Friction as a major uncertainty factor on torque measurement in wind turbine test benches

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Abstract. Most of the existing multi-MW nacelle test benches (NTB) measure the MN·m torque before load application system (LAS) to reduce the cross-talk effect of the multi-component forces and bending moments on torque measurement. This means that the friction torque of the LAS reduces the applied torque and consequently directly determines the input torque on the device under test (DUT). Therefore, the knowledge of the friction torque is necessary for the precise experimental investigations (e.g. efficiency measurement). At the beginning of this paper, a Computational Fluid Dynamics (CFD) simulation method for the determination of the friction torque of the LAS, which is suspended by hydrostatical plain bearings, is introduced. Subsequently, this method is validated with experimental results. Afterwards the friction torque of the LAS is quantified under different operation conditions (e.g. variation of rotational speed, multi-component load and temperature). Finally, the influence of the quantified friction torque on the uncertainty of the MN·m measurement is compiled.

1. Motivation and Objectives
Input torque measurement in MN·m range is one of the most relevant operating parameters during the investigations on the multi-MW NTB. Precise measurement of MN·m input torque is essential for the
- determination of the drive train efficiency,
- power curve determination and estimation of energy yield,
- control of the nacelle test bench and validation of simulation models as well as
- evaluation of the drive train behavior during critical operation modes [1].

Nowadays, the uncertainty of the MN·m torque measurement is between 2 % and 5 % with respect to the measured value [2]. This can be attributed to the lack of calibration machines and calibration methods above 1.1 MN·m [3] as well as to the unknown system-dependent influences on the torque measurement. Such specific, system-dependent influences in the multi-MW NTB are
- multi-component loads (forces and moments in the range of MN and MN·m),
- parasitic loads due to assembly misalignments and control of the load application system (forces and moments in the range of kN and kN·m),
- deformations, temperature distribution and rotational speed of the drive train
- gravity forces as well as friction of the load application system (LAS) [4][5].

Consequently, the MN·m torque measurement is not traceable according the metrological standards. Furthermore, current high measurement uncertainty does not meet the requirements of the industry and research facilities, which is in the range of 0.5 % for MN·m torque measurement with respect to the measured value [1]. To enhance the MN·m torque measurement accuracy a diversified consortium of
the national metrological institutes and NTB operators has been established within the scope of the EU-funded project “Torque measurement in MN·m range” [6].

During this project the system-dependent influences on the MN·m torque measurement have been determined and quantified on the example of the 4 MW RWTH NTB. It could be shown that the influence of the structural deformation under torque and wind loads can lead to an expanded standard torque measurement uncertainty of 0.116 % (coverage factor k = 2) [4]. Furthermore, the single influence of LAS system control (6·10⁻³ %), rotational speed (4·10⁻⁴ %), geometry of the transducer (3.5·10⁻² %) as well as friction and parasitic loads in the NTB gear coupling (4·10⁻² %) have a small contribution to the torque measurement uncertainty [2], [9]. One of the most relevant influences on the input torque of the device under test (DUT) and thus on the measurement uncertainty is the friction of the LAS with possible contribution to the torque measurement uncertainty between 0.1 % and 0.5 %.

Most of the existing NTB measure the MN·m torque before LAS. This means that the friction torque of the LAS reduces the applied torque and consequently determines the input torque on the DUT, see Figure 1. Therefore, the knowledge of the friction torque of the LAS is necessary for the precise experimental investigations.

In general, two possibilities exist to suspend the LAS main shaft either with roller bearings e.g. as tapered roller bearing in O-arrangement (10 MW FhG DyNaLab, 10 MW LORC HALT Test Facility, 6 MW CENER) or with hydrostatic plain bearings (4 MW RWTH, 5 MW NREL, 3 MW Catapult ORE). Validated and generally accepted state of the art methods exist for the calculation of the friction torque and power loss of the roller bearings such as [10], [11]. In case of the hydrostatic plain bearings there are several approaches for the determination of the friction torque under isothermal, laminar and steady flow, incompressibility, without consideration of the detailed geometry and body forces, see section 3.2 and [12]. These approaches cannot precisely describe the operation behavior of the specific custom-made hydrostatic plain bearings. Consequently, there is a need for numerical methods which are applicable for the calculation of the friction torque of the LAS. After a literature research no publications could be found that discuss the friction torque calculation of the LAS and its influence on the torque measurement.

The objective of this paper is to introduce and compare several methods for the determination of the friction torque of the LAS and subsequently to quantify the influence on the MN·m torque measurement uncertainty due to friction torque losses on the example of the 4 MW RWTH NTB.

2. Load Application System of the 4 MW RWTH Nacelle Test Bench (NTB)

The conventional multi-MW NTB comprises a prime mover for torque generation, an LAS for the generation of external forces and bending moments as well as DUT, see Figure 1. The LAS in NTB mainly consists of housing, load application disk and hydraulic cylinders. The load application disk of the LAS has to be suspended by bearings to allow the rotation and simultaneously to apply the wind loads with hydraulic cylinders. Furthermore, in the case of 4 MW RWTH NTB it is suspended by four axial and eight radial cylinders, which are preloaded, see Figure 2. Hydrostatic plain bearings are located between each actuator and the load application disk. In comparison with the roller bearing
suspension the hydrostatic plain bearings have lower friction torque, lack of stick-slip, lower deformation, longer life durability and higher reliability [17].

Maximum rotational speed, torque, forces and bending moments, which can be applied with LAS and prime mover of the 4 MW NTB can be seen in the Figure 3. Furthermore, this figure shows the operational loads of the LAS during investigations on the FVA nacelle [13], which have been used as the input loads for the calculation of the LAS friction torque.

| Loads       | LAS, Prime Mover | Operation of the PVA Nacelle |
|-------------|------------------|-----------------------------|
| Fx [kN]     | ± 4000           | ± 500                       |
| Fy [kN]     | ± 3250           | ± 200                       |
| Fz [kN]     | ± 3250           | - 590                       |
| Mx [kNm]    | ± 3400           | -1500                       |
| My [kNm]    | ± 7200           | ± 1500                      |
| Mz [kNm]    | ± 7200           | ± 1500                      |
| Rotational  | 30               | 17.5                        |
| Speed [min⁻¹]|                 |                             |

Figure 3. Load situation on the hydrostatic plain bearing system [14].

The friction torque of LAS depends on the rotational speed of the main shaft, multi-component loads, which are applied on the DUT as well as the temperature of the lubricant of the plain bearings. Two parameters contribute to the LAS friction torque: friction of the elastomeric seals of the main shaft and friction in the hydrostatic plain bearings of the LAS.

3. Approach
The friction torque of the elastomeric seals has been estimated according to [7] under consideration of the rotational speed as well as shaft diameter and reaches a maximum of 322 Nm. The main focus of this paper is the determination of the contribution of the hydrostatic plain bearing to the overall friction. In this case, the experimental method as well as CFD simulation method have been used for the determination of the LAS friction torque.

3.1. Experimental method
For the experimental methods, the measurements of the breakaway torque of the Prime Mover (PM) and LAS (without DUT) have been executed. Additionally, the empirical equation for the determination of the friction torque of the plain bearings, which is based on the measurements by manufacturer, has been used.

The breakaway torque comprises the share of PM friction (cogging torque and roller bearings) as well as LAS friction (elastomeric seals and hydrostatic plain bearings). The measurement has been executed to approximately confirm the manufacturer value of the LAS breakaway torque. The breakaway torque has been determined without any additional operating loads by measuring the force that is required to rotate the main shaft of the drive train from standstill. Subsequently, to determine the breakaway torque, the measured forces have been multiplied by the distance to the axis of rotation. Twenty measurements under steady state temperature of 65 °C of the lubricant have been executed. The breakaway torque of the entire system (PM and LAS) could be determined to 747 ± 99 Nm. The manufacturer value of the LAS breakaway torque could be qualitatively confirmed with 341 ± 45 Nm. The measurements are reproducible if the temperature of the lubricant and consequently the lubricant density and viscosity are held constant.

Empirical equation of the manufacturer for the determination of the friction force of the single plain bearing (not entire plain bearing system) under 23 °C is available. In this case, the friction force depends on the normal force on the plain bearing, linear velocity as well as two fit constants. The influence of
the temperature is not considered. By knowing the friction force under specific normal force and location of each plain bearing in the LAS as well as rotational speed of the main shaft, it is possible to calculate the friction torque of the entire plain bearing system. The empirical equation has been used to validate the simulation models under constant ambient temperature of 23 °C.

3.2. Numerical Simulation methods

The determination of the LAS friction torque should be as precise as possible to get precise input torque on the DUT. Because of the complex bearing geometry as well as different positions of the rotational axis of the main shaft and symmetry axis of the plain bearing, it is not possible to set up an analytical equation for the precise calculation of the friction torque. For the simulation method, a numerical approach, such as computational fluid dynamics (CFD) model (detailed bearing geometry, turbulent flow), has been selected.

CFD model has been used to determine the friction torque of the LAS by considering detailed geometry, turbulent flow, steady state conditions and rigid elastic surroundings. In general, CFD model solves the Navier-Stokes [8] equations for the discretized fluid volume. Furthermore, the turbulent flow of the plain bearing is modeled by using k-ε turbulence model, which uses equations to describe the turbulent kinetic energy k and rate of dissipation of turbulent energy ε. This model is a standard approach in the field of the CFD simulation.

The operating force on each of the plain bearing comprises the normal force caused by the operational loads as well as the preload force, which is necessary for the functionality of the LAS and for the avoidance of clearance. The normal forces have been calculated according to the force and moment equilibrium and are compiled in the Figure 4. Afterwards, the specified preload force is added to the normal force for the calculation of the operation force. Finally, the inlet pressure can be calculated by using the operational force and area of the plain bearing. In the considered case, the outlet pressure is atmospheric pressure. By knowing the geometry of the plain bearing, oil density and its viscosity, rotational speed, inlet and outlet pressure it is possible to finalize the pre-processing process of the CFD simulation.

The results of the CFD simulations contain the pressure field and pressure gradients as well as shear stresses and shear forces. Finally, the friction torque can be determined by multiplying the shear forces with the radius of the rotation. The CFD simulations have been executed for each of twelve cylinders under variation of the LAS loads (axial and radial forces up to 590 kN, bending moments up to 1500 kN·m), rotational speed (between 2 and 17 rpm) and lubricant temperature (between 20 °C and 80 °C).
4. Results

Figure 5 shows the comparison between the experimental approach and CFD method. The experimental approach has limitations, because it is valid only for one specific temperature: 23 °C, but it can be used best as a reference for this specific condition. The CFD method fits the experimental results well. In this case, the real geometry of the bearing as well as turbulence of the fluid have been considered, see Figure 5, right. Subsequently, the CFD method has been used for further sensitivity analysis (temperature, rotational speed, loads), because of a fact that it can easily consider the influence of the real operation conditions and is close to the experimental reference.

Figure 5. LAS Friction torque (plain bearing and seals) under 17 rpm, 23 °C and different LAS loads.

In the next step, the sensitivity analysis has been conducted by variation of the temperature of the lubricant, rotational speed, and operating loads. The results are compiled in the Figure 6.

The sensitivity analysis has shown that the friction torque decreases exponentially by the increase of the lubricant temperature. This can be attributed to the exponential relationship between the temperature and the dynamic viscosity of the used lubricant (0.048 Pa·s by 23 °C). The increase of the lubricant temperature by 10 °C leads to a decrease of the friction torque by about 285 Nm, see Figure 6 a).

The density and the dynamic viscosity of the plain bearing are influenced by the temperature. The relationship between temperature and lubricant density is linear according to the state of the art [15]. The dynamic viscosity decreases exponentially with higher temperature, e.g. according to the Vogel-Walter equations. This is the reason for the exponential decrease of the LAS friction torque with higher temperature.
5. Validation of CFD Method with Measurements on the 4 MW RWTH Nacelle Test Bench

The CFD method has also been validated by measurements on the 4 MW RWTH nacelle test bench. The validation has been executed by a constant main shaft velocity of 6.5 rpm, zero torque load and variation of the axial force and bending moment. The friction torque of the drive train (LAS, main bearing, gearbox) has been measured by a nacelle test bench torque transducer before the LAS, see Figure 7.

Figure 6. Influence of the lubricant temperature a), rotational speed b), bending moment c) on the total friction torque of the LAS plain bearings (without friction of the elastomeric seals of the main shaft). d) Influence of the normal force on the friction torque of a single LAS plain bearing.

Furthermore, an increase of the rotational speed leads to a linear increase of the friction torque of the LAS. An increase of the rotational speed by 10 min\(^{-1}\) causes an increase of the friction torque by about 440 Nm, see Figure 6 b). Consequently, the friction torque is more sensitive to the change of the rotational speed than that of the temperature. CFD results confirm the theory of the fluid friction according Newton's law for the Newtonian fluids (1) (\(\tau\) – shear stresses, \(h\) – film thickness, \(U\) – velocity, \(\mu\) - dynamic viscosity) [16]. Newton’s law implies that the shear stress \(\tau\) in a fluid film is directly proportional to the velocity \(U\) of the moving surface. Simultaneously, the velocity of the moving surface is directly proportional to the rotational speed. Consequently, the relationship between rotational speed and friction torque is also linear.

\[
\tau = \frac{\mu \cdot U}{h}
\]

Finally, the friction torque of the hydrostatic plain bearings increases nonlinear degressive under the increase of the LAS load, see Figure 6 c) as an example of the variation of bending moment \(M_y\). This can be attributed to the fact that the dynamic viscosity of the lubricant has a nonlinear behaviour under variation of the pressure load. In this case, a bending moment \(M_y\) of 1500 kNm leads to the friction torque of 705 Nm.

The applied LAS loads (e.g. bending moment \(M_y\)) generate specific normal forces on each of the radial and axial hydrostatic bearing of LAS. Figure 6 d) shows a relationship between the friction torque and these normal forces. This figure once again confirms the nonlinear degressive behaviour of the friction torque under variation of LAS load.
The simultaneous decrease of the axial force and bending moment by constant velocity leads to the decrease of the friction torque of the LAS as well as of the main bearing. The increase of the gearbox friction torque is very small, because the main bearing absorbs almost all external forces and moments.

On the one side, the developed CFD method has been used for the calculation of the LAS friction torque. On the other side, the friction torque of the main bearing of the nacelle drive train (roller bearings) has been calculated by analytical method of the bearing manufacturer [18]. The main shaft of the nacelle is mounted by the three-point suspension. In this case the fixed bearing (self-aligning roller bearing) is located near the hub adapter. The floating bearing (cylindrical roller bearing) is located in the gearbox (as planet carrier bearing).

It could be shown that the friction torque of the LAS due to external loads is much smaller than the friction torque of the main bearing, see Figure 7. Furthermore, the qualitative trend of the calculated friction torque follows the measured values well. The deviation between the calculated and measured friction torque can be described by the influence of hysteresis and tension effects, control errors, gearbox friction as well as CFD method inaccuracy. Nevertheless, the validation results show that the introduced CFD method can be used for the determination of the LAS friction.

6. Summary

The investigations have shown that the fluid friction in the plain bearing contributes almost 80 % (20 % elastomeric seals) to the overall friction torque of the LAS. The highest friction torque of the LAS is in the range of 1.5 kN·m. This friction torque is a systematic error, which should be compensated. Neglecting this error under operating torque of 1500 kN·m will lead to an uncertainty of 0.1 %, under operating torque of 300 kN·m to 0.5 %, see Figure 8.
Consequently, the friction torque of LAS cannot be neglected because of the actual requirement on the MN·m torque measurement uncertainty of maximum 0.5 %. The introduced simulative methods can be adapted on different design principles of LAS and can be used to determine the friction torque of wind turbine plain bearings. In addition, by the accurate torque measurement it is possible to determine the efficiency of the wind turbine drive train or to compare different wind turbine designs more precisely.

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