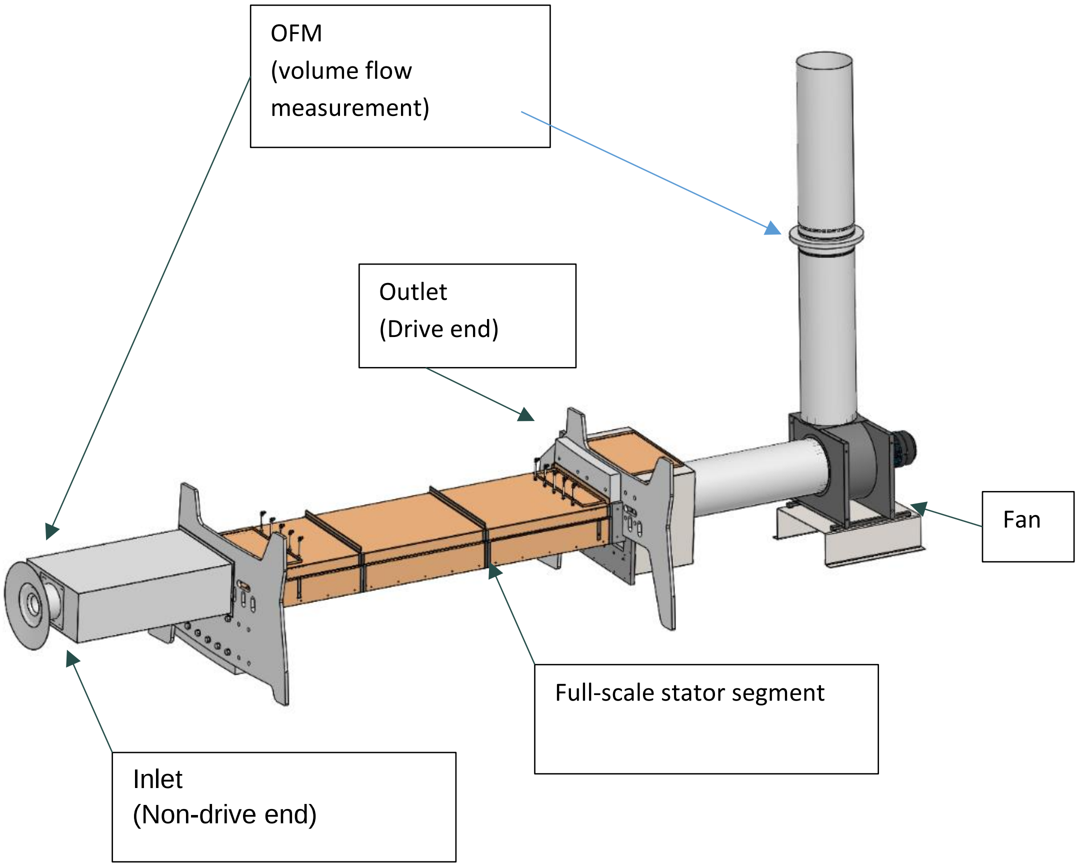

Figure 1-13: Mockup 1 for pressure loss test in airgap, water circuit and for heating tests

5.2.5.2 Specification of cooling system tests

Cooling flow tests	
Test name	**Description**
Volume flow adjustment test	The volume flow adjustment test is constructed to obtain the correct settings for the two frequency converters that results in the desired flows.
Segment heat run	The heat tests of the prototype segment is made to validate the DDM CFD model further than by the pressure losses. The heat is added to introduce parameters which is only active during heating of the segment. Further it should be validated where the hot spot temperature occurs on the segment. This information will be used to position sensors that are used create alarms and I worst case shutdown operation.
Electrical segment tests	
Test name	**Description**
Winding DC resistance	The winding resistance shall be measured for all phase of the segment. This should be done by applying 10 A between the phases' busbars (U-V, U-W, V-W) and calculating the resistance from the applied voltage. For all measurements the ambient temperature and room humidity should be noted.
Test of sensors	All temperature sensors embedded into the segment shall be tested for their functionality and insulation resistance. These tests shall establish certainty for that no sensors have been damaged during the assembly
Insulation system tests	The overall insulation resistance shall be measured for the segment. The insulation resistance shall be found using DC to find immediate continuity between winding system and frame (ground).

High voltage tests	The high voltage test consists of test done with higher voltages than nominal values. These can both be AC and DC depending on the test.
Surge test	The surge test primary goal is to detect abnormalities in the machine windings such as inter turns, Coil-to-coil weakness, Phase-to-Phase weaknesses, wrong turn count, wrong coil or ground connection etc. which could create e.g. asymmetries.

Table 1-5: Specification of cooling system tests.

5.3 WP2: Material testing and data validation

5.3.1 D2a: Specification for gear material and treatment

Based on the deliverable D2a 18CrNiMo7-6 according to EN10084 has been chosen based on the Specification for gear material and treatment listed in Table 1-6.

#	Specification for gear material and treatment
1	**Case hardening depth (CHD):** adequate CHD is necessary to achieve the required fatigue strength at the case and core; the gear material has to be suitable for long heat treatment process times to achieve the high CHD required for large gears;
2	**Surface hardness:** a minimum surface hardness of 660 HV or 58 HRC (Rockwell-C hardness) is required according to existing standards in order to achieve allowable stress numbers for pitting and bending of quality levels MQ and ME; higher surface hardness values do not increase fatigue resistance, but make machinability more difficult; in contrast, wear resistance of the surface typically increases with increased surface hardness;
3	**Core tensile strength and toughness:** increased core hardness is known to especially influence the tooth root bending strength; higher core toughness allows higher core hardness for optimized strength; Furthermore, increased core strength and toughness are assumed to reduce the risk of tooth flank fracture damages; gear steels with improved hardenability are required to achieve the desired properties for large gears;
4	**Microstructure and grain size:** fine acicular martensite in the case, as well as fine acicular martensite and bainite in the core are required for optimized load carrying capacity; fine grain size, particularly ASTM 8 and finer, is known to positively impact gear flank and tooth root load carrying capacity; adequate alloying elements are required to ensure grain size stability and fine microstructure even at long process times of arburization;
5	**Residual austenite:** a certain amount of retained austenite in the case is, due to its ductility, assumed to be beneficial for micropitting load capacity and may also contribute to an improved pitting strength; a higher amount of residual austenite may reduce case hardness and bending strength; up to 25% finely-dispersed retained austenite is allowable according to existing gear standards;
6	**Cleanness:** non-metallic inclusions are known to act as local stress raisers; depending on inclusion size and its chemical composition, the gear load carrying capacity, especially the risk of a crack initiation below the surface, may be diminished; as the highly stressed material volume increases with the gear size, the probability of critical inclusions located in critical material sections is increased; consequently, high demands on the cleanness of the gear material especially for large gears result;
7	**Area reduction ratio, material homogeneity and intergranular oxidation depth:** these are further parameters that gain special importance for large gears; requirements according to existing gear standards have to be fulfilled even for larger gear sizes; intergranular oxidation can act as a fatigue fracture initiation site and may reduce the fatigue strength of the tooth;
8	**Hardenability:** improved hardenability of the gear material is a basic requirement to achieve several of the above described properties for large gears.

Table 1-6: Specification for gear material and treatment

5.3.2 D2b: Report on data validation findings (DTU)

In this section the materials used for the main gear on the MDS concept is further investigated. Subsequently the materials are tested on a twin disc rolling contact fatigue test rig at DTU.

5.3.2.1 Material choice

During the work in WP1 where the transmission concept was developed, it was decided that the MDS concept should be based on traditional involute pinion-and-gear transmissions.

The relevant and most promising method for testing materials for very large involute gears is “Rolling contact fatigue test (RCF)”. This test mimics the combined rolling and sliding condition under normal operating conditions for an involute gear transmission.

As described in “Annex A – Detailed description of project content”, section 2.2, the two alternative test setup “Pin-on-disc” and “Block-on-ring” are mainly used to characterize the wear situation under conditions were oil supply is insufficient or the sliding speed between heavily loaded components is very low.

Since the work in WP1 revealed that a traditional involute gear transmission should be used in the MDS the RCF-testing were selected for the experimental work in WP2.

The conceptual analysis in WP1 revealed that the physical/geometric dimensions of pinion and gear were very large. The teeth on pinion and gear need to be hardened, and due to the dimensions only flame hardening and induction hardening are possible. After discussions with the company Niebuhr Gears it was decided that the material 42CrMo4 was the most promising candidate.

5.3.2.2 Test description

The main components of the twin disc rolling contact fatigue test rig shown in Figure 1-14 are two drive shafts and a hydraulic loading arrangement. One test disc is mounted on each drive shaft, respectively. The two disc surfaces are brought into contact against each other by applying a normal load, using the hydraulic loading arrangement.

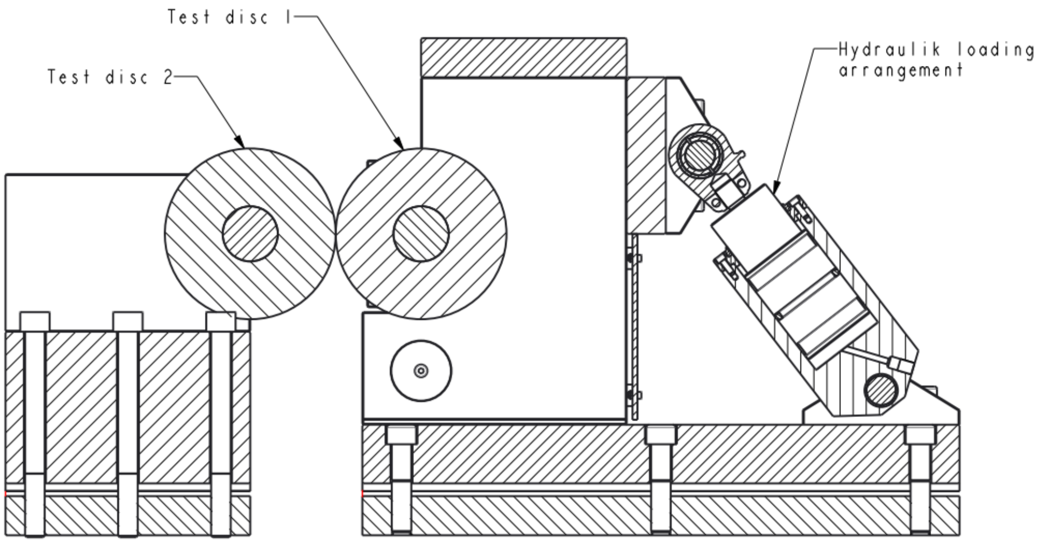

Figure 1-14: Illustration of the twin disc rolling contact fatigue test rig at DTU.

Each test disc has a conical bore, fitted onto a corresponding cone on the shaft. Each drive shaft is capable of rotating at different speeds (1200 - 3000rpm). There is a provision for relative slip between the two rings when the two shafts rotate at different speeds. Hence it is possible to create conditions of rolling and sliding. Drive shaft 1 is connected to the master motor and drive shaft 2 is connected to the slave motor. The slave motor acts as a power generator in this setup since it receives input power from master motor through the contact between the discs.

In the twin disc rolling contact fatigue test rig, the gear material 42CrMo4 was tested in an induction hardened condition. The contact between the steel discs was lubricated with Mobil Delvac Synthetic Gear Oil 75W-140.

The objective was to study the fatigue strength of the material and to study the failure development under conditions with very high contact pressure.

5.3.2.3 Test results

An unexpected outcome of the first test run was that the shaft, carrying one of the test discs, failed. The failure was clearly a fatigue failure in the shaft material. The failure on the shaft occurred at a location along the shaft where the bending moment is close to its maximum value and at the same time with a high stress concentration due to change in shaft diameter.

It was the intention to run the tests with specific slip percentage between the discs, but as the contact pressure between the discs was increased, it became difficult to maintain the slippage. This was because the oil in the contact was able to transmit an unexpected high shear stress.

To initiate surface failure on the contacting discs, the oil supply was closed in time intervals of varying length in 6 tests. When the oil supply was closed the temperature increased, but the oil present on the surface was sufficient to maintain surface separation. The longest time interval without oil supply was 47min.

The test discs show no visible signs of incipient failures along the contact path. The contacting surfaces are slightly bronze-coloured, but there are no signs of surface wear as shown in Figure 1-15. The grinding tracks are clearly visible on the test discs.

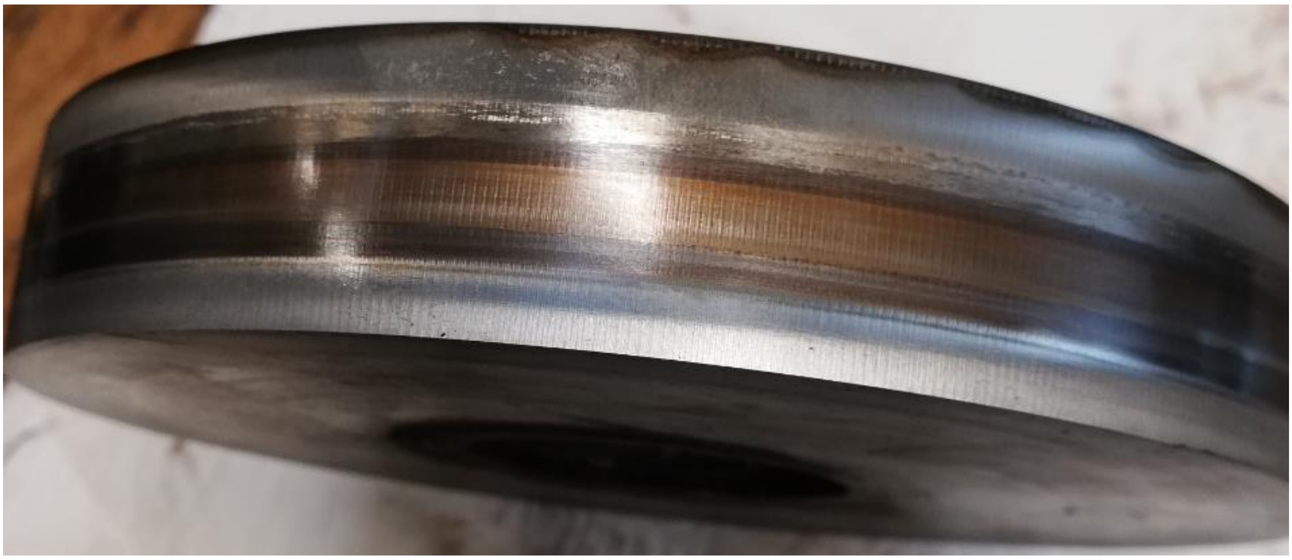

Figure 1-15: Picture of test specimen with slightly bronze-coloured contacting surfaces.

5.3.2.4 Conclusion

Based on the conducted tests it is concluded that the selected material 42CrMo4 in an induction hardened condition is very well suited for application in the MDS. Further the selected gear lubricant Mobil Delvac Synthetic Gear Oil 75W-140 is an obvious candidate for the MDS.

5.3.2.5 Potential for articles

The work performed has potentially formed the basis for several articles within the subject, which are being pursued by DTU:

- Estimation of gear lubricant limiting shear strength under Hertzian contact pressures between 1 and 3 GPa.
- Roughness development in highly loaded and lubricated gear contacts where film thickness prevents asperity contact.
- Pitting failure development under combined normal and shear load conditions in lubricated elliptical contacts.

5.4 WP3: Demonstrator design, manufacturing, and installation

5.4.1 MDS demonstrator concept

In this section the scaled MDS demonstrator is outlined. A conceptual model of the MDS demonstrator is shown in Figure 1-16 with one motor driving the ring gear and one counteracting to slow it down. The torque transmitted in the demonstrator is not significant and therefore the mounting of the gear does not need to transmit much energy.

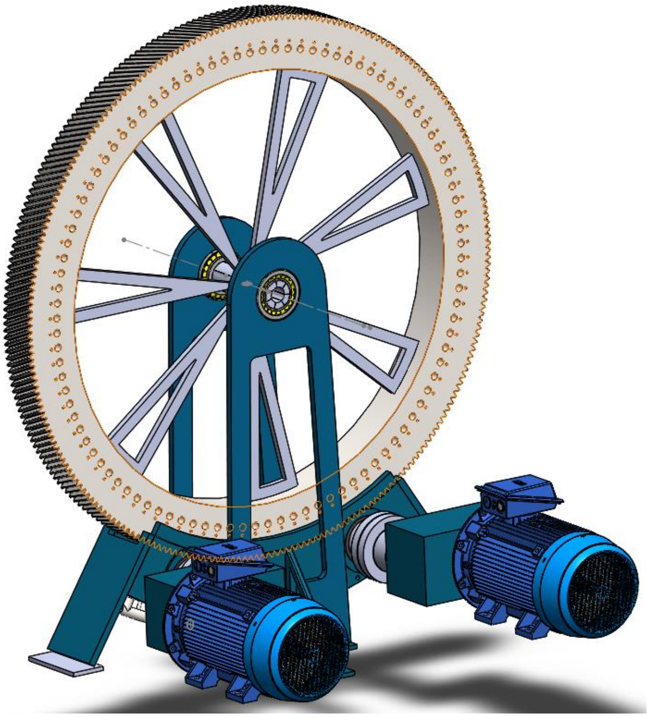

Figure 1-16: Conceptual model of the MDS demonstrator.

The following tests were planned if the demonstrator had been manufactured:

- Run-in tests
    - Purpose: Investigate initial wear on the pinion and ring gear.
- Short test of control performance
    - Purpose: Testing the controllability of torque
- Long term testing
    - Purpose: Investigate wear on the pinion and ring gear over a longer period of time.

Due to the following factors the scaled MDS demonstrator was abandoned.

- Behaviour observed during scaled tests is not ensured to be representative to full scale behaviour.
- Long term testing is very expensive and required a regenerative system to reduce cost.

- Expenses required to establish MDS demonstrator was better used on investigating the feasibility of journal bearing technology for MDS and further investigations of DD technology.

5.4.2 Journal bearing technology for MDS

It is a major concern that the high amount of pinions interfacing to the main gear will introduce significant wear due to misalignments or vibrations. Therefore, it is investigated whether journal bearing technology could be applicable for the MDS concept.

Initially the different types of journal bearings are described alongside design guidelines and equations for performing basic analytical calculations. Considerations regarding fatigue life and temperature distribution within journal bearings are covered alongside guidelines for balancing to prevent instability. Journal bearings are compared to rolling element bearings in Table 1-7.

The oil lubrication system required for hydrodynamic journal bearings is described. General design guidelines covering the following sub systems is described: Pump system, Filtration system, and Temperature control system. Bearing condition monitoring is described covering both internal bearing monitoring and general lubrication monitoring.

Current journal bearing technology has been commercially implemented on bearings sizes up to 1.2m. Development of larger bearings using multiple bearing pads is something being studied. For high-speed machinery such as turbines journal bearings are often used for their excellent dampening and self-aligning abilities. At low speeds journal bearings could be self-contained where no external lubrication systems are required. For designs where a long running time with retentively few stops are require hydrodynamic journal bearings are an excellent choice. For designs where there are low loads and lot of start/stop operations hydrostatic bearings could be an excellent choice. When using shaft diameters above 1.2m journal bearing technology is not a well proven technology. These larger bearings are currently being considered. However, not viable for commercial use without taking a major risk. Therefore journal bearings are only considered feasible for the pinion bearings.

5.4.3 Rotor dynamic investigation to compare journal bearings with ball bearings.

In the following section the effect of exchanging Journal bearing with ball bearings is investigated. This investigation is performed to obtain knowledge of when it is appropriate to use each of the two types.

The Journal bearings investigated are from Main Metall, with a shaft diameter of Ø70mm. A suitable ball bearing alternative have been found in SKF 6014 M. The bearing Stiffness and damping data for the two bearing are investigated and it is found that the stiffness in the K2 direction (main direction of the force) is almost identical. The ball bearings are however slightly more compliant. For the K1 direction the journal bearing stiffness in only half of the ball bearings. Furthermore, ball bearing does not provide any damping and therefore the bearing do not have any damping constants.

Based on the rotor dynamical analysis it can be observed that the some of the modes related to the shaft are lower for the journal bearings. The primary reason for this is most likely due to damping, as the damped natural frequency is lower than the undamped.

Figure 1-17 show the bearing loads in the bearings for a harmonic unbalance on the driveline. From the figure it is observed that the ball bearing loads have two very distinct peaks in the load. This is when the driveline speed is coincident or close to a natural frequency. The journal bearing data have a peak just below 100Hz, but the peak is much lower and wider.

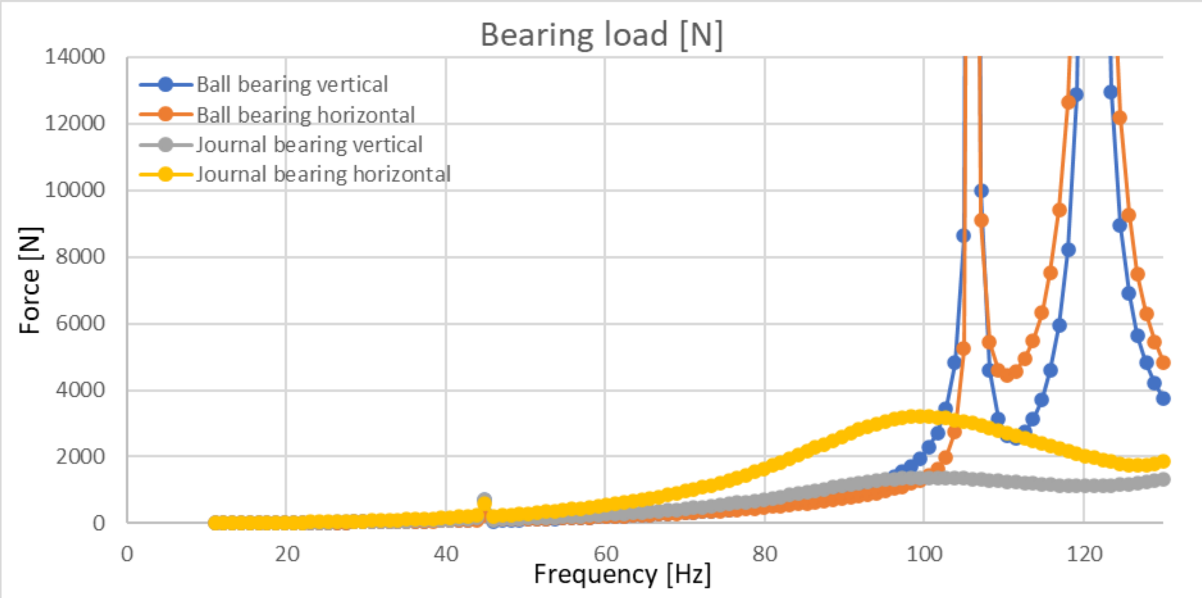

Figure 1-17: Bearing forces from a harmonic analysis of the two different bearing configurations.

From the data in Figure 1-17 it's observed that the journal bearings have significant different response due to the damping. The force peaks at the natural frequencies for the journal bearings comes more gradually which makes the system more predictable as small changes in operation speed does not alter the response significantly. While a small speed increase for the ball bearing model could increase the force 10 times. Due to the higher damping the system with journal bearings can operate at the natural frequency as the damping ensures the loads is not above the allowable. The high damping also ensures the force amplitudes increase slower which means more reaction time before failure, and the system is easier to drive fast thru speeds where natural frequencies occur. With ball bearings the system would most likely not be able to operate between 100-130Hz (6000-7800 RPM) or it should be done with caution. Both ball bearing solution and journal bearings require oil lubrication, but the consumption of the journal bearings are significant higher.

Condition	General note	Rolling element bearings	Hydrodynamic	Hydrostatic
High temperature	Verify: - Fits and clearances at room temp and high temp. - Sealant material - Heat removal from bearing housing	Up to 100 deg - No limitations From 100 to 250 deg - Possible with special bearings and special lubrication	Dependnt on the oxidation resistnce of the bearing. Correct bearing coating must be considered.	Dependent on the oxidation resistance of the bearing. Correct bearing coating must be considered.
Low temperature	Verify: - Fits and clearances at room temp and low temp. - Sealant material	Bellow -30 deg Special considearion are nessecary Starting torque might be effected	Lubricant viscosity will be higher therefore, starting torque must be considered.	Lubricant must be considered
External vibration	- Identify operating conditions and ensure that there are no large excitations - Attenditon to the possibility of fretting damage	Bearing choice must take vibrations into account. Could give limitations to the design.	Satisfactory	Excellent
Space requirements		- Many sizes available - Short axial lenght - More compact than fluidfilm bearings.	- Many sizes avalible - Short radial height compaired to axial length - Size depending on lubrication feed system	- Many sizes avalible - Short radial height compaired to axial length - Size depending on lubrication feed system
Dirt or dust		- Sealing must be used	- Sealing must be used - Proper filtration of lubricant is important	- Sealing must be used - Proper filtration of lubricant is important
Simplicity of lubrication system		- Excelent with self-contained grease - Decent with oil lubrication	- Self contained bearings can be used - Oil circulation is needed	- High pressure is necessary
Simplicity of lubrication				
Frequent start - stops		Excellent	Decent - Larger wear rate is observed when going between hydrodynamic lubrication to mixed lubrication. This is the primary driver of the fatigue life.	- Excelent, due to high supply pressure
Frequent change of rotation direction		Excellent	Good	Excellent
Operating cost		Very low	Operating cost is dictated by lubrication system	
Life		Finite and predictable	In theory infinite, effected by: - Running conditions, - Amount of start and stops , - Particles and quality of oil, - No way to calculate fatigue life	In theory infinite, effected by: - Running conditions, - Amount of start and stops , - Particles and quality of oil, - No way to calculate fatigue life
Axial load carrying			A thrust force must be provided in order to carry axial loads	
Availability of components		- Large standard range selection - Special design available	- Decent standard range avalible - Special deisng available	- Special design only
Manufacturing and assembly tolerances		- High requirements to tolerances	- High requirments to tolerances - High requirments for shaft surface finish, grinding is required	

Table 1-7: Comparison of journal bearings and rolling element bearings on several parameters.

5.4.4 DD demonstrator manufacturing

Based on the demonstrator design described in section 5.2.5.1 a 1/5 of a full-scale DD segment demonstrator was manufactured as shown in Figure 1-18. This demonstrator forms the basis for the demonstrator testing covered in section 5.5.

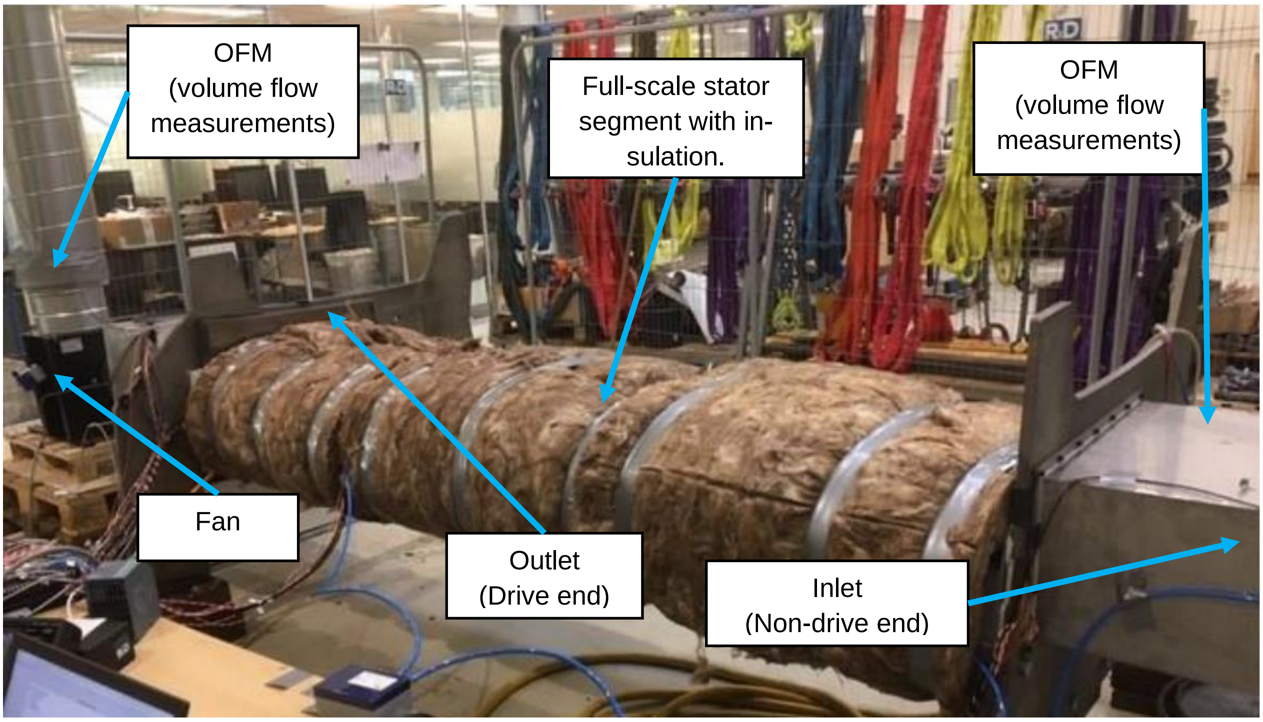

Figure 1-18: Manufactured full-scale DD segment demonstrator.

5.5 WP4: Demonstrator testing

In this section results from the demonstrator testing performed at R&D test site will be presented alongside findings from gear component analysis performed by DTU at their microstructure lab.

5.5.1 D5a: Gear component analysis

The microstructure and hardness of two 42CrMo4 cracked and failed teeth from two different yaw rings have been investigated by optical microscopy, scanning electron microscopy and micro-hardness indentation. The surface quenching and tempering treatment introduce heterogeneous microstructures with a thick, millimeter scale, low-temperature tempered hard martensite surface layer and a high-temperature tempered soft inner matrix. This is common to both teeth. Cracks are seen at the contact areas between gear pairs, and scratches can also be observed in these areas for both the cracked and failed teeth. For both teeth, 3D crack networks are seen in the samples when they are cut into pieces for characterization.

The failed gear tooth has a serious scratch-introduced fall-off. Good correspondence between the surface heat treatment traces and the hardness profile is found. The average Vickers hardness of the hard martensitic surface layer is 600 ± 7, while for the soft matrix it is 264 ± 7. The microstructure in the hard surface layer is thin plate martensite with over-tempered areas. These areas are characterized by large carbide precipitation within a ferrite matrix. There is a gradient hardness and microstructure change from the hard surface layer to the soft matrix with a width of around 810 □m. The hard surface layer thickness is around 4 mm.

For the cracked gear tooth, although its shape is intact, long cracks can be observed from the surface and the tooth broke into several pieces during cutting due to the internal 3D crack network. Good correspondence between the surface heat treatment traces with the hardness profile is also found in this tooth. The average Vickers hardness of the hard martensitic surface layer is 621 ± 14, while for the soft matrix it is 297 ± 19. The microstructure in the hard surface layer is only the thin plate martensite. There also in this tooth exists a gradient hardness and microstructure change, and its width is less than 270 m. The hard surface layer thickness is around 6 mm.

5.5.2 D4a: Demonstrator test results

Highlights from demonstrator testing performed on the full-scale DD segment demonstrator will be covered in the following sections.

5.5.2.1 Cooling flow tests

The cooling flow tests described in Table 1-5 were performed by R&D on a stator segment in the workshop. The results from the test were used to predict the cooling required to maintain stable temperatures during operation. The goal was to verify that the Class F stator segment could be operated at full power below 130°C to ensure 80.000 operating hours. The tests were also used to determine the efficiency of both water cooling and air cooling as well as the power needed to cool a full-scale DD motor as well as verifying 2D/3D thermal models of the system. The test results shown in Figure 1-19 reflect temperatures in coil hotspot and show the effect from cooling utilizing both water cooling and air cooling. Based on tests results the coil design and cooling system was verified and optimized towards the final product.

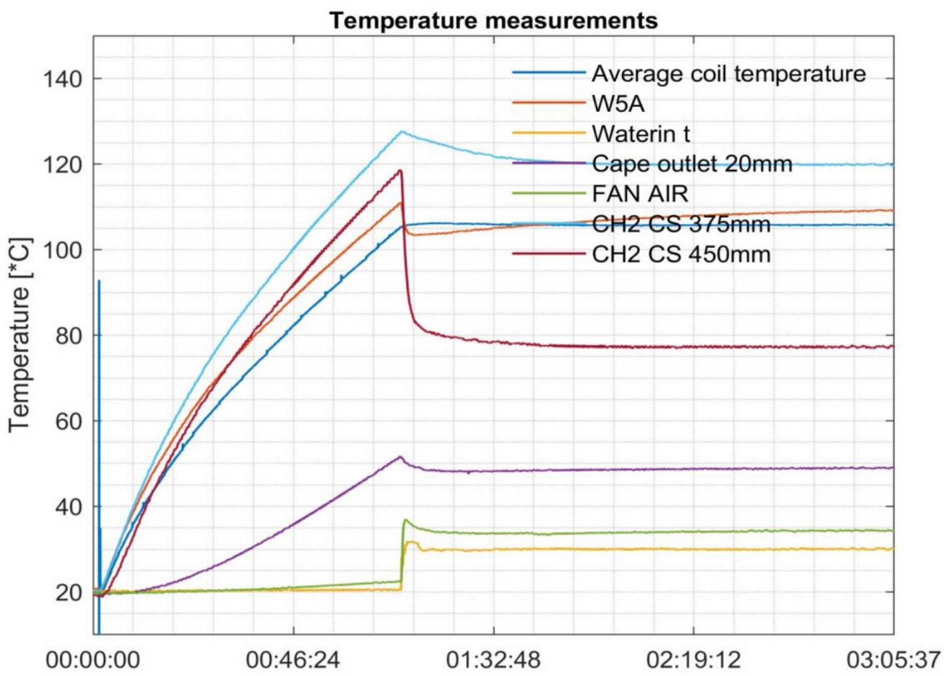

Figure 1-19:Results from test showing temperatures in coil hotspots and water/air temperatures.

Plotted values are explained below:

- CH2 CS 375 mm: Coil hotspot temperature 375 mm into coil
- CH2 CS 450 mm: Coil hotspot temperature 450 mm into coil

- W5A: coil wedge temperature measurement
- Waterin t: Water inlet temperature
- Cape Outlet 20mm: Ait outlet temperature
- FAN AIR: ait inlet temperature

Results were obtained with the following settings:

- Heater power: full power (400 A)
- Cooling water temperature: 30°C
- Cooling water pump frequency: 50 Hz
- Air blower frequency: 50Hz

5.5.2.2 Electrical segment tests

All electrical segment tests described in Table 1-5 were performed by the sub supplier and were used to quantify the DC winding resistance and the insulation resistance. High voltage tests and surge tests were also performed to ensure segment quality.

5.5.2.3 Conclusion

Both the electrical and cooling tests has confirmed that the full-scale DD coil segment design performs as intended and can be cooled during operation with a combination of air cooling and water cooling. The DD design is therefore considered feasible for full scale implementation in a DD driveline solution for a 25 MW Nacelle test bench.

5.6 WP5: Data and material analysis and evaluation

In the following section the failed and cracked gear tooth will be examined more closely to give information for future gear material microstructure design. The final evaluation of the DD and MDS concept will also be performed.

5.6.1 D5a: Component characterization

The comparison of the two failed and cracked gear teeth give information of critical importance for future gear material microstructure design for longer service life and sustainable development.

i) For the broken tooth, the lower surface hardness and the thin surface layer with the over-tempered microstructure seem to reduce the wear properties. Crack initiation and propagation originate from the soft over-tempered layer with large carbides and ferrite;

ii) For the cracked tooth, the hard and thick surface layer provides a better wear resistance, while the thin transition area resulting from the fast heating and quenching is likely to introduce a large internal stress. This internal stress may be the reason for the internal 3D crack network.

Recommendation: The above comparison indicates that a thick and hard surface layer, a reasonable wide transition layer between the hard surface layer and the soft matrix, and appropriate annealing to remove the internal residual stress may improve the gear properties and service life.

5.7 D5b: Design evaluation and overall project findings

In the following section the project as a whole and the MDS and DD driveline designs are evaluated to verify feasibility for full scale implementation and to highlight the general findings in the project.

5.7.1 Project evaluation

The original project focus was the following:

“The project focus is to make large scale nacelle test facilities technologically and economically feasible for the wind power industry, by developing and demonstrating a new modular drive technology”

The original project proposal did only include work for a MDS driveline solution. This was determined to be too narrow a scope since there were several technological challenges to the MDS concept and possible solutions required a fundamentally new concept. This led to the two following restructurings.

- Restructuring 1: In March 2019 the DD driveline technology was identified as another possible candidate to large scale drivelines. Therefore, the DD technology was included in the project in the first restructuring of the project adding a demonstration of a DD coil segment.
- Restructuring 2: In March 2021 a second restructuring was performed based on the identification of the increased potential of the DD technology and the reduced outcome of a scaled MDS demonstrator. Therefore the MDS demonstrator was replaced by a deeper analysis into the driveline design evaluating the feasibility of journal bearings on large scale drivelines. Rotor dynamical simulations were also performed to provide indications on the effects of introducing journal bearings technology in large scale drivelines.

Additional value was added to the project through the restructuring, since the primary candidate suited for large-scale drivelines in the end was determined to be the DD concept.

5.7.2 Technological results

In the following sections the main technological findings are highlighted.

DD concept

Main technological results for the DD concept have been that the design and performance of DD concept has been validated to such a degree that it is considered suitable be used for a full-scale nacelle test bench. To mitigate the technological challenges identified a full-scale DD coil segment was designed, manufactured, and tested. The test results indicate that the proposed design can be maintained at a steady state temperature during operation of 105° by means of a combination of water cooling and air cooling.

MDS Concept

Main technological results for the MDS concept have been the identification of a system architecture which could be suited for a full-scale nacelle test bench. Main concerns are related to gear teeth wear and failure. These concerns were not possible to mitigate through testing of a scaled MDS demonstrator due to expectations that the issues observed and mitigations identified on a scaled-down demonstrator cannot be applied to a full scale MDS driveline. Therefore, the issues have been investigated by means of material testing and component characterization. This area has been identified to be an area suited for further analysis. Here methods for optimizing material and component performance should be investigated to mitigate risks of wear and failure.

5.7.3 Commercial results

In the following sections the main commercial results are highlighted.

R&D test systems

The main commercial result from the project is that the verification of the full-scale DD coil segment provided the confidence required to utilize a 25MW DD driveline design for a nacelle test bench at the Lindø Offshore renewables Center (LORC). The DD design was however not developed further within the EUDP project but was realized in this separate commercially funded project with LORC. R&D won the project and both R&D and LORC has published several articles about the project with the article below concerning the final commissioning of the HALT-XL nacelle test bench shown in Figure 1-20. The HALT-XL Utilize a dual bearing load unit with the DD-motor directly integrated to apply torque and axial loads and bending moments to the Device under test. The fact that the DD motor is integrated directly onto the driveline and the load unit makes it possible to control the torque more accurate since a gear does not transfer the power and allows for higher deflections since a coupling to a stationary motor is not needed.

https://www.rd-as.com/insights/largest-most-powerful-halt-test-rig/

Figure 1-20: A picture of the 25MW HALT-XL nacelle test bench at LORC.

Additionally, the MDS technology was evaluated to be an alternative to the DD technology for applications in the 10MW range. Here R&D has received several requests from customers finding the MDS concept interesting for such test application due to a more competitive price.

For both the MDS and DD technology the focus user groups are both OEM's pursuing end of line test capabilities, test centres performing HALT testing as well as universities collaborating with the OEM's on improving large scale nacelle testing capabilities.

DTU

Main commercial results from DTU have been the submission for the following paper for the 42nd Risø International Symposium and published open access by IOP as well as an additional spin off project:

Xiaodan Zhang, Niels Leergaard Pedersen, Peder Klit, Dorte Juul Jensen. (2022). Heterogeneous microstructure and failure analysis of 42CrMo4 yaw rings. Proceedings of the 42nd Risø International Symposium on Materials Science: Microstructural variability: processing, analysis, mechanisms and properties. Risø, Roskilde.

As a 'spin-off' of the present project, Senior Researcher Xiaodan Zhang, applied for and succeeded in getting funding for the project "*LimWind: Limit white structure flaking failures in wind turbine gears and bearings via heterogeneous microstructure design mimicking nature*" (2022-2025) by Independent Research Fund Denmark. This project will focus on the fundamentals in microstructural design and as such pave the way for the future collaboration with the present team to transfer the fundamentals of microstructural design into component design and manufacturing (by R&D and other companies), which in 2 years time could be submitted as a joint application for funding from EUDP or Innovation Foundation Denmark.

5.7.4 Dissemination

Moving forward R&D will continuously be communicating the increased capabilities of the technology developed in the project. The DD technology enables top of the line test capabilities within large scale nacelle test benches, end of line test benches and gearbox test benches.

DTU will continuously present the findings from the project at symposiums on material science and through publications. Additional publications are expected to arise from the project and the spin-off project described above. These publications should then again be disseminated in journals and symposiums on material science.

6. Utilisation of project results

6.1 Future utilization of technological and commercial results

The technological results from this project has confirmed the potential of the DD technology for large scale nacelle test benches. The results from DD demonstrator provided further knowledge about the thermal and electrical performance of the DD motor designed proposed by R&D. This has provided R&D with sufficient confidence in the DD technology to have the proposed DD motor design form the basis for a 25 MW nacelle test bench. The DD design was however not developed further within the EUDP project but was realized in a separate commercially funded project with LORC. This 25WM nacelle test bench is shown in Figure 1-20 and proven to outperform geared drive solutions in the same order of magnitude. R&D has obtained unique experience with designing, installing, commissioning and maintaining the DD motor.

The realization of this EUDP project has paved the way for the project at LORC which has put R&D test systems in a leading role when it comes to designing and delivering large scale test benches. This has led to several enquiries for large scale test benches up to 30MW from test sites and drivetrain manufacturers around the world. R&D has the ambition to become the market leader within large scale test benches and expect to utilize the DD technology to lead the way.

6.1.1 Competitors

Competitors in the large scale market mainly utilize geared drives, which introduces limitations to performance since couplings capable of handling both high torque and large deflections are needed to couple the drive and the gear. Some of the main competitors are among other IDOM and RENK. Renk has delivered a 15 MW nacelle test bench to Clemson university in South Carolina utilizing geared drive motor and a so-called load disc technology to apply loads to the device under test. IDOM has delivered a 10 MW nacelle test bench to DyNaLab in Germany utilizing a geared motor and a hexapod technology to apply loads. Both test benches are shown in Figure 1-21.

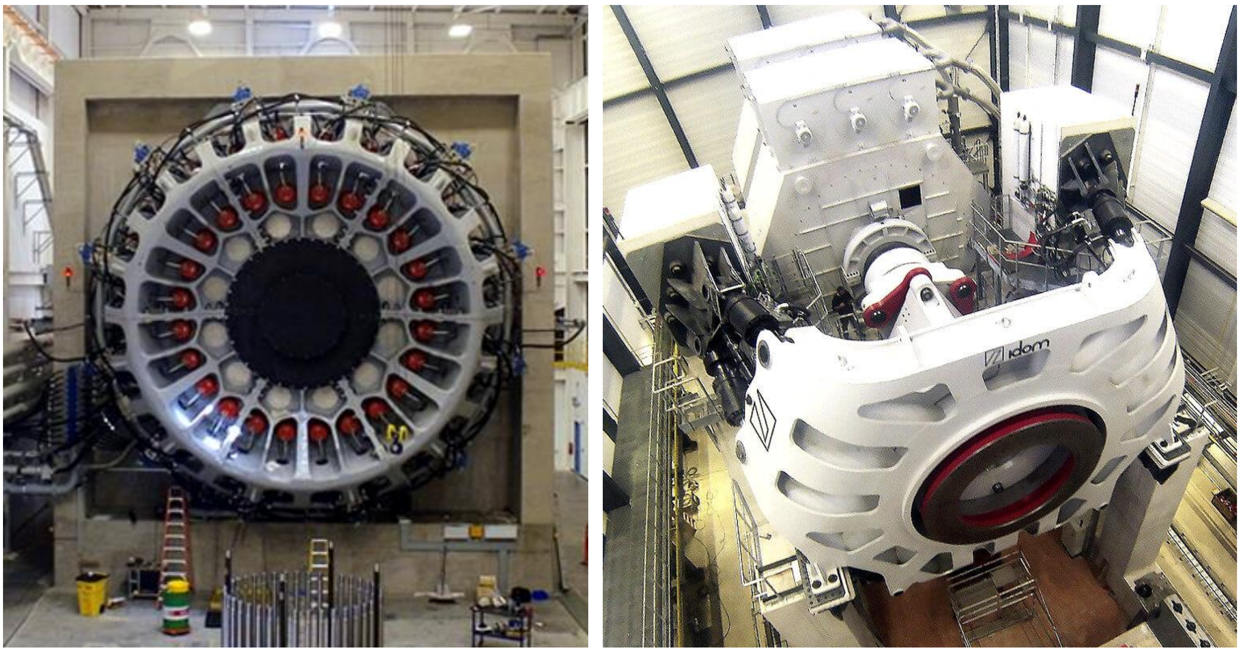

Figure 1-21: Left: 15MW nacelle test bench from RENK - Right: 10MW nacelle test bench from IDOM,

6.1.2 Sales Barriers

The two key factors relevant to overcoming the sales barriers in the test bench market for the wind industry are:

- To match the rapid increases in torques and loads needed to perform HALT (highly accelerated life test) tests on large offshore nacelles for wind turbines.
- Obtaining a track record for successful commissioning of large-scale test benches capable of testing drivetrains, bearings and full nacelles worldwide.

The HALT-XL test bench outmatch the other test benches in performance with specifications listed in Table 1-8. In addition to the HALT-XL test bench R&D has also commissioned a 16MW HALT test bench at LORC which does however not utilize the DD motor or dual bearing load unit.

HALT-XL at LORC (25 MW)			HALT at LORC (16 MW)		
Parameter	Unit	Magnitude	Parameter	Unit	Magnitude
Torque – M_x	MNm	32	Torque – M_x	MNm	15
Pitch moment – M_y	MNm	85	Pitch moment – M_y	MNm	25
Yaw moment – M_z	MNm	65	Yaw moment – M_z	MNm	25
Nacelle drive train size	MW	16-18MW	Nacelle drive train size	MW	10MW

Table 1-8: Performance specification of HALT-XL test bench at LORC.

In addition to its unique performance the DD and dual bearing technologies are highly scalable due to its modular design. The scalability alongside advantages elaborated in section 5.7.3 are the key features making the DD technology and dual bearing load unit superior to solutions with geared motors.

Combining top-level performance with a great track record of commissioned test benches will serve to reduce sales barriers within the large-scale test benches capable of testing drivetrains, bearings and full nacelles worldwide.

6.1.3 Supporting energy policy objectives

The main goal for R&D test system within the wind industry is to be capable of supporting the development of larger and larger wind turbines. Here the OEM define the phase of which testing requirements evolve over the coming years. On- and Offshore wind turbines are continuing to grow in power. Since 2015 Offshore turbines have grown at a constant rate of 16%. In 2020 the average rated capacity of turbines installed was 8.2 MW, which represents only 5% more compared to last year. But turbine orders in 2020 already show a trend towards the next generation in size, with turbines ranging from 12 to 15 MW for projects coming online after 2022.

GE Renewable Energy launched its Haliade X 12 MW DD turbine in 2018 with the first prototype installed in Rotterdam for onshore testing from November 2019. When the Haliade X, which was the world's largest off-shore wind turbine model at the time, was released, the company stated that 12 MW was not the end, it was the beginning and will get bigger. This was proven true when its competitor, the world's largest offshore wind turbine supplier in total installations, Siemens Gamesa, released its SG14- 222 DD model in May 2020. This new turbine with potential to increase power will be commercially available from 2024. Vestas announced their V236-15MW platform in February 2021 which prototype will be installed in 2022 and serial production initiated in 2024. The most recent release is the Chinese company MingYang which in august 2021 announced the

MySE 16.0-242 (16 MW) which also is planned to have a prototype rollout in 2022 and initiation of commercial serial production in the first half of 2024.

In the following the currently officially released turbines are listed. It is expected that all of them have the potential to an increased rated power in the near future. The Vestas V164 was also originally launched with a 7 MW rating but has subsequently been upgraded to 9.5 MW (about 35% increase in rating).

- Vestas: V236 (15MW),
  - Commercially available in 2024
- Siemens Gamesa Renewable Energy: SG14-222 DD (14MW)
  - Commercially available in 2024
- GE: Haliade X (14MW)
  - Installation will begin at Dogger Bank in 2025
- MingYang: MySE 16.0-242 (16 MW)
  - Commercially available in 2024

A graphical representation of the increase in wind turbine MW rating over the last 30 years is available in Figure 1-22. Please note that the figure does not include the V236, Haliade X (14MW) and MySE 16.0-242.

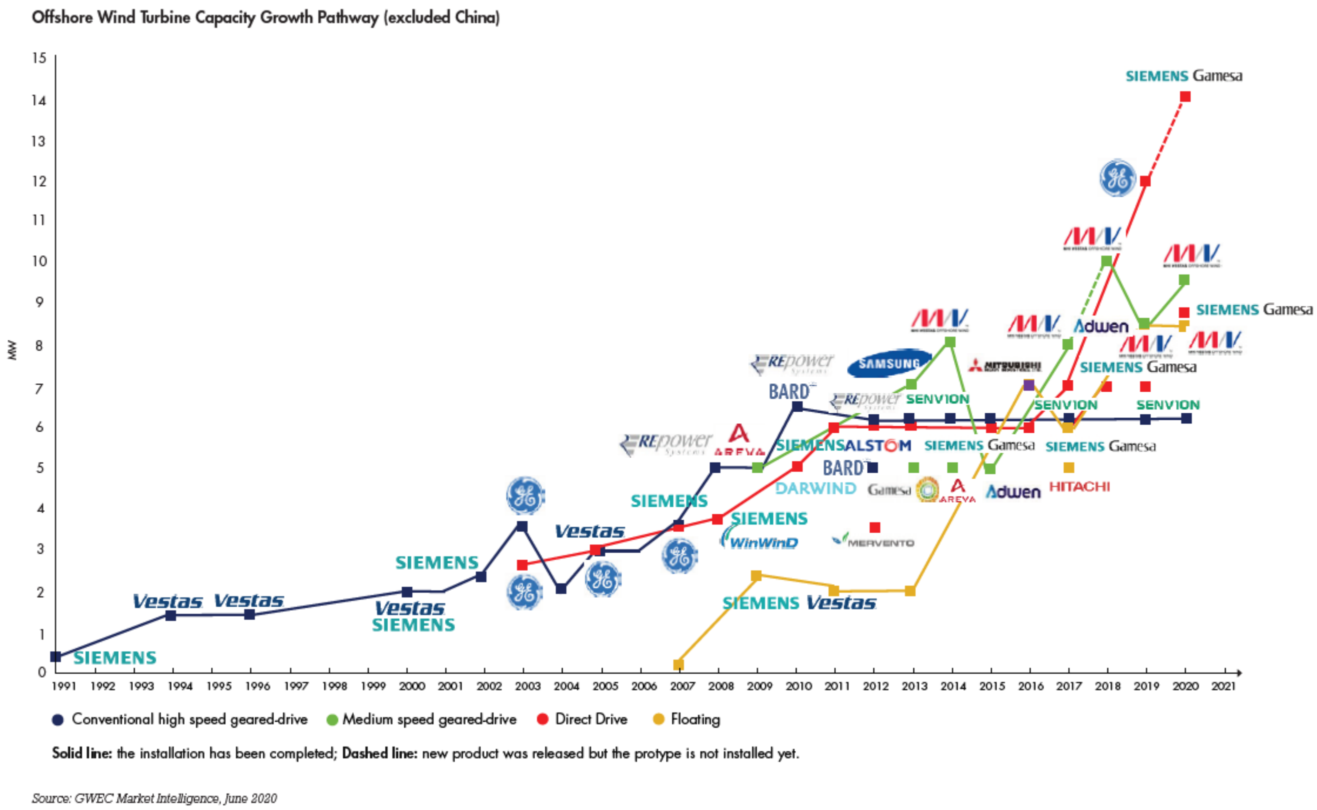

Figure 1-22: Historical development of wind turbine MW rating during the last 30 years.

The offshore wind pioneer Henrik Stiesdal predicted in a recent Global Offshore Wind Technology webinar that the next generation of offshore turbine technology could probably be around 20 MW with a 275m rotor diameter by 2030. This is in line with the prediction made by GE renewable energy depicted in Figure 1-23.

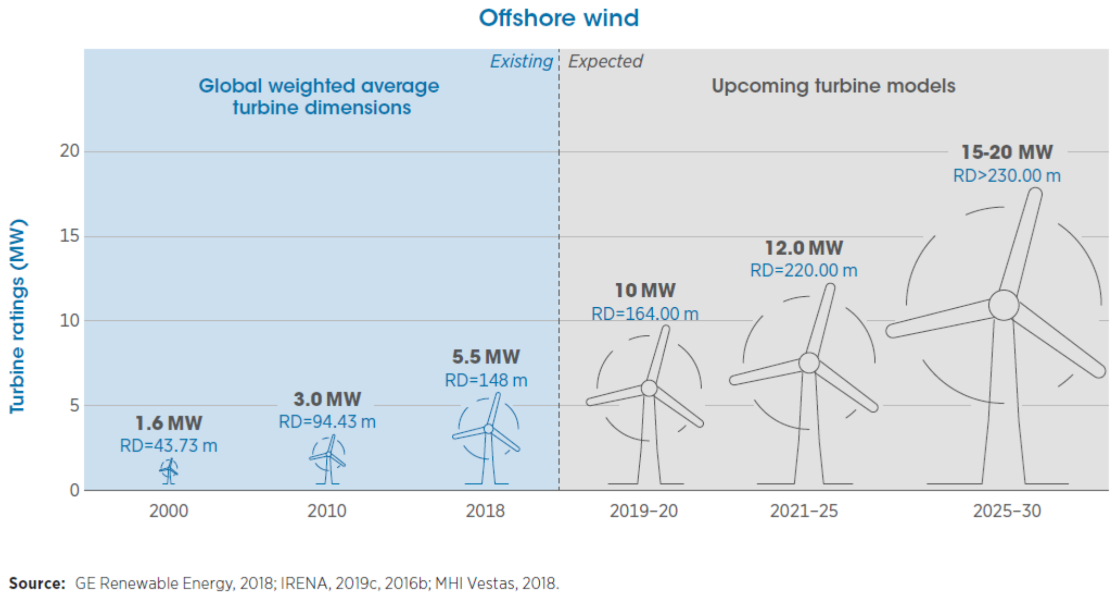

Figure 1-23: Prediction of future turbine sizes published by GE in 2018.

R&D see similar tendencies in their close cooperation with the OEM which indicate that the 20 MW milestone could be reached before 2030.

Keeping in mind an estimated time to market period for wind turbines of 2 year, all relevant components of a wind turbine including the main bearing have to be already developed and tested a certain time before releasing the turbine to the market.

The HALT-XL test bench currently has the capacity to support wind turbine development and test in the years to come, but further increases in test capabilities are expected, which the DD technology and Dual bearing load design is expected to be capable of supporting.

7. Project conclusion and perspective

7.1 MDS concept

Main conclusion for the MDS concept is that a system architecture which could be suited for a full-scale nacelle test bench has been identified. Main concerns are related to gear teeth wear and failure. These concerns were not possible to mitigate through testing of a scaled MDS demonstrator due to expectations that the issues observed and mitigations identified on a scaled-down demonstrator cannot be ap-plied to a full scale MDS driveline. Therefore, the issues have been investigated by means of material testing and component characterization. The characterization indicates that a thick and hard surface layer, a reasonable wide transition layer be-tween the hard surface layer and the soft matrix, and appropriate annealing to remove the internal residual stress may improve the gear properties and service life. Through a general investigation of bearing technology and a rotor dynamical analysis it is concluded that journal bearings could be very well suited to support the pinions on the MDS because the dampening introduced mitigates issues with testing at the natural frequency of the driveline.

7.2 DD concept

Main technological results for the DD concept have been that the design and performance of DD concept has been validated to such a degree that it is considered suitable be used for a full-scale nacelle test bench. To mitigate the technological challenges identified a full-scale DD coil segment was designed, manufactured, and tested. The test results indicate that the proposed design can be maintained at a steady state temperature during operation of 105° by means of a combination of water cooling and air cooling.

The development of the DD-technology for large scale test benches has contributed greatly to increasing the test capabilities within wind turbine development. It has also improved the market position of R&D test systems in general.

7.3 Future development

The DD technology developed will be further matured and is expected to be used in several large test benches in the years to come. The DD technology is expected to be scalable beyond 25 MW with minimal impact to the system architecture. Since the DD technology is already on operation at LORC, large amounts of data are collected from the condition monitoring systems active during operation. This will provide R&D with valuable data regarding performance of the system as well as the lifetime of different components. R&D expect the DD technology developed to be the preferred drivetrain solution for large scale test benches with high power and high torque requirements.

The MDS technology will need to be investigated further with focus on how to mitigate the risks identified in Table 1-2. Here most of the risks are related to material failure or wear due to either vibrations, or control or lubrication issues. DTU will investigate both wear and material failure in the spin off project Lim Wind where they will investigate how to limit white structure flaking failures in wind turbine gears and bearings via heterogeneous microstructure design mimicking nature. This project will focus on the fundamentals in microstructural design and as such pave the way for the future collaboration with the present team to transfer the fundamentals of microstructural design into component design and manufacturing. If failure in wind turbine gears can be sufficiently mitigated this could pave the way for the MDS project to be developed further and possibly be developed into a full-scale drive train technology and commissioned on a full-scale nacelle test bench. Based on the limitation within the design the MDS the concept is expected to be most suitable for solutions within 10-12,5 MW range since only 1 main gear ring will be required.