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Published in:
Journal of Constructional Steel Research

DOI:
10.1016/j.jcsr.2020.106361

Publication date:
2021

License:
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Document Version:
Accepted author manuscript

Citation for published version (APA):

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Bolted ring flanges in offshore-wind support structures - in-situ validation of load-transfer behaviour

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Abstract

Offshore wind turbines are subjected to cyclic loads from wind and waves, ultimately resulting in a fatigue-driven design. These cyclic loads result in cyclic shell forces above the bolted ring flange and ultimately into the cyclic loading of the bolt tension forces. To model the transfer of the shell forces into the bolt, Load Transfer Functions (LTF) are used, of which the tri-linear function according Schmidt/Neuper is the most known approximation but more contemporary approaches exist. As the LTF has a significant impact on the estimated fatigue lifetime of the bolt, it is relevant to investigate the accuracy of the currently used LTF. This paper uses in-situ monitoring data from three offshore wind turbines at the Nobelwind windfarm, each equipped with an instrumented bolt in the monopile to transition piece bolted flange connection and strain-gauges above said flange. Long-term monitoring data is used to derive empirical load transfer coefficients (LTC), i.e. the derivatives of the LTF. In parallel a finite element model (FE model) of the flange connection was developed. The found load transfer coefficients from measurements were compared to the values obtained from the function according to Schmidt/Neuper and the FE model.

It is concluded that the observed LTC in the offshore wind farm were favourable compared to the values expected from Schmidt/Neuper. The FE model matched more closely, especially when the MP-flange inclination was included. The inclusion of the MP-flange inclination in the model had a positive effect on the final LTF, which was confirmed by the measurements.

Keywords: (Offshore) Wind energy, Bolted flange connection, Load transfer function, Measurements in-situ, FE Modeling

1. Introduction

1.1. Motivation

Offshore wind turbines (OWT) are typically built up out of three main parts; tower, transition piece (TP) and foundation. The tower structure is part of the wind turbines and constitutes several bolted ring-flange connections between the different tower segments. This configuration is common with most onshore wind turbines with a steel tower. Unique to offshore wind is the TP and foundation, which bridges the bottom of the tower, typically several meters above highest possible sea-water level, and the seabed. The combination of TP and foundation is typically referred to as the substructure of the OWT. While several designs exist for the substructure, the market is currently dominated by monopile and jackets, respectively representing 81.9% and 6.6% of installed capacity in 2018 [27]. In both jacket and monopile substructures the bolted ring-flange connection is the most common connection between tower and substructure. For monopile substructures the connection between transition piece and monopile (MP) foundation has historically been implemented using a grouted connection [28, 13]. However, the early concerns with the grouted MP-TP connection and increasing demands due to growing turbine sizes, have led industry to look into alternative connections for the MP-TP interface, including innovative concepts such as the slip joint [23], double slip joint[29] and the wedge connection [30]. However, the most popular alternative to the grouted connection for the MP-TP interface has been the bolted ring-flange connection. In recent years several offshore wind farms have adopted the bolted ring-flange connection, starting with the Amrumbank West offshore wind farm. The 2017 offshore wind farm Nobelwind was the first Belgian offshore wind farm with a bolted connection between MP and TP. The Nobelwind offshore wind farm consists of 50 3.3MW Vestas wind turbines on monopiles, Figure. 1.a. As part of the research project O&O Nobelwind research effort was put into the
behaviour of the novel bolted MP-TP connection, Figure 1.b, comprising 84 M72 bolts, Figure 1.c. Three
28 turbines were instrumented with measurement bolts, see also Section. 2, to validate the design and monitor
29 the long term behaviour of the bolted connection. In this contribution the transfer of cyclic loads from the
30 shell into the bolted connection is investigated. Which is a key element in the determination of the fatigue
31 lifetime of the bolts.

1.2. Bolted ring-flange connections in offshore wind turbines

Ring flanges in wind turbines are usually connected with hot-dip galvanized, high-strength bolt assemblies
35 (HV-sets) with large diameters M36 and bigger according to (DIN EN 14399-4 [5]) and (DASt - Guideline
36 021 [3]), respectively. The magnitude of the acting loads on the support structures due to wind and wave
37 usually requires the application of very large bolt diameters M64 or M72, Figure. 1(c).Throughout their
38 service lifetime, HV bolting assemblies are subjected to cyclic loads with considerable numbers of load cycles
39 and variable amplitudes. In the design practice, the calculation of the cyclic load amplitudes $\Delta F_b$ requires
40 the approximation of the load transfer behaviour from the tower shell to the HV-bolt inside the flange. In
41 accordance with the current design standards, this is performed with a load transfer function (LTF). Figure. 2
42 schematically illustrates the LTF $F_b(Z)$, with the external load $Z$ acting in the tower shell on the abscissa
43 and the bolt load $F_b$ on the ordinate.

For the reduction of fatigue loads, preloading of bolts with a force $F_p$ is essential for the structural in-
44 tension. The preload is induced by either torque controlled or tractive tightening methods. With the broadly
45 used torque controlled tightening procedure the bolt shall be preloaded up the reduced nominal Preload
46 $F_{p,C} = 0.7 R_{p0.2} \cdot A_{sp}$ according to (DASt - Richtlinie 021 [3]). Thereby, $R_{p0.2}$ is the materials plastic strain

Figure 2: Schematic illustration of the load transfer function under perfect and imperfect geometrical flange conditions; centric
and eccentric clamp solid due to the preload depicted in red
Under perfect geometric conditions – as the surface pressure between the flanges increases due to the tightening – a clamp solid emerges centrically around the longitudinal bolt axis as a counterpart of the preload, see Figure 2 where the clamp solid is depicted in red. Due to the clamp solid and the eccentricity in the load transfer inside the connection, the load transfer from the tower shell to the bolt is nonlinear. With the clamp solid located symmetrically around the bolt axis, external forces acting in the tower shell first diminish the clamp solid before stressing the bolt. However, under imperfect geometrical flange conditions, meaning the flange contact surface are not ideally plane due to a flange inclination $\alpha_s$ or a flange opening/gap $k$, the clamp solid may emerge eccentric from the bolt’s longitudinal axis. In the worst-case scenario, the clamp solid emerges flange sided, see Figure 2 (right), not being able to diminish the bolt force amplitude $\Delta F_b$ significantly. Eventually, under these conditions, the load transfer behaviour is characterised by a much steeper and less favourable load transfer function (LTF) because $\Delta F_b$ increases. Apart from the geometric flange imperfections, also the material properties, the preload level and the selected HV-sets have a major influence on the shape of the load transfer function.

Fatigue verification of HV-Bolts is usually based on the nominal stress approach in combination with classical linear damage accumulation hypothesis. The applicable S-N curves from EC3 [1] (FAT 50) and VDI 2230 [4] have been validated in [19, 16, 17] for HV-bolt sets of bolt sizes M36, M48 and M64; as a consequence reducing uncertainty in the verification against fatigue. Furthermore, fatigue of HV-bolt sets is affected by a variety of parameters. The investigation of these parameters with experiments is time consuming and the possibilities, especially for large bolt assemblies, are limited due to lack of testing facilities with sufficient testing frequency and load level. Thus, advanced analytical and numerical models have been applied in [18] to investigate the influence of further parameters, such as bending loads, on the fatigue of HV-bolt sets. In his PhD thesis Eichstädt [8] presents elaborated fatigue analysis with the strain notch approach depicting the influence of bending among other parameters.

### 1.3. Design practise of bolted ring flange connections

In accordance with DNVGL-ST-0126 [6] and the German design guideline for wind-turbine support structures DIN 18088-3 [7], the load-transfer behaviour can be approximated by utilising the segment approach. In this approach, the focus lies on the most heavily loaded ring flange segment, see Figure 3. Utilising elastic beam theory, a LTF can be derived which describes the load transfer from the tower segment to the bolt. The segment approach is easy to implement, because analytical approaches exist, which allow the approximation of the LTF. Such are for instance the bilinear approach according to [15], the tri-linear approach according to [21], the nonlinear approach according to [24] and the approach according VDI 2230 [4]. However, with the segment approach spatial load distribution is neglected. Furthermore, explicit consideration of flange imperfections is not possible. Flange imperfections between two flanges can be characterised by the flange inclination $\alpha_s$, the flange opening $k$ and the opening length $l_k$, see Figure 3(b).
In the fatigue design practice, the tri-linear load transfer function according to [21] is state of application for the calculation of fatigue loads $\Delta F_b$. The function is easy to implement and is in use for designing ring-flange connections for wind-turbine support structures for almost twenty years. In general the Schmidt/Neuper approximation of the load transfer function is conservative. However, it is acknowledged that it therefore implicitly covers imperfections of various shapes and magnitudes, see also Section 1.4. However, with increasing flange and bolt diameters in the offshore wind industry the applicability of this function is questioned.

The load transfer behaviour of a ring flange connection can be characterised by four ranges in total as a function of external stresses, these four ranges are [24]:

1. Stresses between the flanges are small, but the contact is closed.
2. Stresses increase resulting in a successive opening of the contact and in a degradation of the clamp solid.
3. Large stresses lead to an open connection. In this range the slope of the LTF depends on loads and geometry of the flange.
4. Stresses leading to a plastic deformation of the flange and bolt.

For the estimation of fatigue loads, the last range is of less importance. In fact, the most fatigue loads appear in the first range. Figure 4 depicts a schematic illustration of the LTF according to Schmidt/Neuper [21] (SN). The LTF according to SN is a progression of the bilinear approach according to [15]. Petersen’s function distinguishes between a non-gapping and a gapping connection. With the trilinear approach, a third range is introduced allowing the differentiation between a closed connection (1st range up to an external load $Z_I$), a range describing the successive opening of the flange (2nd range up to an external load $Z_{II}$) and a range describing an open connection with a maximally reduced clamp solid and the resulting clamp force $F_D$.

The LTF according to Schmidt/Neuper is defined by the pretension $F_p$, the geometry of the bolt and the flange ($t_f, a, b, s$) as well as the stiffness of the flange and the clamp solid, $EI$ and $C$ respectively. Figure 4 depicts an elevation of a ring flange connection and the corresponding elastic-static beam model according to [15]. The flange is idealised as a beam. The connection of the flange to the tower is characterised by a torsional stiffness $C_{\theta}$. However, this stiffness is neglected by both Petersen [15] and by Schmidt/Neuper [21]. The clamped effect of the connection is approximated by the axial stiffness $C$, which is the sum of the axial bolt stiffness $C_s$ and the stiffness $C_d$ of the clamp solid. These stiffnesses are combined into the variables $p$ and $q$:

$$ p = \frac{C_s}{C_s + C_d} $$

$$ q = \frac{C_d}{C_s + C_d} $$

Schmidt/Neuper utilize and further develop this approach in [21]. The calculation of the three ranges of the

![Figure 4: Schematic illustration of the LTF according to Schmidt/Neuper [21]](image-url)
LTF according to Figure. 4 are carried out by the equations:

\[ F_{b,1} = F_p + pZ \]  

\[ F_{b,2} = F_p + pZ_I + (\lambda^*Z_{II} - (F_p + pZ_I)) \frac{Z - Z_I}{Z_{II} - Z_I} \]  

\[ F_{b,3} = \lambda^*Z \]

(3)  

(4)  

(5)  

(6)

The external load \( Z_I \) at which the computational successive opening of the flange begins can be calculated by:

\[ Z_I = \frac{a - 0.5b}{a + b}F_p \]

(7)

The external load at which the clamp solid is completely diminished can be calculated by the equation:

\[ Z_{II} = \frac{1}{\lambda^*q}F_p \]

(8)

The parameter \( \lambda^* \) is related to the geometric features \( a \) and \( b \) and the location of the resulting counter force at the inner edge of the flange. Assuming that the location of the counter force is at 0.3a from the inner edge, \( \lambda^* \) can be calculated by:

\[ \lambda^* = \frac{0.7a + b}{0.7a} \]

(9)

Note that the slopes of the linear functions describing the first and the third range are functions defined by the coefficients \( p \) and \( \lambda^* \), the intermediate part of the LTF results from interpolation between \( F_b(Z_I) \) and \( F_b(Z_{II}) \).

The trilinear model was verified by the original authors with the help of finite element calculations for perfect flange geometries. Thereby, a satisfactory agreement for thick, compact flanges was found. For thinner flanges the results of the model, however, are insufficient. Therefore, a limitation criterion was established which limits the applicability of the model. The width of the flange must not exceed three times the Flange thickness

\[ \frac{a + b}{t} \leq 3 \]

(10)

Finally, according to the German standard DIN 18088-3 [7] and DNVGL-ST-0126 [6] a simplified fatigue design, with the LTF according to Schmidt/Neuper, is possible. Using this LTF, flange imperfections are not considered explicitly but implicitly due to the functions conservatism. Furthermore, only bolt axial forces are considered in fatigue load calculation. These simplifications result in the fact, that the fatigue damage is calculated with FAT36 from the DIBt-Guideline [2] instead of FAT50, which is the actual fatigue strength of axially loaded HV-bolts according to DIN 1993-1-9 [1].

1.4. Flange imperfections

Flange imperfections have a significant influence on the LTF. In [20] Schaumann and Seidel firstly present measured LTFs of a ring flange of an onshore wind-turbine support-structure. The results of the field tests reveal strong fluctuations of the measured LTF. The authors highlight that the cause lies in the influence of geometrical flange imperfections. Jakubowski performed first systematic experimental and numerical analyses on the influence of flange imperfections on the LTF and summarized the results in his dissertation [10]. According to [10] the imperfections can be classified in flange sided, tower sided and parallel gapping, Figure. 5.

Figure. 5 shows that the location of the clamp solid and the resulting clamp force strongly depends on the imperfection type. In [11] the authors present results of large scale experiments of imperfect ring flanges. The experimental investigation encompass the presented imperfection types in Figure. 5. The results show that parallel gaps are the most difficult to close. Closing of these gaps results in significant bolt bending and an unfavourable flange sided clamp solid. The tower sided gapping affects the LTF less severe. In contrast, the authors of [11] observe that the flange sided gaps have a positive effect on the LTF, since it causes the resulting clamp force to move closely to the tower wall and thus to the resulting axis of external forces. Further experiments on imperfect ring flanges have been performed by Feldmann et al. [9]. The authors
confirm the results presented in [11] and endorse the suggestion to shim the flanges with shim-plates if gaps are too big.

The LTF according to Schmidt/Neuper does not consider geometric flange imperfections explicitly but implicitly. The current German design standard DIN 18088-3 [7] as well as the DNVGL-ST-0126 [6] for wind turbine support structures lists requirements for the flange imperfections such that the LTF can be approximated with the Schmidt/Neuper method: The flatness divergence of one flange must be less than 2\( \text{mm} \) over the entire flange circumference. In addition, it must be less than 1\( \text{mm} \) over a segment of 30\(^\circ\) after the completion of the tubular sectors. In accordance with the insights from [11] only the outside divergence values are of importance. It is further mentioned in DIN 18088-3 that after bolt preloading flange angle \( \alpha_s \) must be less than 2\(^\circ\). Especially for large flange and bolt diameters, these requirements are rather difficult to fulfil without a major effort during the fabrication.

2. Field measurements of bolt forces

At the Nobelwind offshore wind farm three out of 50 turbines are equipped with instrumented bolts to measure the bolt forces \( F_b \) in situ, these turbines were BBG08, BBG10 and BBK05. The geometry of the bolted connection at Nobelwind is described in Section 2.1 along with a discussion about the known flange imperfections. The measurement setup is discussed in Section 2.2 and the methodology of how the data is processed is provided in Section 2.3.

2.1. Flange geometry

2.1.1. General geometry

All turbines at the Nobelwind wind farm have an identical flanged connection, only tolerances on the manufacturing of the flanges may differ within the farm. All flanges consist out of 84 bolts at a design pretension of 2439 kN.

2.1.2. Flange imperfections

In the scope of the Nobelwind Project flange tolerances have been measured for each of the MP-TP connection. The tolerances satisfy the design standards (DNVGL ST-0126 [6], and DIN18088[7]).

All MP flanges have rather small flatness divergence along the flange circumference. A large part of the MP-flange is tapered and the rest (approx. 20\%) is horizontal. This area is meant for the anvil during the pile driving the monopile into the seabed. In total, the MP flanges of all locations are rather smooth. The TP flanges of the three locations show generally stronger imperfection magnitudes. However, the flanges still fulfil the criteria according to the design standards. The TP flange at location G10 is the flange with the largest divergence over the entire circumference. It is furthermore the only flange that is inclined “downwards” towards the TP-center. The flatness divergence over a segment of 30\(^\circ\) for all three flanges also fulfils the requirement of the design standards clearly.

The requirements defined in DIN 18088-3 and the DNVGL-standard have been originally derived by Seidel and Faber [26], which was implemented and later partially adjusted in the DIBt Guideline for Windturbines.
[2]. The requirements are partially based on scientific basis and partially on experience of a technical com-
mittee [25]. Initially, the requirements have been developed for rather constant tower diameters for onshore
turbines. Thus, the 1 mm criterion to a 30° segment is related to gap length of approx. 1 meter. With the
increasing flange diameters in the on- and offshore-wind industry the 30° segment encompasses a larger gap
length. A driving factor influencing the gap closing behaviour is the gap length and the tower wall thickness
[25]. This is also shown in [14] with finite element analysis of various imperfect ring-flange models with bolt
sizes ranging from M36 to M72 as well as various wall thicknesses. The results elucidate that gaps having a
shorter length in combination with thicker tower walls are more difficult to close by the preload.

To account for this circumstance, Seidel published a suggestion in [25] for a different perspective on the
requirements. The criteria presented by Seidel are based on linear elastic models and focus on the closing
behaviour of parallel gaps only. The results reinforce openings with larger gap length are easier to close than
those with shorter gap length. Thus, Seidel proposes to focus at the flatness tolerance of a 90° sector instead
of the entire flange. Furthermore, it is proposed to relate the 1.0 mm criterion to a segment of a 1 m length.
For the here presented flange diameters, this is equivalent to a 25.5° sector. Eventually, the DIBt-Guideline
as well as [25] give both recommendations for the flatness tolerance of one flange because during fabrication
the imperfection of the counterpart is unknown. However, the final gap shape as well as gap length – after
the two flanges meet on site – is of importance.

The authors have evaluated the final gap magnitude $k_1$, $k_o$ and length $l_k$ on the basis of the measurement
protocols of the flange tolerances. At all three locations, the HF-bolts do not lie a ring flange sector with the
largest gap magnitude at the outside of the flange. The decision making of the placement of the HF-bolts
in the flange did not initially account for flange imperfections. Other criteria – such as location related to
wind direction – have been selected. Eventually, the HF-bolts at all three locations lie in areas with rather
indulgent gap shape and magnitude. The gap magnitude $k_o$ varies between 0.23 mm and 0.33 mm and the
gap length between 130° and 200°. In comparison with the requirements according to DIBt-Guideline [2] and
the requirements derived by Seidel [25] the gap magnitudes at the areas of the three HF-bolts are significantly
smaller and therefore noncritical. Thus, the gaps at these location are expected to be closed easily by the
preload leading to contact pressure between the flange surface. Furthermore, at the location of the HF-bolt,
the flanges are additionally open flange sided. Thus, presumably the clamp-force is located eccentric towards
the outside of the flange, see [11]. Consequently, resulting in an favourable clamp solid leading to a rather
small slope of the LTF.

2.2. Measurement setup

At Nobelwind three wind turbines are equipped with a measurement bolt referred to in this paper as
an High frequency-Bolt (HFB). For these bolts the data acquisition is performed using an external data
acquisition system that reads out the internal strain gauge. Using this hardware a high sampling frequency
of 100 Hz is possible without any intermission. An example of an installed HF-bolt is shown in Fig. 6.(b).
The instrumentation bolts were distributed considering the dominant South-Western wind direction. In
addition to the HFB each turbine was instrumented with 4 low frequency measurement bolts. These bolts
have their own battery powered data acquisition, but only provide an estimate of the mean bolt tension at
discrete intervals, e.g. once every 10 minutes.

Aside from the instrumented bolts the setup at Nobelwind also features accelerometers and both resistive and
optical strain gauges at different levels of the OWT (including the monopile). For the current investigation
the 6 axial resistive strain gauges, installed approximately 35 cm above the MP-TP flange are of interest.
The strain gauges are equally distributed around the circumference and are referenced to based on their heading
w.r.t. North. In the current setup strain gauges are spaced every 60 degrees, starting at 30 degrees. Each
strain gauge is paired with a temperature sensor that is used to compensate the temperature dependency of
the strain gauges. Each strain gauge is individually read out in a quarter-bridge configuration. Two of these
strain gauges are shown in Fig. 6.(b).
2.3. Data processing

2.3.1. Measurement of shell force $Z$

The recorded strains $\varepsilon_m$ at each position can be related to the stresses in the tower shell $\sigma_s$ through:

$$\sigma_{s,m} = E\varepsilon_m$$ (11)

in which the Young’s Modulus $E$ is set to $210\,\text{GPa}$. However, the strain gauges are not positioned directly above the measurement bolts. The measured shell stresses thus need to be transformed to the heading of the measurement bolts. This is achieved in two steps. First, the bending moments are calculated using the strain gauges. This calculation is done using the as-designed geometry of the cross-section, the known positions of the strain gauges $\theta_m$ and bending theory of a hollow cylinder:

$$\sigma_{s,m} = \frac{N}{A} + \frac{R_i}{I_c}(M_y \sin\theta_m - M_z \cos\theta_m)$$ (12)

in which cross section surface area $A = \pi(R_o^2 - R_i^2)$ and moment of inertia $I_c = \frac{\pi}{4}(R_o^4 - R_i^4)$. With $R_o, R_i$ respectively the outer and inner radius of the TP at the sensor positions just above the flange.

Eq. (12) is computed for each of the 6 strain gauges. Using these six equations the bending moments $M_y, M_z$ and axial force $N$ are computed using a least squares formulation[12]. As the strain gauges were installed after the OWT was completed it is not possible to derive an accurate estimate of $N$ from the measurements. Instead the strain gauges are calibrated to have $N = 0$. An estimate of $N$ is obtained from the combined mass of the Rotor nacelle assembly, the turbine tower, auxiliary components and the total weight of the TP, resulting in an estimated compression load $\hat{N}$ of $-4331\,\text{kN}$.

In a second step the shell force $Z$ above the bolts is calculated by using the found bending moments $\hat{M}_y$ and $\hat{M}_z$, the estimated load $\hat{N}$ and replacing sensor position $\theta_m$ with the position of the bolt $\theta_b$ in Eq. (12).

$$\sigma_{s,b} = \frac{\hat{N}}{A} + \frac{R_i}{I_c}(\hat{M}_y \sin\theta_b - \hat{M}_z \cos\theta_b)$$ (13)

The shell force $Z$ is then calculated using:

$$Z = \frac{\sigma_{s,b} A}{N_{\text{bolts}}}$$ (14)

The shell force in absence of any bending moment on the flange, and thus originating purely from $\hat{N}$ is referred to as $Z_0$. 

Figure 6: Pictures of the instrumentation used in this project. The instrumented HF bolt is combined with 6 strain gauges above the bolted connection to determine the shell force $Z$. 
Example. In Fig. 7 a 30 minute record of the strains recorded on the shell of BBG08 is shown during a stop-start cycle. At the start of the record, up to 475s, the turbine is operating under normal load. The main bending moment is aligned with the wind direction, approx. 255 degrees. Strain gauges in the upwind positions (210, 270, 330) record a positive strain, while sensors on the opposite (downwind) side record a negative strain. At 475s the turbine is stopped. As a result the main bending moment is reduced, i.e. the thrust load disappears, and strains in the shell all reduce to approximately 0 µm/m. With the turbine stopped the system is subjected to a minimal quasi-static bending moment. At 1500s the turbine is started again into normal operation and strains return to their normal values. A similar event was (almost) simultaneously triggered on the two other turbines.

The record shown in Fig. 7 offers an interesting view into the loading of the turbine. Within a 30 minute window the load is varied from a (large, quasi-static) operational load, to near zero load.

The recorded strains can be transformed into the shell force using the method proposed in the previous section. In Fig. 8 the bolt tensions recorded by the HFBs on all three turbines are plotted along with the associated shell forces $Z$.

The results in Fig. 8 immediately confirm that the variations in the bending moments are transferred into the bolts. Both the shell force, and the resulting bolt tension, variations differ between the three turbines due to the different position of the HFB in each turbine. At G10 and K05 the bolts are positioned at upwind positions, at 317 and 231 degrees respectively. As a consequence shell forces at K05 and G10 are in tension ($Z > 0$) when the turbine is operational (up to 475s). In contrast the HF bolt at G08 is positioned at the
downwind compression side at 51 degrees, resulting in $Z < 0$. As discussed earlier, between the stop (475s) and the start (1500s) the bending moment is reduced to almost zero, as a result the shell force reduces to the axial contribution $\frac{N}{N_{\text{bolt}}}$. For K05 and G10 this results in a drop in the tension in the bolt. For G08 the removal of the thrust load results in an increase in the tension. The measurements at all three turbines confirm that shell force variations $\Delta Z$ are transferred into the bolt for both tension and compression loads.

2.3.2. Estimation of load-transfer behaviour

The main objectives of the current work is to investigate the transfer of cyclic loads from the shell into the bolt, with a particular focus of fatigue loads. As both the shell forces and the bolt forces are available at the three Nobelwind turbines it is possible to reconstruct the load transfer function (LTF) from field measurements.

Fig. 9.(a) shows the relation between the shell forces and the bolt forces (LTF), during the same 30 minute period as shown in Fig. 8. As the three turbines at the time of the event had slightly different pretensions $F_v$, the y-axis represents the variation from the pretension ($F_b - F_v$). Fig. 9 shows that the slope of G08 is steeper than those of the two other turbines. The slope of the graphs is the load transfer coefficient (LTC) between the shell force variations $\Delta Z$ and the bolt force variations $\Delta F_b$.

From the results in Fig. 9 a single estimate of the load transfer coefficient can be derived. However, as data is continuously available from a long period of monitoring it is possible to determine the LTC over the long term, covering a large variety of environmental and operational conditions. In the current research long term monitoring data is processed in ten minute intervals, in which the LTC is estimated independently for each interval. The interval length of ten minutes is consistent with the standard interval length of the turbine Supervisory Control and Data Acquisition (SCADA) system. The SCADA system collects relevant environmental parameters such as wind speed and wind direction as well as parameters related to the control of the turbine. The ten minute interval length was motivated by the assumption that wind and wave conditions are only considered constant within a ten-minute time window. For this reason the SCADA system only stores ten-minute statistics. To estimate the load transfer coefficient for each ten minutes following algorithm is applied to every ten minute interval:

1. Measurement of shell strains and bolt tension at 12.5Hz, split in 10 minute intervals,
2. Calculation of the shell force $Z$ associated with the HFB,
3. For each 10 minute interval estimation of the load transfer coefficient using a least squares solution, i.e. a curve fit of

$$F_b(t) = pZ(t) + \bar{F}_b \tag{15}$$

Figure 9: Variation of the bolt forces w.r.t. the pre-tension $F_v$ versus the variation of the shell forces. The dashed lines indicate the estimated load transfer function using a Least Squares fit of Eq. (15), the slope of which is the load transfer coefficient $p$.

(a) Results based on timeseries as shown in Fig. 8 (b) Results based on on normal operation.
where $t = 0, \ldots, 600$ s, $\bar{F}_b$ is the ten-minute mean bolt force and $p$ is the LTC.

4. Store the least squares estimate of the LTC along with turbine SCADA data for each ten minute interval.

The proposed algorithm is demonstrated in Fig. 9 for both (a) stop-start event and (b) normal operation. Note that Eq. (15) does not impose a fixed value for the pretension $F_p$ but rather estimates the variation of the bolt force around its mean value. This explains why in Fig. 9.(b) the linear curves do not pass through the origin $(0,0)$. Since long term data was collected it is then possible to evaluate the variability of LTC under different load conditions and environmental conditions.

The results of this analysis are given in Section 4.1.

3. Finite element model

Load bearing analysis have been performed at the Institute for Steel Construction of the Leibniz University Hannover. The Analysis have been performed with a finite element model generated with the software ANSYS. The objective of the finite element analysis (FEA) is to complete and verify the plausibility of the results from the measurement campaign. The adequate implementation of the measured flange imperfections in the finite element model is a challenging engineering practice. Coincidentally, the three HF-bolts lie all in a ring flange area with rather small imperfection amplitudes (see section 1.4). Thus, in the here presented FEA the authors considered only the flange imperfections resulting from the flange inclination of the MP-flange.

The entire finite element model generally consists of three components, the TP and MP flange as well as the HV-bolt. Figure 10, left, shows the general topology of the flange. To reduce model complexity only a part of the TP and MP shell was considered in the model, see Figure 10, middle. The thickness of both flanges and the shell thicknesses of both TP and MP is modelled to match the exact geometry at Nobelwind. The initial geometry depicted in Figure 10, left, of both the TP and MP was modelled without including any flange imperfections. Thus, the contact face between the TP and MP flange is perfectly horizontal. However, the MP flange is actually inclined downwards towards the MP centre point. The purpose of the inclination is to avoid collision between the anvil of the hammer and the cantilevered part of the flange during pile driving. However, a horizontal contact face with a depth of approximately 20% is left for transferring the piling forces. The inclination is introduced after meshing the model by physically moving the nodes. The inclination is considered constant along the ring flange circumference in the TP-MP flange contact face, see Figure 10, right. Furthermore, Figure 10 depicts a part of the grout skirt, which was not considered in the FEA. The grout skirt has a length of approx. 7 m. It strongly increases the model complexity and computation time. FEA with a grout-skirt length of 1 m did not show significant influence on the computed LTF. Nevertheless further, more detailed analysis (especially with the focus on the contact between MP and grout skirt) is required to quantify its full influence on the LTF.

Figure 10: Left: CAD model of the ring flange; middle: FE Ring-flange model; right: detailed view of the FE-model and the location of the modified flange contact face to account for MP-flange inclination
The ring flange model partially consists of quadratic tetrahedral and hexahedral elements, with 10 and 20 nodes per element, respectively. The meshing is performed in ANSYS Workbench, model generation however in ANSYS APDL. Since quadratic element were used, the element size of the TP shell was chosen to be 30 mm and 40 mm for the MP shell. The element sizing of the TP and MP flange was chosen to 30 mm. In the analysis nonlinear geometric effects were considered. Due to the relative small load range which is observable in the measured data, the authors neglected nonlinear material effects in FEA. The analysis have been performed with constant Young’s modulus $E = 210 \text{GPa}$. However, preload force can lead to local plastic deformation, see e.g. [14] and [22]. This effect can lead to softening of the flange stiffness and result in a steeper LTF.

The finite-element bolt representation is depicted in Figure 11. The bolt model consists of quadratic tetrahedral elements representing the bolt head and the nut. The bolt shaft is approximated with beam elements which are connected with rigid elements to the bolt head and the nut. The washers have been merged with the bolt head and the nut. Friction contact is considered between the upper and the lower flange as well as in the contact face between the bolt and the flanges. The preloading of the bolts is realised via a pre-stress element. This element is located in the middle of the bolt shank. The preload for all bolts is assumed constant, $F_p = 2439 \text{kN}$. The installed M72 have been produced with a pitch thread of 4 mm. Thus, the minor diameter of the bolt thread is larger than diameter of a M72 bolt geometry according to DASt-Guideline 021 [3]. Furthermore, each M72 bolt is equipped with a mechanical sensor, which indicates when target preload is reached. Therefore, the bolt have been preloaded up to the 70 percent of the tensile strength of the bolt material, which results in approx. $F_p = 2439 \text{kN}$.

4. Results

Section 4.1 encompasses the results govern over more than two years in the measurement campaign at Nobelwind. Section 4.2 comprises the results gained from finite element analysis.

4.1. Measurements

The monitoring campaign at Nobelwind ran from August 2017 to 1 January 2020. Data was collected almost continuously aside from some downtime of the system. The bolt at K05 was removed after two years in August 2019. The bolt was then returned for testing to confirm the bolt was still functional after two years. It was concluded that the sensitivity of the bolt was still as desired, but an offset in the mean bolt tension had developed, for more details see Appendix A. Because the conclusions in this work rely on the sensitivity of the bolt, rather than its mean tension, it was concluded the bolts have collected feasible measurements over the years. Data was collected and processed in ten minute intervals. For each ten minute interval the relevant environmental and operational conditions were collected, along with an estimate of the mean bolt tension. An estimate of the shell force $Z$ and the load transfer coefficient were simultaneously stored. The number of data-points used in the current analysis for each turbine is listed in Table 1.
Table 1: Number of data-points (10-min. interval) collected for each turbine and used in this research

<table>
<thead>
<tr>
<th>Location</th>
<th>No. records collected</th>
</tr>
</thead>
<tbody>
<tr>
<td>G08</td>
<td>94.228</td>
</tr>
<tr>
<td>G10</td>
<td>87.318</td>
</tr>
<tr>
<td>K05</td>
<td>70.697</td>
</tr>
</tbody>
</table>

4.1.1. Collected data

Over the monitoring campaign the 10-minute mean bending moments in the wind direction, just above the bolted connection, were calculated. The results of which are plotted against the mean wind speed in Fig. 12.(a). The measurements for the three turbines were very consistent, with a peak bending moment of 35MNm at 11m/s. The fact that the bending moment decreases at higher wind speeds is consistent with the controller of the wind turbine. Once the peak output power is reached, the turbine starts feathering the wind load, as a result the bending moment decreases. Beyond 25m/s the wind turbine shuts down operation due to the high wind loads. The negative bending moment at very low wind speeds is also consistent with the design of the wind turbine. As the center of gravity of the rotor-nacelle-assembly is not directly above the tower center line, a small bending moment exists that, in absence of any significant wind loading, tips the turbine slightly into the wind direction.

The shaded area in Fig. 12.(a), which reflects the 10-90th percentile, reveals that there is more uncertainty on the measurement at G10. The larger scatter in Fig. 12.(a) at higher wind speeds, e.g. exceeding 20 m/s is consistent with the fact that these wind speeds are more rare. Consequently, far less bending moment data is available for these higher wind speeds conditions, ultimately resulting in the larger spread.

On the right y-axis of Fig. 12.(a) the scale is related to the shell force $Z$. It shows that at peak load an additional $Z$ of 297.5kN is introduced due to the bending moment. At zero bending moment the shell force equals -51.6kN which corresponds to $Z_0$. Some peak measurements (top 90-th percentile) go up to 339.8kN.

Of course it strongly depends on the wind direction whether or not that peak shell force is also reached above the HF measurement bolt. This is illustrated in Fig. 12.(b) where $Z$, above the bolt, is plotted against the wind directions for all considered measurements. The figure illustrates that only when the wind direction is aligned with the bolt these peak shell forces can be reached, and then only when the wind speed reaches the peak value at 11m/s.

The results presented in Fig. 12 also provide an inside of which is the expected range of shell forces for an operational wind turbine. It is clear, that the loads observed are not sufficient to determine the entire LTF of the bolted connection. More specifically, when putting these load ranges aside the Schmidt/Neuper LTF, it can be concluded that the currently observed shell forces only exceed $Z_I$ at peak loads. This means that the vast majority of the data only covers the first segment of Schmidt/Neuper. But since the turbine spends all of its operational life in these load ranges it is essential, that the LTF is estimated correctly in this area to allow for an accurate prediction of fatigue life of the bolts.

Fig. 13 shows the histograms of both the wind direction and the shell forces at the location of the HFB. Fig. 13.(a) shows that the dominant wind direction at the site is South-West to West-South-West. The HFB of G08 and K05 both are both aligned with this dominant wind direction. With the bolt on G08 positioned on the compression side and at K05 on the tension side of said dominant wind direction. This is also reflected in Fig. 13.(b) which shows, that the majority of data for K05 is in tension and for G08 the majority of data is the compression side. At G10 the bolt is positioned perpendicular to the mean wind direction, resulting in a more symmetric distribution. For G10 there is relatively little data in the compression side, something that is also visible in Fig. 12.(b) where the compression side of the measurements is clearly less dense, wind directions between 100 and 150 degrees.

The large peak in Fig. 13.(b), present for all turbines, occurs at $Z_0$. The shell force equals $Z_0$ in absence of any wind load. Because this value is independent of the wind direction the most common shell force value is the one that corresponds with zero bending load.

4.1.2. Estimation of load transfer coefficient and transfer function

For each ten-minute interval the load transfer coefficient is calculated based on the procedure introduced in Section 2.3.2. The resulting estimates as a function of the shell force are plotted in Fig. 14.(a). Fig. 14.(a)
Figure 12: (a) Bending moment versus wind speed for the three turbines, the full line indicates the median value for each 1 m/s interval, the shaded area covers the 10 to 90th percentile within this 1 m/s interval. (b) Scatter plot of the shell forces as recorded for each bolt position versus the wind direction.

Figure 13: Histograms of (a) recorded wind directions with indication of the position of the HFB on each turbine. (b) Distribution of recorded shell forces for each turbine at the location of the HFB shows that all three turbines have LTC coefficients close to 1%. Over the entire load range the LTC for G08 is on average greater than the LTCs for both G10 and K05. Interestingly the larger LTC for G08 was already visible in the single result presented in Fig. 9.(a). In said graph the ratio between bolt force of G08 measurements was already notably steeper than for the other two turbines, hinting at a higher LTC for G08.

Some scatter appears in the area between -100 kN and 0 kN. At these values for the shell force, close to the load $\hat{Z}_0$ there is only a minimal load acting on the turbine and stress variations of both the shell force and the bolt tension are small. These small variations lead to a larger uncertainty when estimating the load transfer coefficient, ultimately resulting in the larger scatter around that particular area.

In Fig. 14.(b) the figure is binned for fixed ranges in shell force. The central lines, which reflect the median value in each bin, confirm that the LTC for G08 exceeds respectively G10 and K05. The shaded area, which again reflects the 10-90th percentiles, indicates that the majority of data is closely spaced around the median values.

For G08 and G10 it can be seen that the LTC slightly increases for higher tension loads. For lower loads the LTC is relatively stable. The results also show a near constant and non-zero LTC when the bolt is situated on the compression side ($Z < 0$). An example of the behaviour of a bolt on the compression side of the shell, and the non-zero LTC, was also shown in Fig. 9.(a-b) where at both timestamps the bolt of G08 was situated...
Since the LTC is the derivative of the LTF we can also compose an empirical LTF from the monitoring data. To do so, a regression model was fitted to the collected LTC data. As the behaviour of the LTC was not complicated, a quadratic polynomial was sufficient to capture the main trend of the data within the considered range. It is a simple step to integrate the found expressions of the LTC to obtain a load transfer function. For ease of reference the as-designed bolt tension was used as the offset of the LTF. In Section 5.1 the empirical LTF are compared to the modeled curves. However, the results should not be used outside the data range used to fit them.

4.1.3. Temporal evolution

As the bolts were installed over a period of two years it is possible to analyse the behaviour of the load transfer coefficient over time. This question was in particular motivated by the observed variations in the mean bolt tension shown in Fig. 15.(a). The observed mean bolt tension of the bolts at K05 and G10 had decreased by over 10% during the monitoring period. Surprisingly, an increase in the bolt tension of G08 was observed. In both cases the variations in bolt tension are seemingly related with the seasons, suggesting some interaction with the ambient temperature.

While the variations in the mean bolt tension were considerable, there seems no direct impact on the LTC which for all turbines remained stable during the monitoring period as can be seen in Fig. 15. At the moment it is not clear what mechanism is exactly driving the variations in the mean bolt tensions. It was considered that the strong variations in the mean bolt tension could be related with a defect of the HFB. This motivated the removal of the K05 HFB at the end of the monitoring campaign to allow for final testing and assess the correct function of the measurement bolt. These tests are described in Appendix A, and identified an offset of 266.6 kN, which is of the same order as the observed decrease of the K05 mean bolt tension. Meanwhile, the measurement sensitivity for variation in the bolt tension remained acceptable. Given the stable observations of the LTC and the outcome of the testing, it was concluded that the long term behaviour of the mean bolt tension is at best unreliable and was not further considered in this research.
Figure 15: (a) Mean bolt tension and (b) Estimated LTC over the entire monitoring period. While the mean bolt tension varied significantly it was later found more plausible that at least part of these variations were caused by the sensor drifting. Any gaps in the data are caused by either the unavailability of the bolt measurements or the strain gauges above the flange.

4.2. Finite Element Analysis

Finite element analysis have been performed with two types of the ring flange models to illustrate the influence of the influence of flange imperfections on the LTF. The “inclined” model considers the flange inclination of the MP flange towards the MP centre (Figure 10), right. In the “non-incl.” model the MP flange is perfectly flat.

The strain sensor is located approximately 35cm above the bolt-flange contact face. At this height, the preload of the bolts causes bending stresses in the shell. Depending on the flange imperfections, these bending stresses can vary strongly. Figure 16 depicts the computed residual stress at sensor location for the inclined and non-inclined flange. The increase of stresses in the shell at sensor location with the non-inclined flange is characterised by linear function. At maximum preload the stress at sensor location is approximately 3.7MPa. For the inclined MP flange the stress increase is characterised approx. by a bi-linear function. Up to approx. 19% of the preload the bending stress rapidly increases. At this preload level the gap resulting from the inclination is closed. Further preload leads to an increase of stresses with the same gradient as in the non-inclined flange. Eventually, Figure 16 shows that apart from the stress due to self-weight of the structure and the turbine, the shell at the sensor location is also affected by residual stresses from the bolt preload.

Figure 17 shows the stresses due to the bolt preload in the global vertical Z-axis of the model with the inclined MP flange. Bending stresses arise directly above the TP flange. Less stress appears in the stiffer MP shell due to larger shell thickness, 60 mm and 80 mm, respectively. In accordance with the results from Jakubowski and Schmidt [11], Figure 17 additionally shows that the clamp solid is located eccentrically, closer towards the outer side of the flange connection. As mentioned in the Section 1.4, this has a positive effect on the load transfer behaviour.

The computed LTF for the ring flange connection with and without the inclined MP-flange is depicted in Figure 18. Contrary to the classical depiction, the abscissa shows the stresses instead of the segment forces. Eventually, Figure 18 shows the load transfer behaviour as a function of stresses evaluated at the sensor location and as a function of stresses in the shell top, arising from the bending moment. Notably, the LTF of the connection with inclined MP flange has a much lower gradient than the LTF of the perfectly horizontal MP flange. This expectedly results from the eccentric clamp solid. The LTF of the connection...
Figure 16: Increase of the stresses at sensor location as a function of preload for the inclined and non-inclined flange; gap closes when approx. 19 percent of the preload is reached.

\[ F_b = 463.4 = 0.19 F_p \]
\[ \sigma = 17.2 \]
\[ \sigma_{res} = 21.0 \]

Figure 17: Vertical stresses in Z-direction due the preload of the bolts in the ring flange model with the inclined MP-flange.
with the inclined MP flange can roughly be approximated by a linear function. Furthermore, in the range of
the measured forces, the shape of the LTF as a function of the stresses at the sensor location and the shape
of the LTF as function of the applied stresses are almost congruent. The difference lies in the shift along the
abscissa resulting from the residual stress introduced by bolt preloading.

The ratio between the applied stress and the stress at the sensor location is depicted in Figure 19. In this
Figure, the stresses at sensor location are shifted by the size of the residual stress towards zero. The ratio
can be characterised by a linear function with a gradient of approximately 1.08.

Eventually, the results of the FEA with an inclined MP-flange show that due to the inclination
1. bending stresses are introduced into the shell and
2. the clamp solid - generated by the preload - is located closer towards the TP and MP shell.

This results in a stiffer ring flange connection compared to the connection without the inclined MP-flange.
Figure 20 shows the LTF from the finite element analysis and the LTF according to Schmidt/Neuper with
\( p = 0.21 \) \( (LTC_{Sk,N} = 0.21) \). In the load range from \(-40 \, MPa\) to \(+40 \, MPa\) the FEM LTF can be approximated
by a linear function with a constant load transfer coefficient \( LTC_{Z>0} = 0.0047 \) and \( LTC_{Z<0} = 0.0035 \),
for the tension and compression loads, respectively. Finally, the load transfer behaviour from the FEA is
approximately equivalent to the measured results, shown in chapter 4.1. A detailed comparison of finite
element and measured results follows in chapter 5.1.
5. Discussion

5.1. Comparison between empirical and modelled load transfer function

The comparison of the LTF derived from the measurements and a numerically computed LTF using the finite element method is a challenge due to the numerous parameters that influence the load transfer behaviour of a ring flange connection, see Section 1.2. As a consequence the comparison between a real-world flange and the results obtained from a FE model should be performed with caution.

A first element that was considered is the effect the bolted flange could have on the measurements themselves. The strain gauges were installed at a height between 35 and 45 cm above the bolted flange. The FEM-results show that at these distances a stress concentration still occurs towards the inside wall of the shell. This stress concentration occurs as the stress transitions from the shell into the clamp solid generated by the bolted connection. With the incorporation of the MP-flange inclination into the FE model, the clamp solid moves more to the outside of the flange, which slightly reduces this effect. For the inclined MP flange the bolt tension loads, that close the flange, induce additional stresses inside the shell, which further complicate the stress patterns in this area. Using the FE model, including the MP-flange inclination, it was attempted to estimate the impact of both effects on the measurements. Evaluated at a height of 35 cm above the flange, it was derived from the FE model that the load estimated from a strain gauge on the inner wall overestimates the actual load by 8%. As such the LTCs from measurements as derived to this part are actually an underestimation of the actual LTCs.

Note that this correction factor only accounts for the flange imperfection resulting from the MP-flange inclination. Hence, the factor likely varies along the flange circumference as a function of the given flange imperfections. Nevertheless, the authors assume that decreasing the measured shell force by 8% does result in a more adequate approximation of the external segment force. It is equivalent to increase the measured LTCs by 8% in order to compare the LTCs obtained from measurement to those from the FEM model.

Table 2 shows the computed load transfer coefficient with FEM of the LTF evaluated at sensor location (LTC*) and the coefficient for the LTF derived with the actually applied load (LTC). The LTC slightly varies for the compression loads and the tension loads. The ratio between both LTCs is equivalent with the correction factor 1.08. The same factor is being used to compute the LTC for the three locations G08, G10 and K05 based on the LTC* derived from the measured data.

Figure 21 (a) shows a comparison of the empirically derived LTF – as a linear regression of the measured data including the correction factor – and the computed LTF with the inclined MP-flange using FEA. The LTF at location G08 is the steepest function, followed by the LTF at location G10 and K05. The steeper the function the softer the connection. Generally, the measured LTF’s are characterised by a comparable LTC. This corresponds with the given boundary conditions. The bolts at the three location have similar preload level close to 2439 kN. Furthermore – and more important – at all three locations the HF-bolts lie in a
Table 2: LTC* evaluated at the sensor location (35 cm above the flange) and LTC for the LTF derived with actually applied loads, both derived from the FE model. The factor that relates both is approximately 1.08.

<table>
<thead>
<tr>
<th></th>
<th>Tension</th>
<th>Compression</th>
<th>Tension</th>
<th>Compression</th>
</tr>
</thead>
<tbody>
<tr>
<td>LTC</td>
<td>0.0044</td>
<td>0.0032</td>
<td>0.0047</td>
<td>0.0035</td>
</tr>
</tbody>
</table>

Figure 21: (a) Comparison of approximated empirical LTFs for the location G08, G10 and K05 (coloured lines), the LTF derived with the FE-Model for the inclined and non-inclined flange as well as the LTF according to SN; (b) Corrected Load-transfer coefficients from the measurements alongside the estimated LTC from the FE model.

ring flange sector with rather small imperfection amplitudes $0.23 \text{mm} < k_o < 0.33 \text{mm}$ at the outer flange edge. However, all three location share the imperfection resulting from the inclined MP flange. The LTF derived with the presented finite element, which accounts for the inclined MP flange, is characterised by a similar linear shape compared to the measured results. Since no further imperfections are considered in the model, the LTF is stiffer and therefore is characterised by a lower LTC. Figure 21 (b) shows that the LTC of the computed LTF lies in the range of the LTC derived from the measured data. On the global scale the LTF shows a linear behaviour. However, on a local scale the function is not linear, which results from model deficiency.

Figure 21 (a) shows a comparison of the empirically derived LTF – as a linear regression of the measured data including the correction factor – and the computed LTF with the inclined as well as non-inclined MP-flange using FEA. The LTF at location G08 is the steepest function, followed by the LTF at location G10 and K05. The steeper the function, the softer is the connection. Generally, the measured LTFs are characterised by a comparable LTC, Figure 21 (b). This corresponds with the given boundary conditions. The bolts at the three locations have similar pre-load level close to 2439 kN. Furthermore – and more important – at all three location the HF-bolts lie in a ring flange sector with rather small imperfection amplitudes $k_o$ at the outer flange edge. However, all three locations share the imperfection resulting from the inclined MP flange. The LTF derived with the presented finite element, which accounts for the inclined MP flange, is characterised by a similar linear shape compared to the measured results. Since no further imperfections are considered in the model, the LTF is stiffer and therefore is characterised by a lower LTC than the values found from measurements, which is reasonable. Figure 21 (b) shows that the LTC* of the computed LTF lies in the range of the LTC* derived from the measured data. On the global scale the computed LTF shows a linear behaviour. On a local scale the function is however not linear. The nonlinearity results from model deficiency. Thus, the results scatter noticeable. Eventually, the results from the FEA correspond adequately with the measured data.
5.2. Future work

In the undertaken measurement campaign, the HF-bolts are located in a flange position with rather small imperfection amplitudes on the outside of the flange \( k_o \). Thus, the measured load transfer behaviour is consequently characterised by a low LTC (see Figure 14). The current results thus may not be representative for flanges with larger imperfections. The current model includes the inclined MP flange only. Thus, it is reasonable to consider the flange imperfections as measured in future finite element analysis.

6. Conclusions

The objective of this paper is the validation of the measured load-transfer behaviour of ring flanges with large high strength bolts in offshore-wind support structures from continuous field measurements. For this reason three turbines at the Nobelwind offshore wind farm were instrumented with an extensive measurement setup. The setup comprised of 6 strain gauges installed approximately 35cm above the bolted flange connection between the transition piece and the monopile. In addition each wind turbine was equipped with a high-frequency measurement bolt. The instrumentation at Nobelwind makes it possible to measure both the stresses in the shell and the resulting stresses in the bolt. Initial analysis of the timeseries reveals that stress variations, e.g. due to normal oscillations of the structure, are transferred into the bolt. Long-term data was collected from August 2017 until the end of the project on January 1st, 2020. A methodology was developed to estimate the load transfer coefficient LTC for each ten-minute interval, resulting in a total of over 250,000 datapoints being collected. The measured data revealed a very stiff flange connection with an LTC way below the LTC of Schmidt/Neuper. Next a Finite Element model was developed to represent the bolted flange at Nobelwind. Results from FEA revealed that the MP-flange inclination significantly influences the location of the clamp solid and thus the stiffness of the connection.

Finally, the results from FEM and measurements were compared. The results show that it was necessary to include the inclined MP flange in the model to match the results seen in the measurements. These results from the FE model confirm the measured values of the LTC. The FE model suggests even lower LTC values than those observed from the measurements, which is consistent as the model did not consider any other imperfections aside from the inclined MP-flange.

Acknowledgements

The current research was conducted in the frame of the O&O Nobelwind, a research project with the support of the agency Flanders Innovation & Entrepreneurship. Wout Weijtjens is a Post-doctoral researcher at the Research Foundation Flanders (FWO). At the Institute for Steel Construction, the research work was conducted within the ForWind joint research project ”ventus efficiens – Joint research for the efficiency of wind energy converters within the energy supply system”. This project is financially supported by the Ministry for Science and Culture in Lower Saxony, Germany. The financial support and cooperation is kindly acknowledged.

The current paper is an equal effort of Vrije Universiteit Brussels (VUB) and Leibniz University Hannover (LUH). In which the VUB executed and processed the long measurement campaign, LUH developed the FE-Models and derived the corresponding load transfer functions.

The authors also want to show their gratitude to the people of Parkwind and Nobelwind for their support during this project.

Bibliography


Appendix A. Post-campaign check of the measurement bolt

In August 2019 the HF measurement bolt installed on K05 was removed from the wind turbine. The bolt accuracy was re-evaluated in November 2019 at Labo Soete at the university of Ghent. As shown in Fig. A.22.a the bolt was installed in a test bench which was set up to apply a tension load of up to 2500 kN. The loads measured by the bolt were compared to the load recorded by the test equipment and are shown in Fig. A.22.b.

The results in Fig. A.22.b suggest that the strain gauge inside bolt has developed an offset of approximately 266 kN over the monitoring period. The origin of this offset, and the mechanism behind it, are currently unknown. However, for this research the main concern was that the sensitivity of the bolt had not altered over time. When correcting for the offset a sensitivity, i.e. the ratio between the loads measured by the test-bench and those recorded by the HFB, of 98.5% was found. Indicating that while the bolt had developed an offset, it was still able to accurately monitor the variations of strain inside the bolt. Combined with the stable results over the two-year monitoring campaign, it was concluded that the HF bolts had operated as expected.
Figure A.22: (a) The HF measurement bolt of K05 was removed in August 2019 and tested at the University of Ghent in November 2019. (b) The measurements of the HFB (labeled: SG Bolt) were compared with those obtained from the load cell inside the test bench (labeled: Test bench). After correction of the offset (labeled: SG bolt corrected), results conclude that the sensitivity of the bolt to variations in tension had not deteriorated over the two year monitoring campaign.