Measuring MNm torques as part of a prototype testing campaign of a high-temperature superconducting generator for wind turbine application in the scope of the Ecoswing project

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Abstract. Modern wind turbines have a driving torque in the range of up to 10 MNm. The measurement of such high torques poses challenges, as there is no standard measurement equipment for this torque level available today. During the EU funded EcoSwing project, the world’s first superconducting low-cost and lightweight multi-megawatt wind turbine generator has been designed and tested on the DyNaLab nacelle test rig of Fraunhofer IWES. For this test campaign, a specifically designed torque measurement system has been developed to measure the torque directly at the flange of the device under test in order to evaluate the efficiency of the power train.

1. Introduction
The aim of the EcoSwing project has been to build, test and operate a 3.6 MW generator with superconducting rotor coils. Prior to the field test on a wind turbine (see Figure 1), the prototype generator has been tested in various scenarios in the Fraunhofer IWES DyNaLab test laboratory. One of the key characteristics of a wind turbine is its efficiency. The higher the efficiency of the drive train, the higher is the energy yield. Due to the stochastic nature of the wind, the turbine operates primarily in different operating points. Its efficiency is not a constant value but varies with speed and torque of the drive train. The determination of an accurate efficiency map during field measurements is not possible as the wind field defining the input power can only be estimated. The availability of nacelle test benches as described in [1], however, facilitates a determination of the efficiency map at an even earlier stage within the development process. To
identify the efficiency of the drive train, the mechanical input power as well as the electrical output power are needed. However, accurate mechanical torque measurement in the range of MNm is very challenging and not state of the art. The largest torque measurement calibration system in Germany gauges up to 1.1 MNm [2]. Furthermore torque measurement systems in this class are expensive. In the EU-funded Ecoswing project, a novel high-temperature superconducting generator has been developed and tested on the nacelle test bench of Fraunhofer IWES. In order to determine the efficiency map of the generator, a cost efficient project specific torque measurement system has been developed as a demonstrator and integrated into the mandatory hub adaption between the test bench and the generator rotor. The unique torque measurement system is based on a force-lever system in a symmetrical arrangement, such that the driving torque coming from the outer flanges can be measured with three calibrated load cells on its way to the driven generator rotor (see Figure 2).

2. Kinematics and simulation
The adaption with the integrated measurement system has been designed in order to enable a testing scenario as close to real on-site conditions as possible. This requires the generator rotor to be loaded primarily with torque and shear forces during testing on the DyNaLab test rig. Thrust and bending moments shall not be transferred to the specimen.

2.1. Kinematics of the chosen approach
The design aim has been to create a measurement system that accounts for deformations and avoids significant bending moments in the generator shaft. At the same time, a direct and accurate measurement of the transmitted mechanical torque can be achieved via the load cells placed in the assembly’s rods. The rods are spherically mounted at each end in order to create a pendulum support (see Figure 2). Therefore only pure axial loads can be transmitted by the rods and captured by the load cells, while shear forces or bending moments acting on the load cells are avoided. In this way, the torque can directly be measured as the axial force in the rods via the proposed load cells, considering the lever arm of the system as depicted on the left hand side in Figure 3 (blue line). Because of the symmetry of the design the loads in the rods can be assumed to be approximately equally distributed over the three rods.

In the real wind turbine configuration, bending loads are not transmitted to the generator due to the rotor kingpin support and a torque transmission with a flexible carbon fiber main shaft. Shear forces however might occur. To simulate this scenario, the hub adaption with the integrated measurement system was designed to enable two rotational degrees of freedom, making it impossible to transmit bending moments from the test bench load application system (LAS) to the device under test (DUT). There might however be shear forces due to axial misalignment of test rig and generator flange. To evaluate the effects of such loadings due to misalignment of the shaft, a multi-body system (MBS) simulation model has been created for further analysis. Furthermore, the developed design provides an additional degree of freedom in axial direction of the shaft that can be constrained at an adjustable position. This allows for axial positioning of the adaption and shaft and consequently of the generator rotor with respect to the stator. Using this approach possible misalignments of the flange of the test bench load application system and/or the generator mounting adaption can be compensated. The degrees of freedom are shown in Figure 3 on the right hand side.
2.2. Modelling approach
The proposed design has been verified during the design process by means of a multi-body system simulation (MBS). The following topics have been addressed:

- Verification of the overall kinematics of the hub adaption system (i.e. two rotational degrees of freedom (DOF) as well as one axial DOF)
- Analysis of parasitic and/or unexpected forces on the load cells that do not act in axial rod direction, which is supposed to be the main loading direction for the load cells
- Estimation of shear forces and bending moments in the load cells due to centrifugal forces
- Assessment of the influence of gravitational loadings in a dynamic scenario.

The MBS model has been created in Samcef Field. For a purely kinematic analysis, all bodies are assumed to be rigid. As the solver requires a flexible system instead of a purely kinematic assembly, some artificial stiffness is added to the system: the six spherical bearings at the ends of the force levers are modelled as flexible bushings with both translational and rotational stiffness and damping. The load cells are modelled as flexible bushings as well, neglecting their mass and inertial forces as these are very small compared to the overall system loading. Inertial and gyroscopic forces are still acting on the load cells due to the mass of the rods. The pretension of the load cells is not considered in the analysis as only the resulting operational forces and moments are evaluated.

2.3. Simulation results
To examine the issues listed above, a simple run-up scenario was considered in the simulation as shown in Figure 4. The rotor speed increases from 0 to 12 rpm, which is roughly the rated rotor speed of the given generator, during a time span of 10 s. Simultaneously the torque increases from 0 to 3.6 MNm. After reaching these set points the operating point stays constant for another 10 s. The loading scenario of the model is achieved by imposing a rotational speed to the outer flange, while applying the torque to the inner flange of the adaption. Besides gravity no other forces or misalignments are considered in the first loading scenario.
Figure 4: Loading scenario for the simulations.

The simulation results in Figure 5 show that the main axial force in each load cell $F_x$ is almost constant over the rotational period and does only show negligible variations (see detail). The out-of-plane force component $F_y$ is almost zero and the radial force $F_z$ stays at about 250 N for the maximum rotational speed of 12 rpm. This component, however increases slightly, if the load cell mass is included, but it will not affect the results in a significant way. It is also visible that the bending moments experienced by the load cells are small and will not damage the load cells or compromise the measurements, as expected due to the pendulum support design.

Figure 5: Results of the first load case "perfect axial alignment". Forces and bending moments on load cell 1.
It is important to note that these results were obtained with a perfectly aligned force-lever-system assembly. If there are considerable bending forces/moments that might lead to misalignments, the load cells can experience higher loadings due to pretension and rotational imbalances.

In a consecutive load case, the influence of such radial displacements between the rotation axes of the driving and driven flanges is analysed. An external force in y-direction combined with an external bending moment in z-direction acts at the inner flange of the assembly (Figure 6). In order to evaluate realistic loading on the assembly, a finite element analysis has been performed to find the equivalent stiffness of the shaft attached to the inner rotational part. These values are used for the stiffness of the rotational support of the inner flange, i.e. generator side of the assembly.

For this loading scenario, the average axial force of the load cells at constant rotational speed stays about the same as for the case without shear forces, meaning that the same averaged torque is transmitted (Figure 7). However, compared to the first load case, a notable sinusoidal loading occurs, that is phase-shifted between the three load cells by 120° and oscillates with the frequency of drive train rotation (1P frequency). Again the non-axial loading on the load cells is negligible due to the pendulum support of the rods.

The reaction of the bearing support shows that the external shear force $F_y$ is mainly supported via the force-lever assembly. The applied bending moment $M_z$, however, is supported by the generator-side bearing support, as the assembly is designed to provide a rotational degree of freedom around the z-axis and consequently cannot transmit this load.

2.4. Kinematic analysis and mode shapes of the system

By means of a modal analysis of the MBS assembly, the kinematic degrees of freedom of the system have been verified. As intended by the design of the hub adaption, the simulation shows three main DOF modes: two rotational modes (around y and z axes) and one mode that allows axial movement of the inner flange against the other, allowing for an axial positioning of the generator as described above.

2.5. Conclusions of the MBS simulations

The simulation results show that the desired kinematic behavior was fulfilled with the designed system. The loading of the load cells in axial direction can be expected to comply with the static results of the analytic calculation even for the dynamic rotational loading scenario. It was further shown that only
minor non-axial loadings act on the load cells, i.e. unexpected damaging of the load cells due to parasitic loading should not occur. The harmonic loads from gravity are negligible compared to the target loads. Even with significant shear forces, e.g. caused by a misalignment of the rotational axis, the load cells do not experience unforeseen or critical loading conditions.

3. Preparatory tests

As all rod forces are needed to calculate the resulting torque, a force transducer is installed in each rod. The load cells have been calibrated in-house in a calibrated compression-testing machine taking the whole measuring chain into account (Figure 8 left).

The deployed force transducer can only measure compressive forces, but the rods needed to be designed to transmit tension forces. To enable the rods to measure operational tension forces, the load cells are adequately pretensioned with a centric bolted connection. Knowing the load factor $\Phi$ and the pretension of the bolt connection $F_V$, the operational load $F_B$ on the rod can be determined by the change of the clamping force $F_{KL}$ on the load cell. The correlations can be expressed through the bolted joint diagram (see Figure 8 right). The operational load $F_B$ can be calculated using the equation:

$$F_B = \frac{F_V - F_{KL}}{1 - \Phi}$$

The pretension $F_V$ is the reading of the load cell after assembly without operational load. During operation the readings from the load cells correspond to the joint clamp force $F_{KL}$. The load factor $\Phi$ is dependent on the elastic behavior of the material in the joint. Two approaches have been used to determine the load factor. On the one hand an analytical calculation was done as described in [3] and on the other hand experiments as described below were executed in accordance to the force calibration standard [4]. At first a deviation of 17% between both approaches was found. A finite element based calculation revealed an improper use of the load introduction factor $n$ in the analytical method. In this way, the deviation to the experimental results have been drastically reduced. This topic is currently further investigated. For the operational torque measurement the more confident load factor results from the experiments have been chosen as they have been acquired using calibrated measurement equipment.

The experiment consists of a test setup of the assembled rod in line with a calibrated reference load cell as shown in Figure 9. This setup allowed only a calibration in a partial load range (up to 27% of the total load cell
measuring range) due to the limited load application capacity. The reference load cell enables the evaluation of the load factor $\Phi$, because in contrast to the later operational use, now the operational force $F_B$ is also measured. In accordance to [4] load steps have been applied to each of the rods and the data has been read out with two repetitions. The load levels have been held constant until steady-state has been reached as the ropes are prone to creeping. Readings of the pretension have been taken during the assembly of the rods on the ground. According to the readings the pretension was set to 1400 kN for each rod. As Superbolt tensioners have been used a fine setting of the pretension has been possible using a balanced tightening process. The screw connection tends to creep, therefore setting times have been considered. After the setting time an averaged reading has been taken as the clamping force input for the load factor $\Phi$ calculation. After the calibration process the rods have been directly mounted to the drive train assembly.

The hub adaption with the integrated load cells facilitates the determination of the acting shear force in addition to the torque. As the connecting shaft with its length of 2.8 m is a relatively long force lever, the shear forces on the hub adaption have been identified as the critical loads for the integrity of the rotor shaft support. The rods as well as their axial forces have an angular distance of 120° to each other (Figure 10). The shear force can be calculated as the resultant force vector of the different rod forces. There are two passive effects to result in a shear force: eccentricity of the rotational axes and gravitational force, both can be quantified with the given instrumentation.

4. Assembly

For the hub adaption, a very thorough tolerance planning as well as quality assurance is needed as both aspects influence directly the quality of the measurement system. For the same reason, the assembly procedure has been planned in a step-by-step procedure. At certain steps a laser tracker system has been used to match the real positioning of the components to the designed one (see Figure 11).

The three rods connecting the driving flange of the test bench with the rotor of the generator have been preassembled and their lengths have been controlled with a laser tracker (length $x_{\text{rod}}$ in Figure 12). By means of shims the lengths have been adjusted to the distance measured between the connection points on the outer driving plate and the inner centric plate ($x_{\text{dist}}$ in Figure 12), which is connected to the rotor of the generator.
By these aforementioned means, the hub adaption can be further assembled with minimum pre-stress (see Figure 13), so that the impact on the measurement quality is minimized.

5. Test campaign conduction and results
The assembly process of the specimen to the test bench has been done without transmission of any bending moments as strain gauges on the connecting shaft have indicated (see Figure 14). A non-critical shear force has been measured. During conduction of the test campaign, no critical shear forces and thus bending moments have been monitored. A safe and almost free of parasitic loading operation has been possible. Figure 15 depicts the nacelle test bench with a dummy specimen adapted. This test bench in general can apply torque and parasitic loads onto the specimen, even though in the Ecoswing project the operation has been limited to the
torsional degree of freedom. During the Ecoswing project, a set of individual tests was performed. Unfortunately the specimen arrived later than expected and still needed to be finally assembled on site. This circumstance resulted in a much reduced testing scope compared to the original planning, such that the efficiency measurement has been limited to a couple of hours and below rated conditions. A more extensive testing of the hub adaption load measurement system has not been possible.

![Figure 15: General overview of the whole test setup and indication of power measurement positions.](image)

As mentioned earlier, the major aim of the integrated torque measurement is to determine the efficiency map of the high-temperature superconducting (HTS) generator. Figure 16 depicts the measured powers during the power test after low-pass filtering at the different positions along the drivetrain:

1. Test bench motor (signal PECMC_power_tp, dark blue line). The power was determined from built-in sensors from the converter. The measurement is afflicted with a high uncertainty, which is estimated as approx. 10%.
2. Hub adaption (signal LoadCell_power_tp, red). The power was determined using three load cells as described in this paper. The uncertainty calculated according to [5] is 2.4%.
3. Shaft adaption (signal Shaft_power_tp, light blue). The power was determined using strain gauges glued onto the shaft adaption. Again in this case, the estimated uncertainty is as high as 10%, which must be taken into consideration in the assessment of the results.
4. Grid connection point (signal JunctionBox_MV_power_tp, pink). The power was determined by measurement of voltage and current at the interface between transformer and IWES grid simulator. The measurement in this point is considered the most accurate with an estimated uncertainty of 0.5%.

One would expect the highest torque on the test bench motor (black) and the lowest one at the grid side of the specimen (pink), as all losses occur between these points. In contrast to this, the highest powers have been measured on the rotor adaption (red). This is not plausible, but the deviation of +9.3% remains explicable with the before mentioned large uncertainty of the test bench motor power. The measured power on the shaft adaption (light blue line) is 0.5% below the reference power of the drive. The signature is plausible, even though one would expect a larger difference resulting from the losses in the test bench drive. The measured power on the grid connection point (pink) is, as expected, the lowest of all power values. It is 4.4% below the reference input power on the driving side. Also in the comparison of these two signals, one would expect a larger difference as in this measurement all losses from the test bench and the specimen (generator, converter and transformer, excl. generator cooling system) are
included. Again, this deviation from the expectation can be attributed to the low accuracy of the power measurement at the test-bench drive.

In view of the large uncertainties of the power values obtained in positions 1 and 3 (i.e. the test bench motor and the shaft adaption), it is not meaningful to determine an efficiency based on these. Taking the load cell power measurement as reference instead, a specimen system (generator, converter, transformer) efficiency of 87.5 % (1.199 MW / 1.37 MW) with an approx. uncertainty of 2.5 % is determined.

6. Conclusion
In this paper, a comparatively inexpensive way to measure torque in the MNm range is presented. Its technical feasibility could be proven during the realization of the test campaign of a HTS generator on the DyNaLab nacelle test rig of Fraunhofer IWES as part of the project Ecoswing. Based on pre-test and uncertainty calculations according to [5], the uncertainty of the proposed torque-measurement method was determined to remain below 2.4%. This is an improvement compared to the use of strain-gauge measurements on the shaft adapter to the specimen. Nevertheless the achieved accuracy is not sufficient to determine the efficiency of a specimen with a reasonable certainty. Some potential for improvements has been identified, which will be further investigated in future nacelle test bench projects. An alternative approach for the challenging efficiency determination of machines with MNm torques and low rotational speeds was developed at IWES and described in [6]. While it requires additional testing scenarios, it has the advantage of a high accuracy. This provides a good basis for further evaluating and improving the torque measurement method described in the present paper.

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