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Experimental estimation of the residual fatigue life of in-service wind turbine bolts

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ABSTRACT

This study presents an experimental methodology aimed at estimating the residual fatigue life of in-service wind turbine bolts. The main objective is to assess the residual life of the bolts to plan their replacement and to avoid unexpected breakages of wind turbine blade connections. To develop the methodology, M16 bolts of quality 10.9 with controlled predamage were used, simulating in-service operating conditions. The fatigue tests were carried out taking care to place the nut at the point on the bolt that produces the highest damage at the same point where the predamage was performed. In addition, the influence of a possible angular positioning error on the residual fatigue life has been investigated. The residual fatigue life is estimated from the difference in fatigue life of new bolt tests and the fatigue life of predamaged bolt tests, simulating service conditions. Special care has been taken to guarantee that the most damaged zone of the bolt in service is also in the position that produces the highest damage during tests. An experimental procedure for determining the fatigue life of a new bolt from tests conducted on a bolt under the same operating conditions was developed. The developed methodology has been applied to M20 bolts belonging to real turbines in service.

1. Introduction

Bolted joint elements in wind turbines are subject to both static and cyclic loads, which frequently results in fatigue failure of the connecting bolts. The magnitude and direction of these loads depend on several scattering parameters, such as wind loads, bolt preload and geometric imperfections, so they are difficult to predict. It is usual to perform measurement campaigns that statistically describe the wind forces and their effect on the bolted connection loads. Due to load variation and randomness, fatigue failure of bolts in ring–flange blade connections or in flange structural connections are frequent causes of failure in wind turbines, and there are many studies to improve the methods for estimating fatigue strength in wind turbine bolted connections. Chou and Tu [1] reviewed a total of 62 cases of wind turbine tower accidents worldwide in the period 1997-2009; 35% of these accidents were blade failures, which also impacted the structures. Among the 44 cases with a known origin, there were three cases of fatigue failure and three cases caused by earlier blade failure, which subsequently impacted the structure. Fig. 1 shows a case of failure of one ring–flange blade connection in northern Spain, where the blade fell down. Some fractured bolts are also shown in the figure.

There are a number of experimental, analytical and numerical studies devoted to improving the knowledge of the fatigue behaviour of wind turbine bolted connections. For instance, Schaumann et al. [2] were able to assess the probability of failure using Monte Carlo simulation of bolt sets for ring–flange connections, subsequently validating their results by fatigue testing. Schaumann

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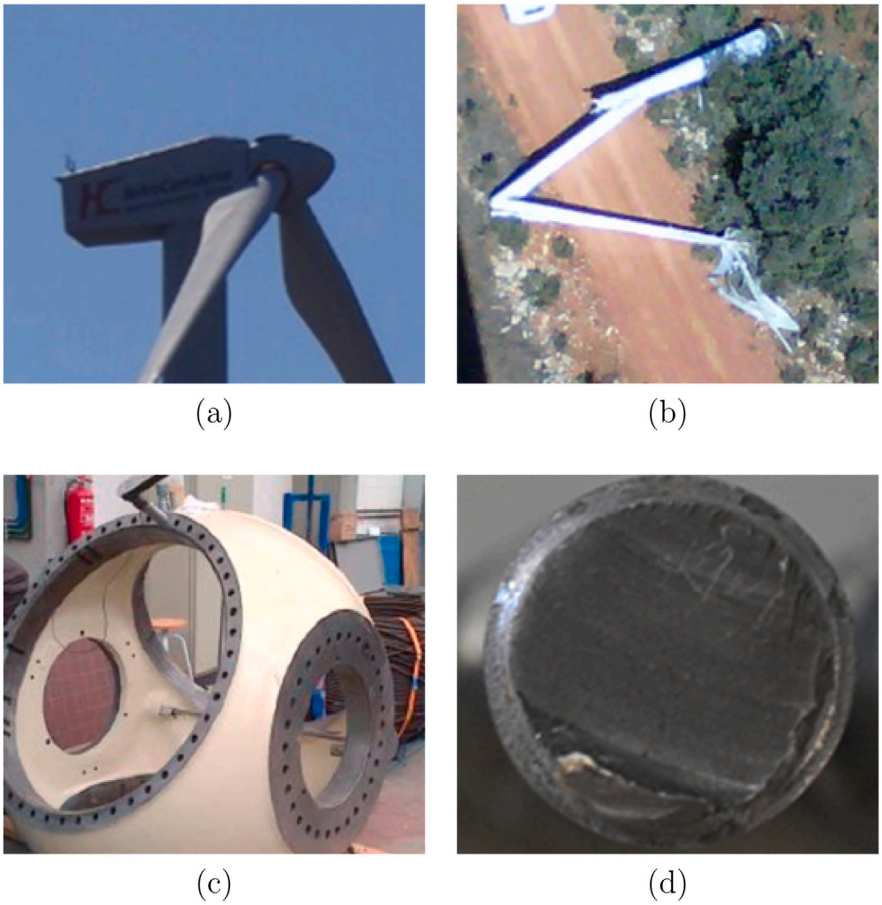

Fig. 1. (a) Wind turbine with broken blade; (b) broken blade; (c) joint blade-housing; and (d) fatigue section of a failed bolt.

and Eichstädt [3] presented experimental results of fatigue tests on M36 high-strength bolts, quantifying the effect of different zinc coatings and validating relevant design $S-N$ curves. Yun et al. [4] investigated the fatigue failure of 6 bolts belonging to a wind turbine hub by conducting morphology and chemical composition analyses, mechanical and microstructure testing and fracture microarea analysis. Yu et al. [5] were able to assess the fatigue life of the flange connecting bolts of a wind turbine tower by using finite element modelling, the rainflow counting algorithm and a cumulative damage criterion. Fu et al. [6] proposed a reliability study of a 1.5 MW wind turbine by determining a probability density function of fatigue damage of the flange and the bolts by using a rainflow counting method. Based on continuum damage mechanics, Liu et al. [7] proposed a modelling method for the cumulative fatigue damage assessment of metallic bolted joints. In many other applications, such as grid structures, the fatigue damage of connecting bolts is a matter of research [8]. In addition, welded joints connecting the flange to the tower shell may have to be assessed as a whole with the connecting bolts [9]. Regarding slip-resistant bolted connections, a comprehensive review of the main scientific studies of fatigue strength and performance of shear bolted joints can be found in [10].

Most of the available studies on fatigue of bolted connections in wind turbines focus on offshore applications. A complete review of the main issues involved in fatigue life assessment and the different factors that influence the fatigue performance of monopile bolted connections (used in offshore wind turbines to connect the monopile to the transition piece) was presented by Lochan et al. [11]. Additionally, for offshore wind turbine monopile foundations, Madsen et al. [12] investigated the fatigue life by numerical simulation. A review of the knowledge regarding the influence of manufacturing and surface conditions on the fatigue strength of bolts in assemblies with large diameters is presented in [13,14], including results from 20 different fatigue test series conducted on three different high-strength bolt assembly systems with varying manufacturing conditions. Offshore wind turbine bolted connections may have their own problems, such as the combination of seawater ageing and fatigue loading [15]. An overview of support structure types, the character of dynamic loads and the fatigue resistance in offshore wind turbines is given by Schaumann et al. [16]. The transfer of the shell forces into the bolt has a great impact on their predicted fatigue lifetime. Among the available load transfer functions, the one by Schmidt/Neuper is the most extended, which was compared with an empirical load transfer function devised by in situ monitoring of instrumented bolts by Weijtjens [17].

Fatigue failure in wind turbine blades is also a subject of study where many contributions can be found. Fatigue damage in blades usually occurs at the blade root's bolted connection [18]. The fatigue life of a horizontal axis wind turbine blade has been predicted considering a stochastic approach [19]. Kong et al. [20] described the structural design of a composite wind turbine blade considering fatigue life. K. Ha [21] used 2D and 3D finite element models to determine the stress concentration factors in bolt threads both between an insert and an M42 bolt used for a large offshore blade and between an M42 bolt and a nut. K. Haciefendioglu [22] studied the dynamic seismic response of an offshore wind turbine, including the fluid–structure–soil interaction, while they did not consider the fatigue life of the structural elements. Alonso-Martinez et al. [23] studied, mainly by numerical simulation but also through experimentation, the causes of the collapse of a wind turbine tower at the connecting flange between the tower and its base. The experimental assessment of the fatigue strength using large-scale test setups such as large HV bolts and ring–flanges was investigated by Schaumann et al. [24,25].

When analysing fatigue failure in bolted connections, it is of fundamental importance to calculate the stiffness of the members to estimate how the loads are distributed among the bolts and the clamped surfaces. A simplified approach for the calculus of member stiffnesses in different types of bolted connections can be found in Wileman et al. [26]. The load state of bolts is usually complex, including traction, shear and bending. Fatigue due to bending loads has been appointed as a failure cause in a megawatt-class wind turbine blade root bolt [27]. Wang et al. [28] suggested the importance of shear fatigue for a higher accuracy of fatigue design. According to Badrkhani, Ajaei and Soyoz [29], who studied the effect of the bolt preload level on the fatigue damage in eccentrically loaded ring–flange connections, bending stresses have an important influence on the fatigue life of bolts belonging to wind turbine tower flange connections. With respect to the joined flanged plates, the fatigue behaviour is improved by the presence of tightened bolts. Chakherlou et al. [30] and Mínguez and Vogwell [31] studied both by experimental and numerical means the positive effect of the clamping force in bolted plates, which is expected to be due to the reduction of the self-loosening tendency and to the decrease of the shear load on the bolts. According to their results, the fatigue life increases due to the compressive stresses around the plate hole, and this increase is more significant in the high cycle regime. Nevertheless, due to the influence of the stress ratio R on the fatigue $S-N$ field as it is reflected in the crack growth rate curve, a too large preload is detrimental for the bolt fatigue life [32].

Despite the number of analyses and the efforts dedicated to improving the capability for fatigue assessment of bolted connections in wind turbines, there are still failures of these connections in wind farms. Considering the number of parameters affecting the fatigue behaviour and their randomness, predictive maintenance becomes mandatory. When fatigue failure occurs, it is very difficult to know the failure origin. Therefore, it is difficult to know whether it was due to a general underestimation of the loads produced or an inadequate design decision, something that might require a modification of the design and, for instance, changing the bolts of that connection in all the turbines of the farm or, at least, installing new bolts of the same size and quality to avoid further failure of bolts that may be dangerously damaged and establishing an overhaul program. However, it may be the result of the concentration of wind loads in the area of the turbine, which failed due to its position in the group of turbines. In this case, any decision should concentrate just on that turbine. Additionally, the failure could be a consequence of any kind of geometric imperfection of a particular flange or ring, which would only affect that connection.

Thinking, for instance, about a blade-ring connection failure, the decision to change all bolts in the wind farm, the bolts of a turbine blade connection or just those of one connection will depend on the failure origin and will have very different economic consequences. According to this, it is of paramount importance to know whether the fatigue-damaged bolts are in general throughout the whole farm/park, only of one turbine or only those of one connection. The knowledge of the level of damage is determinant to estimate the residual fatigue life of those bolts. Therefore, the objective of this paper is to present a methodology to predict the residual life of in-service wind turbine bolts. A case study on the analysis of the maintenance costs for the monopile and the transition piece bolted connections can be found in [33], supporting the importance of inspection results.

Knowledge gap. There is no experimental methodology in the scientific literature to predict the residual life of in-service bolts of wind turbine blades to avoid unexpected catastrophes. To this end, a step-by-step experimental procedure was developed to determine the residual life of a group of predamaged bolts. This procedure has been developed with new M16 bolts and subsequently tested with M20 bolts belonging to an in-service turbine where one of its blades unexpectedly fell. During the study, the importance of the relative position of the bolt–nut pair with respect to its original position in the turbine and its influence on the residual fatigue life is demonstrated. Since it is not possible to know the fatigue life of a new bolt in service turbines if some unused bolts are not kept, this work explains how to obtain the fatigue life of a new bolt from a used or predamaged bolt. The results of this work are useful to carry out predictive maintenance in wind farms by having an estimation of the residual life of bolts in service, which may have been damaged for certain reasons. Future research will focus on the probabilistic estimation of the bolt's damage including also the consideration of their relative position in the flange to achieve more approximate predictions of their useful life. This way the results of this work are expected to have an impact on increasing safety during operation of wind farms.

This article is structured as follows. Section 2, the methodology used to estimate the residual fatigue life of wind turbine blade connection bolts in service as well as details of the experimental setup and testing program are shown. Section 3 describes the procedure designed for determining the fatigue life of a new bolt from experimental tests on a damaged bolt. Additionally, the application of the methodology to the real case of in-service M20 bolts is described. Finally, the conclusions of the study are shown, as well as the potential of the developed procedure.

2. Materials and methods

This section explains the methodology used to estimate the residual fatigue life of wind turbine bolts in service. The bolts studied belong to a wind turbine in service in northern Spain (EU). In this wind turbine, fatigue failure occurred in one of the blades, as shown in Fig. 1b. To reproduce the operating conditions of the tightening torque of a bolt inside the joint, a device was designed and

Table 1
Material properties of 10.9 quality bolts, where S_y(N/mm^2) is the 0.2-yield stress; S_{ut} (N/mm^2) is the ultimate tensile strength; A_s(mm^2) is the cross-section; p(mm) is the thread pitch; d_2(mm) is the average diameter of the bolt thread; d_3(mm) is the core diameter of the bolt thread; D_e(mm) is the maximum diameter of the friction surface of the nut; D_i(mm) is the minimum diameter of the friction surface of the nut. The values are obtained from the ISO-965-1-2 standard and checked in the specimens.

Bolt	S_y (MPa)	S_{ut} (MPa)	A_s (mm^2)	p (mm)	d_2 (mm)	d_3 (mm)	D_e (mm)	D_i (mm)
M16 × 120	900	1000	150	2	14.70	13.84	17.70	22.50
M20 × 140	900	1000	235	2.5	18.38	17.29	22.40	28.19

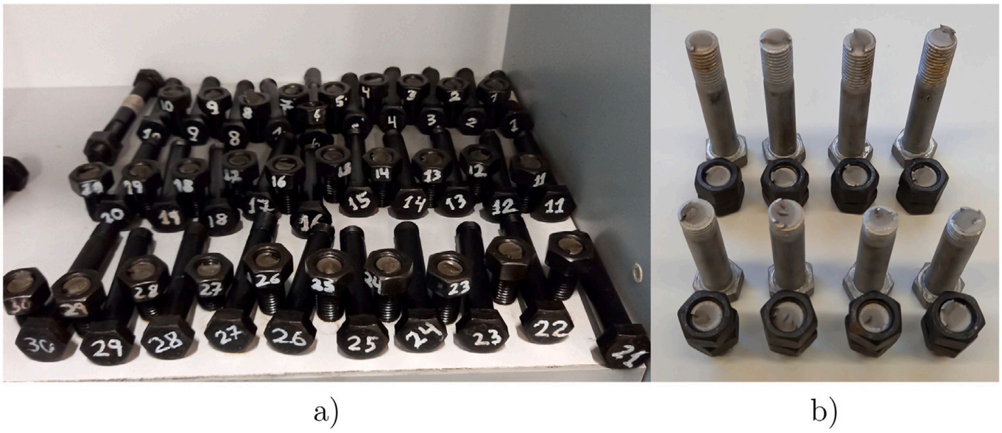

Fig. 2. (a) M16x120 bolts used to develop the methodology; (b) M20x140 bolts belonging to a wind turbine in service for the application of the methodology to a real case.

instrumented to measure the compressive force caused by the tightening torque, allowing the bolt traction force measurement. In what follows, the methodology applied to estimate the residual fatigue life of bolts is described. Additionally, to test the capabilities of the procedure to estimate the residual life, its application to the estimation of the residual lives of M16 new commercial bolts is presented.

2.1. Specimen description

To test the procedure, it was applied first to commercial M16, and the same procedure was applied later to 8 units of M20 bolts taken from one of the failed blades of a wind turbine located in northern Spain, as mentioned above. One of its blades fell down due to fatigue of the bolts of the ring–flange connection, as shown in Fig. 1. This figure also shows the hub, where the blades and main shaft are connected, and one of the failed bolts.

As mentioned above, two types of bolts have been used in this work: on the one hand, new commercial bolts that have not been in service, M16 with a length of 120 mm and quality 10.9 (Fig. 2a), with the aim of experimentally demonstrating the methodology developed in this work; and on the other hand, M20 bolts with a length of 140 mm and quality 10.9 (Fig. 2b), which belong to a wind turbine in service in which a fatigue failure occurred in one of the blades, are used as a real case application of the developed methodology. The geometrical and mechanical properties of both bolts are shown in Table 1.

2.2. Testing devices

The machine used to perform the fatigue tests is a servo-hydraulic push–pull INSTRON 8033, with a load capacity of 500 kN. To ensure a reliable clamping system, two clamping heads were designed where the bolted joint was installed. The clamping and fatigue testing machine is shown in Fig. 3. To simulate the operating conditions, a device is designed consisting of two threaded heads that are coupled to the clamping system of the machine and which compress a cylindrical bushing when the tightening torque is applied, as shown in Fig. 3b. In addition, this bushing has been instrumented with strain gauges to measure the preload provided by the tightening torque, which should be similar to that of the bolt in service. The bushing is a hollow cylinder made out of steel, having an inner diameter of 44 mm, an outer diameter of 47 mm and a height of 44 mm. Before performing the tests, the bushing was calibrated in the test machine. Bolts were tightened with a COMPUTORQ Model 2503CF-II torque spanner, with a working range of 25–320 Nm.

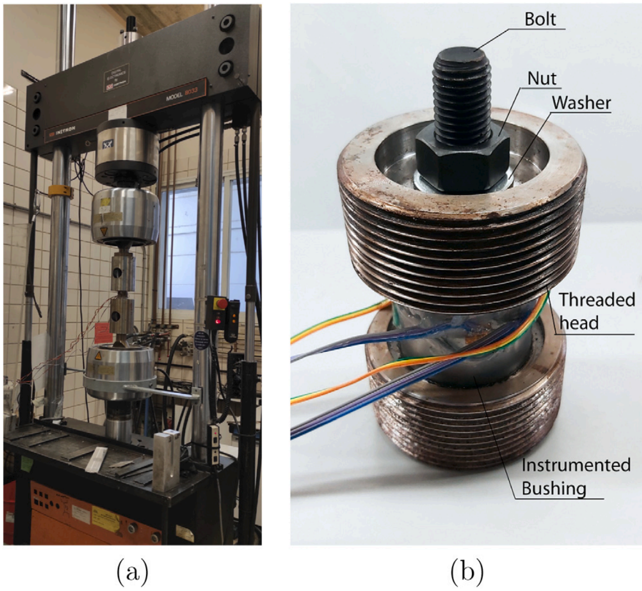

Fig. 3. (a) Fatigue machine INSTRON 8033; (b) Preload device, consisting of one instrumented bushing and two threaded heads for clamping.

Table 2
Tightening torque, bolt preload used in the tests, and estimated friction coefficient.

Bolt	M_0 (Nm)	F_0 (N)	μ
M16 × 120	205.21	93 957	0.101
M20 × 140	403.12	147 199	0.099

2.3. Tightening torque and preload

In the wind turbine bolted blade-housing connection, the bolts are tightened to a specific torque using a torque wrench. This tightening torque causes a traction force or preload, F_0, in the bolt. Such preload is maintained approximately constant during the life of the bolt. The tightening torque, M_0 (Nm), can be estimated using ISO-16047:2005 standard [34] for a preload force, F_0 (N), which produces an equivalent stress that is a percentage of the bolt yielding stress. The friction coefficient, μ, as well as p, d_2, D_i and D_e used in the calculations are defined in Table 1. As an example, Table 2 shows the torque and theoretical preload that would induce an equivalent stress of 80% of the yield stress to the M16 and M20 bolts studied in this work. The preload, F_0 (N), induced by the application of the tightening torque is measured with the help of the strain gauges installed in the bushing.

To check the tightening procedure, the torque to bolt load ratio, M_0/F_0, was obtained experimentally for $\Delta F_M = 0$ by measuring the applied torque with a calibrated torque wrench and the compressive load induced on the instrumented bushing (see Fig. 3b). This way, it is also possible to estimate the friction coefficient between the bolt and the nut. Table 2 shows the torques and bolt loads applied on the M20 and M16 bolts, as well as the estimated friction coefficient.

2.4. Bolt and bushing stiffness ratio

Testing a bolt with a preload F_0, as represented in Fig. 4b, means that the load applied by the testing machine, ΔF_M, Fig. 5, will reduce the bushing compression in ΔF_C and will increase the bolt load in ΔF_B in such a way that:

$$\Delta F_M = \Delta F_C + \Delta F_B \tag{1}$$

where the load increments ΔF_C and ΔF_B are proportional to the respective stiffnesses K_C and K_B, respectively (Fig. 4):

$$\frac{\Delta F_B}{\Delta F_C} = \frac{K_B}{K_C} \tag{2}$$

Therefore, to determine the actual load on the bolt, F_B, when the load ΔF_M is applied by the machine, the stiffness or the load ratio needs to be obtained experimentally before testing or continuously measured with the strain gauges during tests. During the design of the testing devices, in particular during the design of the bushing, the stiffnesses K_C and K_B were estimated following the recommendations of the VDI 2230 standard [35]. This way, the geometry of the bushing could be selected so that its stiffness was not too large as compared to that of the bolt. Nevertheless, the exact values of these stiffnesses are not needed in this research since

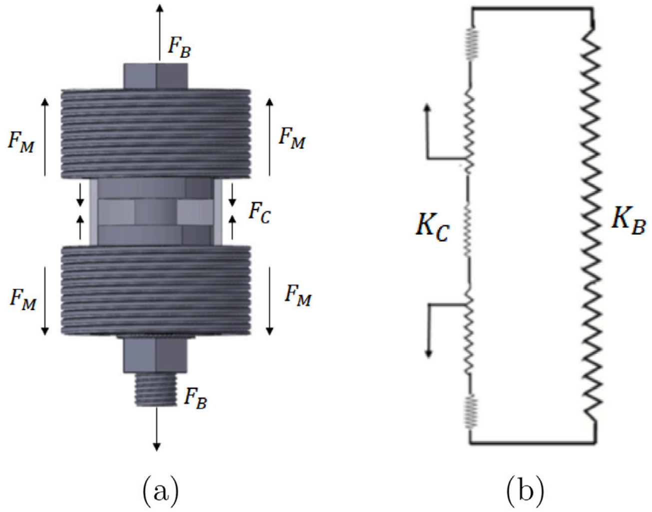

Fig. 4. (a) Distribution of forces on the bushing and bolt of the torque device; (b) Stiffness model on the bushing and bolt of the torque device.

the increment of load on the bolt, ΔF_B, can be deduced from Eq. (1) from the knowledge of the increment of load applied by the machine, ΔF_M, and the decrement of load experienced by the instrumented bushing, ΔF_C, which was previously calibrated. Fig. 5 shows the load distribution obtained for the M16 bolts when an initial preload, F_0 = 103 kN, is applied to the bolt, such that:

$$F_B = F_0 + \Delta F_B \tag{3}$$

$$F_C = F_0 - \Delta F_C \tag{4}$$

with a ratio:

$$\frac{\Delta F_C}{\Delta F_M} = 0.76 \tag{5}$$

For the M20 bolt system, the ratio was

$$\frac{\Delta F_C}{\Delta F_M} = 0.84 \tag{6}$$

2.5. Analysis methodology

Roughly, the experimental process to estimate the residual life of bolts can be described as follows.

1. New or undamaged bolts should be fatigue tested to approach the $S-N$ curve or a part of it corresponding to lives between 10^5 and 10^6 cycles. A stress ratio should be defined before starting tests. A value near the one produced under service loads is preferable.
2. From the $S-N$ curve, a load, P, producing a total life corresponding to high cycle fatigue, but not too high to make the tests very time-consuming, should be selected. For instance, the load corresponding to a fatigue life N = 500,000 cycles.
3. A number of bolts, N_b, from the most loaded zone of the flange should be chosen. The number N_b should be between 4 and 8.
4. Mounting each of the selected bolts in a testing rig with the first teeth of the thread in contact with the nut in the same position as in the flange–ring connection. Mounting it in a different position will produce the highest damage during testing in a different position than in situ, providing unacceptable results.
5. Testing each bolt under the load previously defined until failure after a number of cycles n_j.
6. Estimate the residual life of the bolt, r_j, which represents the residual life relative to the total expected life as $r_j = n_j/\mathrm{N}$. The estimated residual life, r, for the group of N_b bolts will be the average of the individual residual lives:

$$r = \frac{1}{N_b}\sum_{j=1}^{N_b} r_j \tag{7}$$

As will be shown later, there are some details that should be specially considered during tests to obtain useful results, such as the preload applied to the specimen to be tested and the nut position in contact with the bolt.

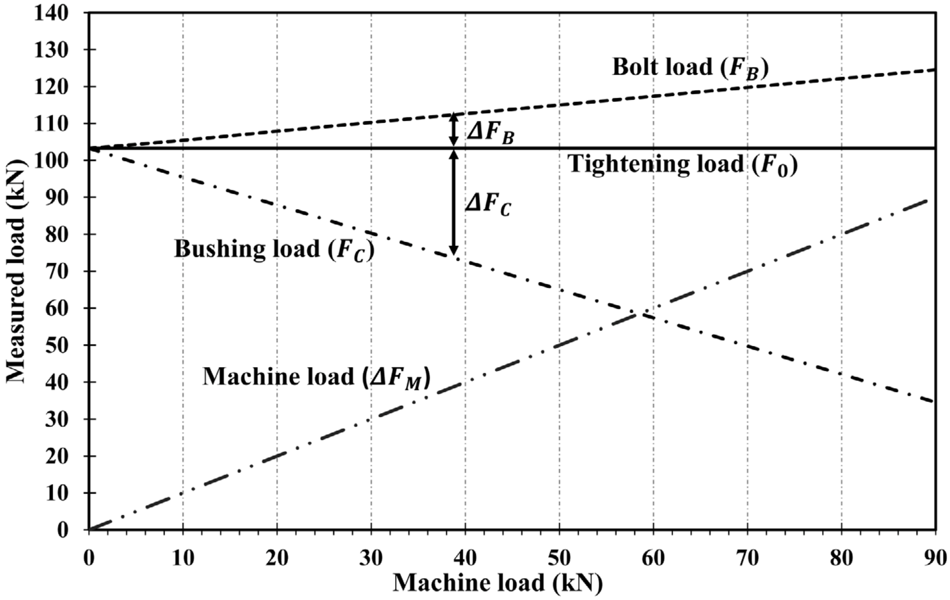

Fig. 5. Evolution of the load on the bolt and the bushing during a quasi-static test on the M16 bolt.

Table 3
Preload, bolt load applied, nominal stresses produced in the cycles and fatigue life for M16 bolts.

Bolt #	F_0 (kN)	$F_{B,max}$ (kN)	$F_{B,min}$ (kN)	σ_{max} (MPa)	σ_{min} (MPa)	σ_m (MPa)	σ_a (MPa)	Life (Cycles)
1	105.33	136.54	107.74	826.56	686.22	777.94	91.72	75,749
2	108.37	135.88	110.78	865.49	705.59	785.54	79.95	166,227
3	103.15	132.48	109.44	843.84	697.09	770.47	73.38	273,962
4	107.04	126.29	105.56	804.43	672.35	738.39	66.04	539,275
5	104.54	126.63	106.95	806.54	681.19	743.86	62.68	675,976

2.6. Application of the methodology on nonserviced M16 bolts

Before applying the methodology to bolts damaged by use in a wind turbine, it has been developed using new M16 bolts with pitch of 2 mm, which have been damaged in the lab by testing. The first step was to define a preload of the bolts. Trying to reproduce a situation such as the one produced in working conditions, a preload of 106 kN was applied, producing nominal stresses corresponding to 75% of the yield stress, that is, 675 MPa. Considering that there was a small relaxation during the first cycles, the preload initially applied with the torque wrench was 112 kN, which after relaxation passed to be F_0 = 106 ± 3 kN.

The following steps are followed to apply the methodology proposed in Section 2.5 on nonserviced M16 bolts.

1. $S-N$ curve. Five new bolts were tested at different load levels to obtain an $S-N$ curve. Table 3 shows the preload, bolt load applied, nominal stresses produced in the cycles and fatigue life for each test.
2. From the $S-N$ curve, a load producing a total life near 500,000 cycles was selected. From the results in Table 3, the stresses selected to be applied were σ_m =675 MPa and σ_a = 67 MPa. Four tests were carried out with this load combination to check the load selection and to obtain a better approach to the expected fatigue life. The results are collected in Table 4. Fig. 7 shows the results of Tables 3 and 4. The average life obtained in those tests was 414,736 cycles. It should be noted that due to the variability of the relaxation of the bolt connection and the scatter of the M_0-F_0 ratio, it was not possible to maintain all testing parameters identical for all tests. Nevertheless, the amplitude of the stresses was always the same, fixed by the testing machine control system
3. To have predamaged bolts, 16 partial tests of new commercial M16 bolts with the same conditions as these last four tests were carried out. All tests were stopped after 207,368 cycles (N_P), which corresponds to half of the average life expected (414,736 cycles). The test rig was disassembled after each test.
4. The identification of the original position of the nut during the predamage is done by considering the number and thickness of the washers between the nut and the threaded head (Fig. 8) and the dimensions of the bolt and the clamping heads.
5. and 6. Steps 5 and 6 of the procedure described in Section 2.5 are developed in detail in Sub- Sections 2.6.1–2.6.2 with the aim of providing more insight into some critical aspects of the procedure related to the position of the initial damage generated during the ageing phase. Finally, Sub- Section 2.6.3 presents an alternative approach to estimate the residual fatigue life

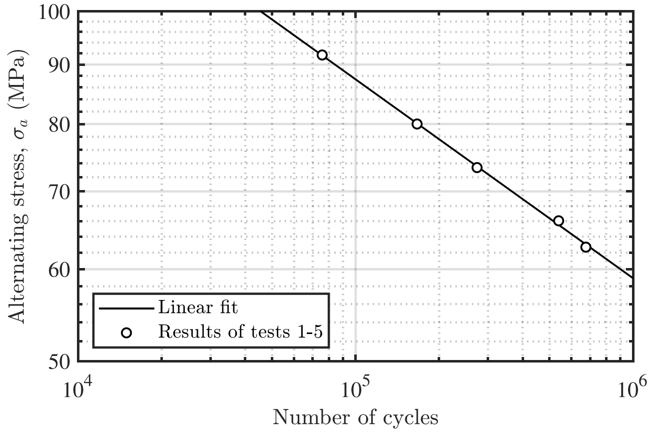

Fig. 6. S–N curve, including a linear fitting, for five tests conducted on M16 bolts of class 10.9 for different alternating stresses..

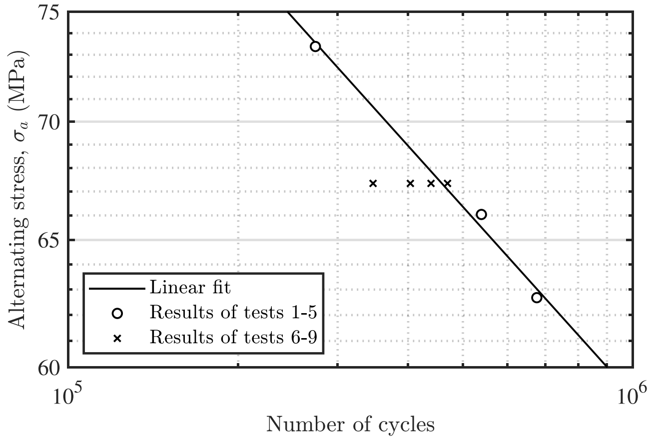

Fig. 7. $S-N$ curve, fitted line, and test results 6–9 for an alternating stress $\sigma_a = 67.34$ MPa.

Table 4
Preload, bolt load applied, nominal stresses, and lives obtained for a fixed stress amplitude in four tests conducted on M16 bolts.

Bolt #	F_0 (kN)	$F_{B,max}$ (kN)	$F_{B,min}$ (kN)	σ_{max} (MPa)	σ_{min} (MPa)	σ_m (MPa)	σ_a (MPa)	Life (Cycles)
6	93.14	116.69	95.54	743.24	608.57	675.9	67.34	346,628
7	95.28	118.82	97.68	756.81	622.13	689.47	67.34	469,645
8	90.78	114.32	93.18	728.17	593.49	660.83	67.34	403,613
9	88.10	111.64	90.50	711.10	576.43	643.77	67.34	439,057
							Average	414,736

without considering the knowledge of the results of the initial ageing of the bolts. This alternative method is of particular interest when dealing with serviced bolts, where the effect of the initial ageing on the bolt life is not precisely known. The alternative approach is based on the following hypothesis: when the initial damage is far enough from the most loaded point of the bolt thread during the residual life estimation test, the initial damage does not progress, and instead, new damage appears, resulting in the bolt showing its full fatigue life under the tested conditions. This hypothesis is supported by the results and discussions of Sub- Sections 2.6.1–2.6.2.

2.6.1. Influence of the initial damage position on the residual life estimation

To analyse the effect of errors when positioning the nut for bolts coming from a wind turbine, four groups of four test each of previously damaged bolts have been carried out until failure: (a) four tests with the nut exactly in the same position as it was in the predamaging tests, using a washer with a thickness of 2 mm; (b) another group adding a 0.5 mm thickness washer to the original one so that the nut will be separated $\frac{1}{4}$ of the pitch from the position that it had during the predamaging tests; (c) the third group was tested after adding a 1 mm washer to the original one, which means a theoretical error of $\frac{1}{2}$ pitch or 180° of positioning the

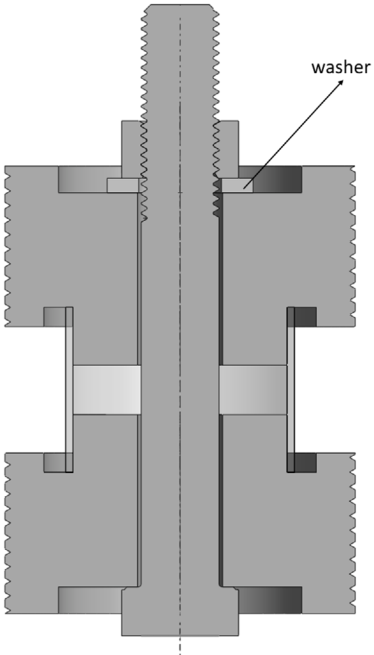

Fig. 8. Washers to simulate ignoring the original damage position.

nut; and (d) finally, the fourth group was tested until failure substituting the original washer with a thickness of 2 mm by another washer with a thickness of 1 mm to have the most damaged zone of the bolt 1 mm inside the nut. Fig. 8 shows the assembly where the position of the washer is visible.

Table 5 shows the loads and stresses applied in all tests, the predamaging number of cycles (N_p), the number of cycles until failure for each test (n_j), and the total life for each bolt, $N_{Tj} = N_P + n_j$ (tests 10 to 25). The residual life is calculated as $r_j = n_j/N_{Tj}$, and finally, the average residual life for each group is also shown. Fig. 9 schematically shows the average residual lives obtained in the four groups of tests. As shown in Fig. 9, the first group of tests, the vertical position of the nut does not vary with respect to the position of the reference tests (0 mm), and therefore the damage accumulates to the previous damage, obtaining a residual life of 36% (151,013 cycles). In the second group of tests, adding a 0.5 mm washer, the nut rotates 90° counterclockwise with respect to the previous case, obtaining a higher residual life of 51% (212,790 cycles), because not all the previous damage is accumulated as in the previous group of tests. Finally, a 1 mm thick washer is added on top of the 2 mm thick original washer in the third group (+180°), while the 2 mm thick original washer is substituted by a 1 mm washer to generate a -1 mm displacement of the nut, which is subtracted in the fourth group (−180°), resulting in residual lives of 47% and 48% (194,963 and 200,903 cycles), respectively. While the residual life corresponding to the angular position of the nut of 90° shown in Fig. 9 is slightly larger than those corresponding to angular positions 180° and -180°, the differences fall within the typical scatter that can be found in fatigue testing.

2.6.2. Influence of the localization of the initial damage by microscope inspection

In real situations, the position of the nut or the ring thread relative to the bolt when assembled may not be known. Therefore, for tests positioning the nut in the correct position, one could proceed as follows: (1) knowing the ring and flange dimensions as well as the preload of the bolts, it would be possible to estimate that position; and (2) the other possibility is by analysing the bolt thread after cleaning in an ultrasonic bath with a stereo microscope to detect the point where there are signals of contact with the nut when assembled. To analyse the effect of this possible error, four more tests were carried out. For that, four bolts were predamaged, but an undefined number of washers of different thicknesses were installed for each test to obtain different nut positions. After disassembling the system, bolts were given to a different technician to estimate the previous position of the nut with a microscope. According to that estimation, a number of washers with different thicknesses were used to put the nut in the adequate position for the test until failure. Due to the variation in the stiffness ratio between the bushing and bolt for each number of washers installed, it was difficult to obtain exactly the same stress variation for all tests. Tests 26 to 29 in Table 6 show the forces and stresses applied, the expected life (N_{Ej}), according to the $S-N$ curve of Fig. 6 for each test, the number of predamage cycles applied, the total number of cycles to failure, N_{Tj}, and the deviation with respect to the experimental total life, ϕ_j. Note that a positive (negative) value of ϕ_j means that the total number of cycles to failure is larger (shorter) than the expected life. As shown in Table 6, locating the position of the nut during service (or predamage partial test) by inspection using a microscope seems to provide satisfactory results and does not significantly influence the residual life.

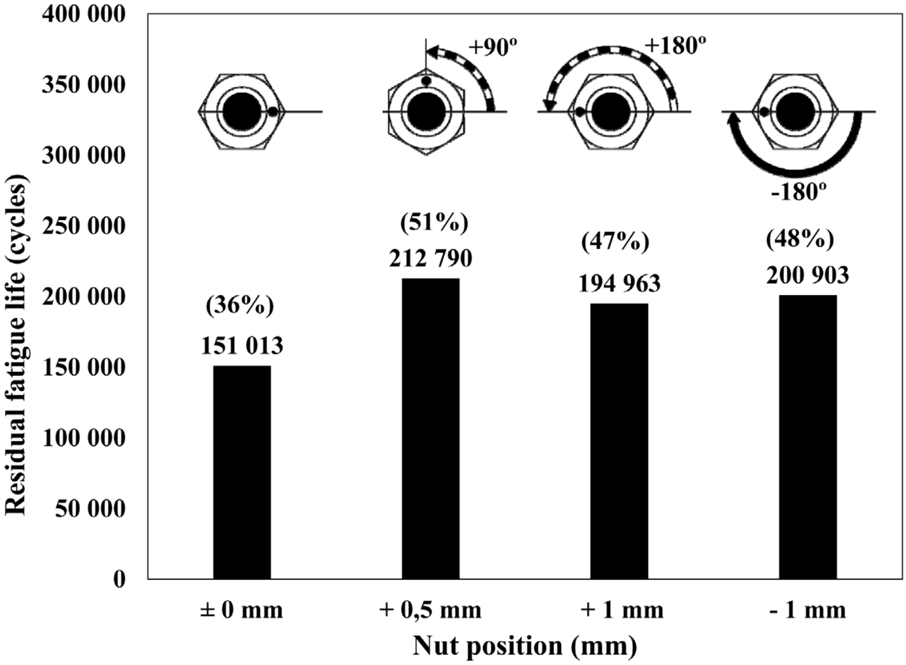

Fig. 9. Average residual lives for different positions of the nut for four groups of four tests. In all cases, a predamage of 207,368 cycles was applied. The arrows represent the angle rotated by the nut from the original position (0 mm). Dashed arrows indicate that the nut was rotated +90° (+0.5 mm) or +180°(+1mm) from the original position, while a continuous arrow indicates that the nut was rotated -180° (−1 mm) from the original position.

2.6.3. Determining the fatigue life of a new bolt from a damaged bolt

In real working situations, after several years running the wind farm before the first bolt failure, it may be difficult to find some unused original bolts to be used as a reference in tests. New bolts with the same specifications may not be identical nor ensure the same fatigue strength. Small variations in the material characteristics, even if manufactured under the same standard as the original bolts, or small variations in the manufacturing process may produce different fatigue lives than the original ones. One alternative may be to take used bolts as a reference but test them with a different position of the nut so that the damage is produced in a zone that was not damaged previously. If the previous damage was not significant in the current zone of interest, the fatigue lives measured in the tests would be the same as those produced with new bolts.

To analyse the possibility of taking used bolts as a reference to determine the undamaged bolt fatigue strength, several tests were carried out on M16 bolts. The first was a test until failure (539,275) on a new bolt with a defined load. Two partial life tests (interrupted before breakage failure) were carried out with the same load to a number of cycles equal to the half-life of the previous test. These two partially tested bolts were retested, but the position of the nut was changed. In one case, the nut was two thread pitches closer to the bolt head than in the first test. Therefore, the previously most damaged region was now the third engaged thread, a zone where the damage will not progress with new cyclic loads. In the other case, the nut was two threads above that of the partial test. In this case, the previously damaged zone will be under tension during the cyclic loading test, but with a stress concentration much lower than that produced in the first engaged thread. In all cases, bolts were preloaded to 103 kN, an average stress of 739.2 MPa and an alternating stress of 66 MPa. The life obtained in the first test was 539,275 cycles, see Fig. 10.

The other two bolts were partially tested until 250,000 cycles with the nut in the position resulting from the tightening torque. In the second part of the test, the nut was moved downwards from the predamage position, continuing the tests under the same tension until failure. Failure occurred after 539,775 cycles. This life was almost the same as that produced in the first test with a new bolt. In the third bolt, the nut moves away from the bolt head in relation to the partial test position and was retested until failure, which occurred after 278,403 new cycles. Fig. 10 schematically shows these results.

As shown in Fig. 10, the second test after changing the position of the nut below the previously damaged position, the number of cycles obtained (539,775) is similar to that obtained in the test of a new bolt (539,275). This is because there was practically no damage in the area of the first engaged thread after moving the nut downwards. Therefore, the bolt behaves similarly to a new bolt. For the third test, where the nut was moved in the opposite position, the previously damaged zone continued supporting the working stresses, and the initiated crack continued growing until failure, which was produced after 53.7% (278,403 cycles) of the new bolt life.

These results suggest that the bolts tested with the most previously damaged zone far enough from the stressed zone during the test inside the nut-engaged zone can be used as a reference to estimate the fatigue life of new bolts.

Table 5
Summary of the conditions and results of the four groups of tests conducted on M16 bolts for different position of the nut with a pre-damage, N_P, of 207,368 cycles. The alternating stress σ_a was 67.34 MPa in all tests. The resulting partial lives until failure (n_j), the total lives ($N_{Tj} = N_P + n_j$) and the residual lives $r_j = n_j/N_T$ where N_T is 414,736 (see result in Table 4) are also shown.

Bolt #	F_0 (kN)	$F_{B,max}$ (kN)	$F_{B,min}$ (kN)	σ_{max} (MPa)	σ_{min} (MPa)	σ_m (MPa)	n_j (Cycles)	N_{Tj} (Cycles)
0 mm								
10	92.91	116.46	95.31	741.76	607.09	674.43	182815	390183
11	95.86	119.4	98.26	760.53	625.85	693.19	144771	352139
12	95.74	119.28	98.14	759.76	625.08	692.42	131250	338618
13	94.64	118.19	97.04	752.79	618.11	685.45	145217	352585
						Average:	151,013	358,381
						Deviation:	22,170	22,170
						Residual life, r_j, of 36%		
+0.5 mm								
14	94.09	117.64	96.49	749.27	614.6	681.94	193147	400515
15	92.32	115.86	94.72	737.96	603.28	670.62	291548	498916
16	94.54	118.08	96.94	752.12	617.45	684.78	197767	405135
17	95.8	119.34	98.2	760.13	625.46	692.79	168701	376069
						Average:	212,791	420,159
						Deviation:	54,031	54,031
						Residual life, r_j, of 51%		
+1 mm								
18	94.89	118.44	97.29	754.38	619.7	687.04	300187	507555
19	92.13	115.68	94.53	736.79	602.12	669.45	138146	345514
20	93.57	117.12	95.97	745.97	611.29	678.63	199678	407046
21	92.42	115.96	94.82	738.6	603.92	671.26	141842	349210
						Average:	194,963	402,331
						Deviation:	75,596	75,596
						Residual life, r_j, of 47%		
−1 mm								
22	93.06	116.6	95.46	742.69	608.02	675.35	205409	412777
23	92.33	115.88	94.73	738.06	603.39	670.73	288952	496320
24	95.14	118.69	97.54	755.97	621.29	688.63	157009	364377
25	95.24	118.78	97.64	756.59	621.92	689.25	152241	359609
						Average:	200,903	408,271
						Deviation:	63,423	63,423
						Residual life, r_j, of 48%		

Table 6
Partial tests of M16 bolts for different washer thicknesses with a pre-damage, N_P, of 207,368 cycles. In this table, N_{Ej} stands for the number of cycles expected for experiment j and ϕ_j is the deviation with respect to the experimental total life, calculated as $\phi_j = 100 \cdot (N_{Tj}/N_{Ej} - 1)$.

Bolt #	F_0 (kN)	$F_{B,max} - F_{B,min}$ (kN)	$\sigma_{max} - \sigma_{min}$ (MPa)	$\sigma_m - \sigma_a$ (MPa)	N_{Ej} (Cycles)	N_{Tj} (Cycles)	ϕ_j (%)
26	95.24	117.19–97.47	746.44–620.86	683.65–62.79	660780	717252	8.55
27	97.55	120.38–99.88	766.76–636.18	701.47–65.29	549770	458083	−16.68
28	95.34	118.39–97.69	754.08–622.21	688.15–65.93	519285	478087	−7.93
29	94.86	117.45–97.16	748.08–618.85	683.47–64.62	583950	492769	−15.61

3. Results and discussion of the application of the presented methodology to serviced M20 bolts

All the methodology developed in this work has been carried out with new M16 bolts where there is no previous damage and the torque loads and machine loads are known and under control. However, the methodology for estimating the residual life of previously used bolts needs to be applied to bolts belonging to wind turbines that have been in service and have had a service life. For this purpose, M20 bolts of quality 10.9 extracted from a wind turbine that has been in service were tested. These bolts were removed from the blade–hub connection and taken to the laboratory for fatigue tests. The objective is to experimentally obtain the number of residual life cycles of the removed bolts and to compare it with the number of cycles that a new bolt can withstand. As there are no identical new bolts, this information must be obtained from bolts in service by means of the procedure described in Section 2.6.3. The type of tests to be carried out is, on the one hand, tests with the engaged thread in the same position of the bolt as it has been in service and, on the other hand, tests carried out in a nut position closer to the head than the original position.

In wind turbines, the bolts are subjected to various stresses at the blade–hub connection during their operating life. The bolts are grouped according to the zone where they are located within 360° of the flange (see Fig. 11). Therefore, to select the bolts to be tested, they should be extracted as close together from the same area as possible so that the residual lives are similar.

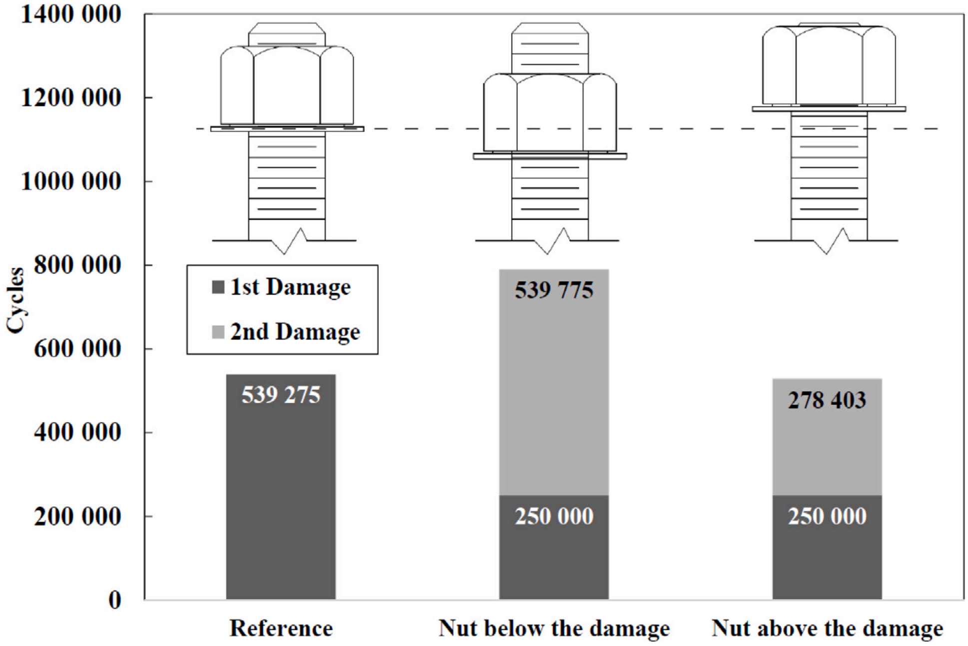

Fig. 10. Fatigue test results for new commercial M16 bolts with different vertical positions of the nut. Test in reference position (new bolt until failure), nut below the reference (with a predamage of 250,000 cycles in the reference position), and nut above the reference (with a predamage of 250,000 cycles in the reference position).

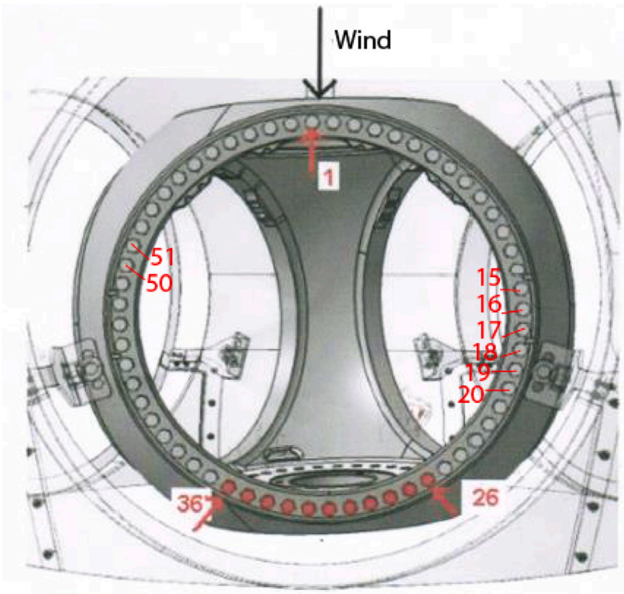

Fig. 11. Bolt position blade-housing.

For the M20 bolts, the working and alternating forces are very high. During the first phase of testing, the procedure described in the proposed methodology was followed, testing the bolt with the torque of Section 2.3. However, as the test loads were high and it was difficult to reach the target torque, the results showed a lot of dispersion, as the testing machine did not reach the target forces in a stable way. For this reason, in the tests, the maximum and minimum forces applied were directly provided by the servo-hydraulic test machine so that the prescribed σ_{max} and σ_{min} were achieved by the bolts without using the torque device.

All tests were performed with the force values shown in Table 7, reaching stable alternating stresses of approximately 51 MPa. Two types of tests were performed: the first test in the original position, where the nut was placed in the same vertical position

Table 7
Results of the residual life estimation method propose for serviced M20 bolts. Test results for the nut located below (B) the original position (O), and test results with the nut in original position (O), identified by microscopic inspection. In this table, n_j is the life until failure, N_{Tj} is the estimated total life as a new bolt, and r_j is the residual life.

Bolt #	Pos.	$F_{B,max} - F_{B,min}$ (kN)	$\sigma_{max} - \sigma_{min}$ (MPa)	$\sigma_m - \sigma_a$ (MPa)	n_j (Cycles)	N_{Tj} (Cycles)	r_j (%)
19	B	206.27–181.15	845.37–742.42	793.89–51.48	–	289438	
16	B	206.23–181.21	845.20–742.66	793.93–51.27	–	299465	
15	B	206.26–181.17	845.33–742.50	793.91–51.41	–	328703	
17	B	206.25–181.21	845.29–742.66	793.98–51.31	–	384628	
					Average	325558	
51	O	206.25–181.23	845.29–742.75	794.02–51.27	259103	325558	79.6
20	O	206.24–181.19	845.25–742.58	793.91–51.33	274785	325558	84.4
52	O	206.28–181.31	845.41–743.07	794.24–51.17	275429	325558	84.6
18	O	206.22–181.15	845.16–742.42	793.79–51.37	289413	325558	88.9
						Average	84.4
						Deviation	±3.8

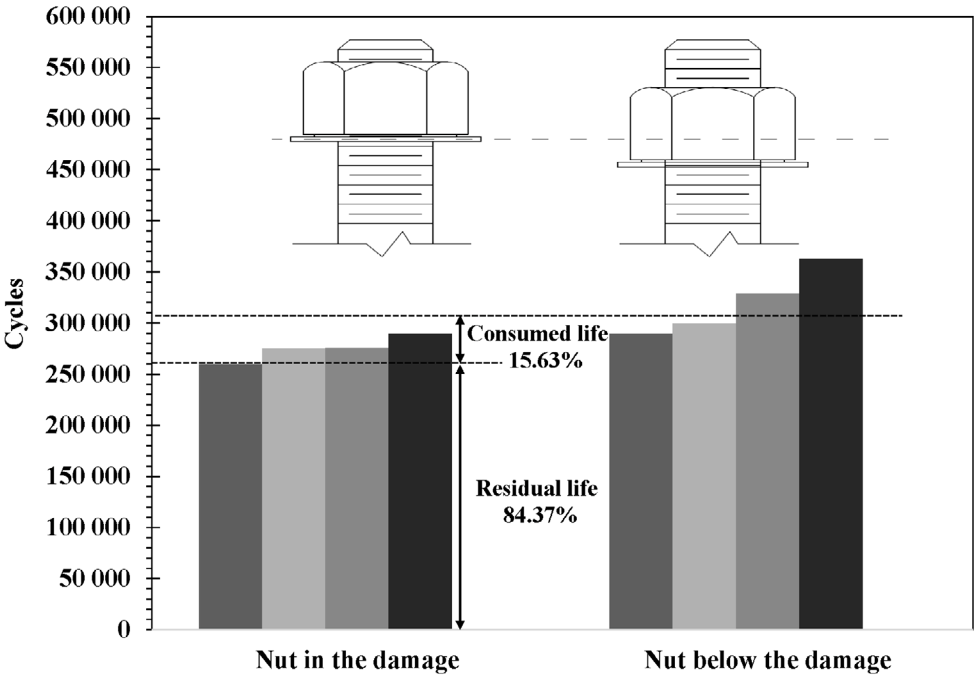

Fig. 12. Test results of serviced M20 bolts with the nut in the original damage position compared with test results for bolts with nut below the damage position. Residual and consumed life results.

as it was in the wind turbine, and the second test where the position of the nut was two thread pitch below the original position. Fig. 12 shows a graphical representation of the two types of tests, showing that the number of life cycles of the bolts tested below the original position is higher than that of the group of bolts tested in the original position. The result shows that the average life difference is 15.6%; therefore, the used bolts have consumed 15.6% of useful life and have an average residual life of 84.4% of their total life.

4. Conclusions

This study proposes an experimental method for estimating the residual life of wind turbine blade–hub connection bolts that have been working for a time and may have suffered significant damage. It has been shown that small errors in positioning the nut for residual life tests (less than 180° of the nut position) produce small deviations of residual life estimations. Additionally, a procedure has been proposed to estimate the fatigue life of new bolts from used bolts, just by testing used bolts with the nut displaced relative to the original position to avoid stressing the most damaged region of the bolt. This simplifies the analysis procedure, avoiding having to store unused bolts during the life of the connection. The procedure has been applied to serviced M20 wind turbine bolts taken from a connection belonging to a blade from the same rotor where another blade connection had failed. Therefore, the information obtained in this study is useful in carrying out predictive maintenance in wind farms to obtain useful information for the replacement of bolts before consuming their useful life and to avoid catastrophic failures. Furthermore, the results of this study can be extrapolated in general to other types of bolted joints subjected to cyclic loads, providing the procedure to be followed for carrying out experimental tests.

Declaration of competing interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

Data availability

Data will be made available on request.

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