



Research paper

Clamping parameters in full-scale tidal turbine blade tests: A case study

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ABSTRACT

Full-scale laboratory tests of composite tidal turbine blades follow standards adapted from the testing of much longer wind turbine blades. Due to the much higher density of seawater, when compared to air, the diameter of an equivalent-power marine tidal stream turbine is approximately 4 times less than that of a wind turbine. In this paper, relevant guidelines regarding load introduction points for testing of tidal blades are assessed for the first time, specifically, the dimensions of the exclusion zones surrounding these loading points, that are ordinarily excluded from structural analysis. The dimensions of these zones are determined as a function of the blade chord length at the loading saddle locations, and the relevant standards suggest an exclusion length of up to one chord length at both sides of the load introduction points along the blade length. This study appraises these boundaries via a full-scale composite tidal blade structural test at the FastBlade facility, using a saddle-hydraulic actuator system. Strain measurements and Digital Image Correlation (DIC) are used to quantify the local strain contributions from saddle to blade clamping, during both static and fatigue loading. We propose a reduction of the exclusion zone boundary length from 1 to 0.45 times the chord length, based on the experimental comparison of two clamping methods: a manual torque-wrench method and a hydraulic-pressure method. The saddle material was also assessed by comparing the structural performance of MDF and plywood during these blade tests, with plywood showing better characteristics for high blade design loads. Our analyses could be used to update current standards to better align with the unique characteristics of composite tidal turbine blades, to effectively minimise the number of structural tests required for blade certification.

1. Introduction

1.1. Tidal energy

Tidal energy has always been an attractive sustainable source of energy as it is more predictable and more reliable than wind and solar energy (O. Rourke et al., 2010). The total potential tidal energy available in the UK is estimated to be 50 TWh/yr and represents 48 % of the total resources available in Europe (Burrows et al., 2009), which could enable a contribution of up to 16 % of the UK's total electricity consumption.

The energy potential of tidal turbines in particular is significant, with the world's oceans containing vast amounts of kinetic energy, estimated to be 2.3 TW (Ferrari, 2009), that can be harnessed for electricity generation. Tidal turbines, adaptations of wind turbines, can be used for this purpose. The first tidal turbine demonstration in the UK was in Loch Linnhe, Scotland in 1994 (Thake, 2005), and consisted of a 10 kW two-bladed turbine with a diameter of 3.5 m.

Currently, there are multiple companies working on various concepts for tidal turbines, but they are still at low technology readiness levels (TRL) (Sleiti, 2017). The lack of a industry-wide accepted tidal turbine concept hinders progress in the R&D of other important components of these devices. This issue has a noticeable impact on the current levelised cost of energy (LCoE) for tidal energy, which is £178/MWh based on the most recent contract for difference (CfD) bids awarded to four projects in the UK in 2022 (ORE Catapult, 2022). This cost is significantly higher than the expected £116/MWh (adjusted for inflation) estimated in a 2011 study (Allan et al., 2011). In fact, the current LCoE is 53 % higher than the projected cost in 2011.

The performance of a tidal turbine device depends strongly on the design of its blades and the properties and behaviour of the materials used to make them. The blades of a tidal turbine are exposed to harsh underwater conditions, including high velocity flows, turbulence, and marine growth, amongst others. As such, the design and testing of tidal turbine blades should ensure structural integrity, optimal efficiency,

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and safe operation. Blade testing in turn combines both experimental and computational techniques (Masters et al., 2015; Harper and Hallett, 2015; Nachtane et al., 2020), scaled testing in water tanks or test rigs, and field testing.

1.2. Methods for the introduction of loads on blades

Full-scale laboratory testing is a cost-effective alternative to field testing and full deployment. Water tanks (Clarke et al., 2007; Milne et al., 2015; Shi et al., 2016; Murray et al., 2018; Barber et al., 2017; Porter et al., 2020) are commonly used to aid in testing blade performance, or indeed, the entire tidal turbine structure. This often involves scaled-down models (e.g. for wind turbine blades Lin et al., 2023) of the turbine subjected to simulated tidal conditions. This water tank testing can provide valuable information about the performance of the hydrodynamic design of a blade, which is commonly validated with computational fluid dynamics (CFD) (Badoe et al., 2022), but could also be coupled with finite element modelling (Motley and Barber, 2014; Nachtane et al., 2017). However, the structural design of the blades is often neglected in these types of experiments. Nevertheless, there are several studies (Lopez Dubon et al., 2024; de la Torre et al., 2018; Fagan et al., 2020; Finnegan et al., 2021; Glennon et al., 2022; Davies et al., 2022; Finnegan et al., 2022) where test rigs such as those used in the wind energy industry are employed to test the structural integrity of the blades.

For these laboratory test facilities, several load introduction configurations have been employed, primarily on wind blades (Malhotra et al., 2012; Melcher et al., 2020; Yenicieli, 2014; Yang et al., 2013; Greš et al., 2022; Castro et al., 2024). Similar systems can be seen in aircraft wing tests (Rößler et al., 2009; Shim and Ahn, 2014; Heida and Hwang, 2014; Sunder, 2017; Ser'eznov et al., 2020; Yang et al., 2020), and long pole structures (Polyzois et al., 1999; Shakiba et al., 2023). Winches (Ser'eznov et al., 2020; Yang et al., 2012), whiffle trees (IABG, 2005), straps (Sunder, 2017), and sleeves (Polyzois et al., 1999) have been used to transfer the load from ballast weights (Malhotra et al., 2012; LeBlanc et al., 2013), cranes (Shakiba et al., 2023; Yang et al., 2012; Chen et al., 2014; Zhang et al., 2019) and hydraulic actuators (Lopez Dubon et al., 2024; de la Torre et al., 2018; Finnegan et al., 2022; Malhotra et al., 2012; Rößler et al., 2009; Shim and Ahn, 2014; Heida and Hwang, 2014; Polyzois et al., 1999; Shakiba et al., 2023; Lahuerta et al., 2018; Williams and Ó Brádaigh, 2021) to these cantilever structures. These systems are often attached to a load distributor to reduce stress concentrations. The most common load distributor is the saddle system, but adhesive pads have also been used in some wind blade and aircraft wing tests (IABG, 2005). The use of rubber pads increases the complexity of the loading methodology, as multiple bearings must be used in conjunction with whiffle trees to reduce second-order moments. Rubber pads are also more prone to slipping when they are not bonded to the blade surface (Lopez Dubon et al., 2024).

Although there has been considerable research conducted on validating tests, the primary focus has been on enhancing the accuracy of the distribution of forces along the blade. As a result, developers have often overlooked load introduction methods, and analyses of the areas of the blade surrounding load introduction points. This oversight is significant because blade failures can occur precisely beneath the saddles because of stress concentrations at these contact points (Joergensen et al., 2004), indicating the importance in thoroughly examining these regions. The only study that focused on these contact regions was done by Klaenfoth et al. (2023), however, the subject was a wind turbine blade and the results are based entirely on numerical simulations. Their results confirm that the pressure applied to the blade is not evenly distributed, thus requiring a lot more research to identify all the variables involved in this interaction.

Regardless of the load introduction method used (Lopez Dubon et al., 2024) or the number of load introduction points (Lopez Dubon

Table 1

Relevant testing standards for full scale tidal turbine blades.

Standard	Description
DNVGL-ST-0164	Main standard for tidal turbines
IEC TS 62600-2	Design requirements
IEC TS 62600-3	Measurement of mechanical loads
IEC TS 62600-202	Pre-prototype scale testing
DS 5760, Part 5	Failure modes, effects, and criticality analysis
DNV-OS-C501	Composite components
DNVGL-ST-0376	Rotor blades for wind turbines
IEC 61400-23	Load introduction

et al., 2024), standards use the blade chord length as a reference to identify the disturbed area at each side of the load introduction point where the performance of the blade cannot be evaluated (IEC, 2001, 2022). As the chord-to-span ratio is relatively high in tidal blades compared to wind blades, respecting the chord length criterion of 1 often dramatically increases the number of tests required to analyse the whole surface along the blade.

1.3. Standards for tidal blade testing

The test procedures at any full-scale laboratory blade testing facility follow international standards that grant operating certification for blade designs before moving on to mass-scale manufacture. These testing standards are based on extreme design load cases for static testing, whereas the blade life expectancy is proven by fatigue testing. Although there are several standards and guidelines for tidal energy developments, DNVGL-ST-0164 is the most used standard for tidal turbines (DNV GL, 2015), which refers to several other service specifications and standards for the structure's full manufacture, assembly, and deployment. Relevant testing standards often referred to in the DNVGL-ST-0164 document are shown in Table 1, following a description of their content of interest. Only the first three standards are specifically tailored for marine energy devices. Guidelines for scaled-prototype stages are only considered in the IEC TS 62600-202 standard (IEC, 2022), however, it focuses primarily on the entire turbine, and thus blade testing is not mentioned throughout the document.

The DNVGL-ST-0164 standard refers to IEC 61400-23 (IEC, 2001) amongst others when analysing the load distribution in a blade through full-scale laboratory testing. Section 18.2 of the DNVGL-ST-0164 standard specifically discusses blade testing procedures and highlights the main structural properties that a test should measure: mass and centre of gravity, stiffness distribution, natural frequency, ultimate strength, and fatigue strength (DNV GL, 2015).

For static tests, extreme load cases should be considered, thus performing separate tests for flap-wise and edge-wise load cases in both directions, pressure and suction for the flap-wise load, and trailing and leading for the edge-wise load.

In the case of fatigue testing, this procedure is only strictly required once the prototype stage has been completed and the blade design is ready for full-scale production. The blade that will be tested for fatigue should already have been tested statically. The requirements that the DNVGL-ST-0164 contains regarding full-scale blade testing come from the IEC 61400-23 standard, which is explicitly dedicated to wind turbine blades. Since a standard dedicated to full-scale tidal turbine blades is still unavailable and the IEC 62600-3 is still not used for blade certification, most design specifications for test setups are derived from the IEC 61400-23 standard.

Ideally, the load would be applied continuously from the root to the tip of the blade; however, because of mechanical test practicalities, design loads are applied along the span at a reduced number of concentrated points. If there are particular areas of interest for testing, the load introduction points must be at a distance from these, since the areas around the actuators will be disturbed and can thus no longer

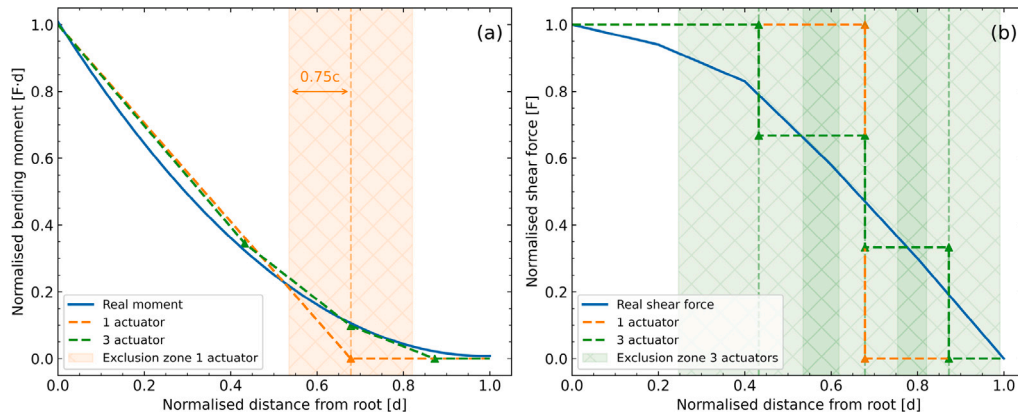


Fig. 1. Comparison between real condition loads, 1-actuator tests, and 3-actuator tests: (a) Bending moment along the blade, (b) Shear force along the blade. 3-actuator tests are likely to have overlapping exclusion zones.

be adequately tested, as shown in Fig. 1. Of the standards in Table 1, the IEC TS 62600-3 and the IEC 61400-23 are the only standards that mention guidelines for load introduction points, based on a chord length variable c . This disturbed area can extend up to three-quarters of a chord length (Section A.8.2 of IEC, 2022) or up to one chord length (Section 7.2 of IEC, 2001) on either side of the fixture. No further specifications regarding the loading structure or common practices to mitigate undesired effects during testing are given.

For the reasons mentioned, the research question of this paper is to understand and consequently minimise the induced effects of load introduction points in full-scale composite tidal blade tests. The following are the objectives of the research: To analyse the performance between different strain measurement devices; to compare two different pre-tension methods; to study the structural performance of two different materials for the saddles; and to identify the maximum deformation contribution from the pre-tension stage.

A few studies have explored laboratory testing with multiple load introduction points for full-scale tidal blade tests (de la Torre et al., 2018; Glennon et al., 2022; Davies et al., 2022; Finnegan et al., 2022; Lopez Dubon et al., 2024), but have focused on different aspects of testing. Only one study addresses these interaction zones (Klaenfoth et al., 2023), however it was carried out numerically on a wind turbine blade and agrees with the current standards. Thus, to the authors' knowledge, no studies have yet focused on the interaction zone between the loading saddles and composite tidal blades. This study addresses the problems with following the standards, which do not consider the short lengths of tidal blades when establishing the exclusion zone around loading points. Consequently, this analysis examines the reduction in the size of these exclusion zones and their impact on the fidelity of the structural analyses of the blade. Less conservative boundaries could effectively optimise the position of the point loads and minimise the number of tests required for certification, while still accurately representing the real moment distribution along the blade.

2. Methods

2.1. Experimental setup

2.1.1. Blade specimen

The blade to be tested, a component of the DeepGen IV tidal project, was designed by Tidal Generation Limited (TGL) and manufactured by Aviation Enterprises Limited. Technical information on the blade is limited as the design company no longer exists. The tidal turbine, shown in Fig. 2, had a capacity of 500 kW and was installed at the European Marine Energy Centre (EMEC) in Orkney in September 2010, being later replaced by a 1 MW turbine in January 2013.

The blade is 5.25 m long and weighs 1589 kg. After several modal tests, the natural frequency was defined at approximately 18 Hz. The



Fig. 2. DeepGen IV tidal turbine prior to first deployment (Huxley-Reynard et al., 2008).

NACA 63-4XX aerofoil series defines the blade profile, which begins with a circular root. As described in Fig. 3, the root is made of cast iron to fit the bolt holes required to fix the blade to the hub. The blade also comprises a skin, ribs, two shear webs, and thick spar caps at the pressure and suction sides of the blade. The thickness-to-chord ratio of the blade decreases from 55 % at the root to a minimum of 18 % at the tip.

The chord length gives the blade a tapered profile, as seen in Fig. 3, and it has been measured at different stations throughout the blade. As blade manufacturers seek certification, the standard guidelines would need to be followed, ignoring the area $1c$ away from the saddle for analysis. Applying these boundaries to Fig. 3 for this specific tidal turbine blade, it is clear that only one actuator could be accommodated per test, significantly increasing the number of tests and therefore the cost of the certification process. It is also evident that the bending moment distribution would be slightly different across the blade.

The blade skin is made of an 8 mm thick glass fibre $\pm 45^\circ$ unidirectional prepreg. The spar, following the centre line along the spar in Fig. 3, is made of carbon fibre unidirectional prepreg, 80 % in the longitudinal direction and the rest filled with prepreg in the transverse direction and a cross-ply woven fabric to provide a bond surface for the shear webs. The shear webs comprise a cross-ply carbon fibre prepreg and are bonded to the spars. Ribs are also bonded perpendicularly to the spar, which are 300 mm apart and spread throughout the whole blade. It follows an "I" channel section, with the web and the flanges being 3 mm and 2 mm thick, respectively. The flanges spread 25 mm, and both components of the ribs are made of cross-ply glass fibre prepreg. Finally, a cross-ply glass fibre rear spar is used 100 mm away from the trailing edge to minimise the peel stresses at the skin joint during operation (Huxley-Reynard et al., 2008).

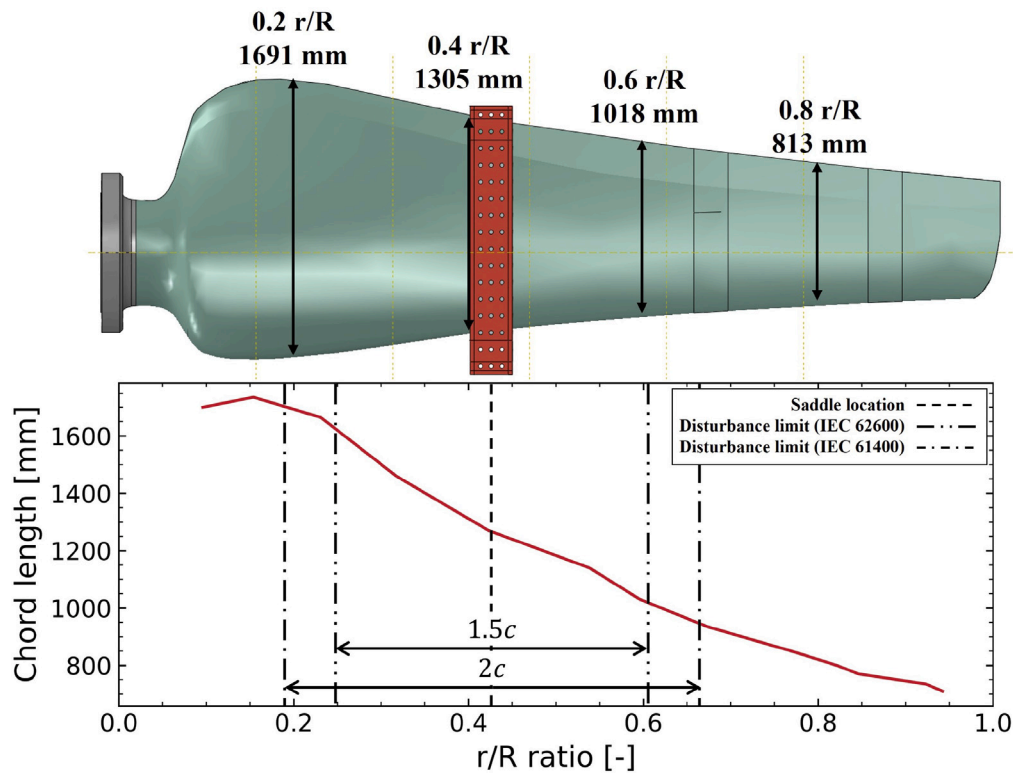


Fig. 3. CAD model and corresponding chord length (c) along the span of the 5.25 m DeepGen IV tidal turbine blade. Based on the chord length at the saddle location, disturbance limits bounding the areas that are not feasible for analysis are drawn.

2.1.2. Testing site and setup

Experimental data was gathered at FastBlade (The University of Edinburgh, 0000), a research facility at The University of Edinburgh. FastBlade is the first facility in the world to carry out full-scale, regenerative hydraulic tests on tidal turbine blades. The testing rig has been specifically designed to assess large and slender structures ranging from 2 m to 14 m in length. These structures can be subjected to static or fatigue loads, and this facility employs a novel Digital Displacement system, which combines regenerative pumping and digital displacement hydraulics. This system enables accelerated testing up to 2 Hz while applying loads of up to 1 MN. More details on the laboratory facilities and capabilities are given in Lopez Dubon et al. (2024), Lopez Dubon et al. (2024).

The setup for this experiment can be seen in Fig. 4, where the blade previously described in Section 2.1.1 is fixed to the test rig. The blade is fixed with the pressure side facing the ground and the suction side facing upwards. Three saddle system actuators are positioned at 2.27 m, 3.56 m and 4.58 m from the root, each applying 95 kN to the blade. These positions and load magnitudes were determined in Lopez Dubon et al. (2024) to maximise the accuracy of the tests with the hydrodynamic simulations. Three M24 steel rods at each side of the saddle were used to generate the clamping force for the first method using a torque wrench. For the second method with a hydraulic bolt tensioner, a single M36 steel rod at each side of the saddle was tensioned. The clamping procedure was followed by a static test at 100 % of the design static load. Each of the three actuators exerted a peak load of 95 kN on the blade. Subsequently, a cyclic load was introduced at 0.5 Hz for 30 cycles with a mean stress of 40 kN and an amplitude of 25 kN, achieving a pure tension R-ratio of 0.23. Table 2 summarises the test regimes for this study. Test cases 1 and 2 are analysed for the tightening mechanism comparison, and test cases 2 and 3 are used for the saddle material comparison.

The analyses in this study were centred around the first actuator, as it had the largest surface area available at both sides of the saddle.

Table 2

Test regimes. Static and fatigue loads are described per actuator.

Test case	Tightening mechanism	Saddle material	Clamp target	Static load	Fatigue load
1	Manual	MDF	95 kN	95 kN	65 kN
2	Hydraulic	MDF	160 bar	95 kN	65 kN
3	Hydraulic	Plywood	160 bar	95 kN	65 kN

Following the procedure from the IEC 61400-23 standard, the chord length (c) from this point of load introduction was estimated to analyse the surrounding surface of the blade. With 1300 mm, this chord length is used as a base for the discussion of results shown in Section 4. If no further analysis was made, the least conservative of the standards would be followed and 0.75 c , or 975 mm, at both sides of the saddle would be excluded. This would leave only half of the available area for analysis between the first saddle and the root connection and using three actuators would not be feasible.

2.1.3. Saddle structure

Various methods for introducing loads to tidal blades have been assessed and examined specifically for this tidal blade (Lopez Dubon et al., 2024; Lopez Dubon et al., 2024), with many of them being adapted from wind blade testing. These methods include using cranes, whiffletrees, winches, and actuators. However, due to the smaller size of tidal blades compared to wind blades, while still needing to withstand high loads, options such as cranes, whiffletrees, and winches are not feasible due to their large-scale structures. As a result, a system comprised of a wooden saddle supported by steel frames was chosen to effectively transfer loads from the actuators to the fixed blade. Medium-density fibreboard (MDF) and plywood are compared in terms of performance for this study, two commonly used woods for saddles in wind blade test facilities (Fagan et al., 2017). MDF is often chosen for its lightweight nature, natural shock absorption properties, affordability, and ease of customisation. Plywood, on the other hand,

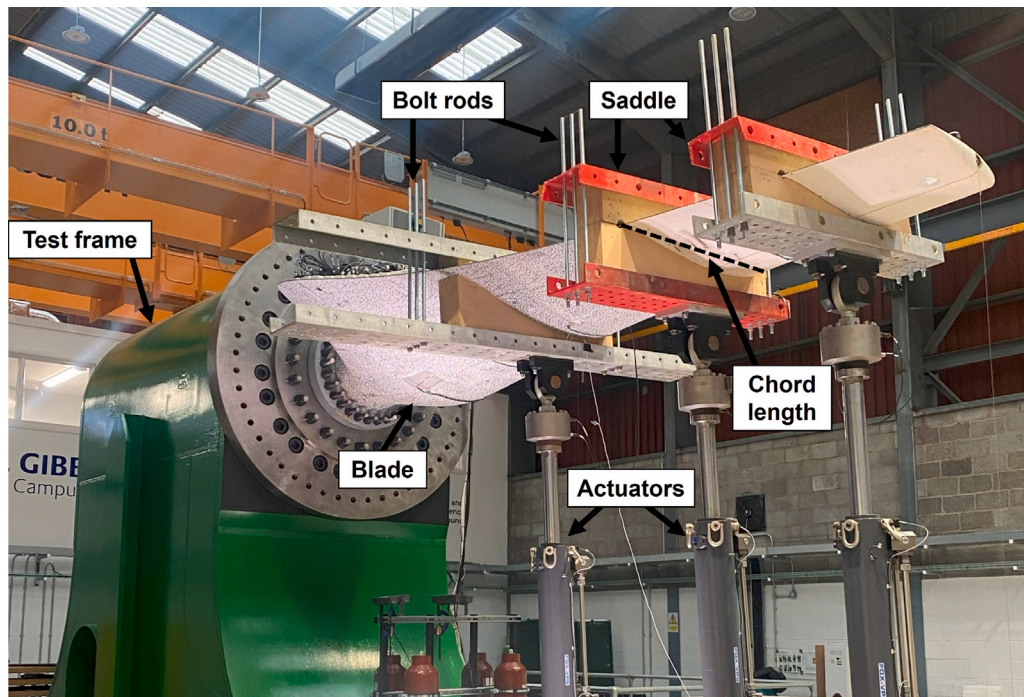


Fig. 4. Experimental setup at FastBlade with three saddles clamped to the blade.

Table 3
Cradle material properties from coupon testing.

Material	Compressive strength [MPa]	Density [kg m^{-3}]	Moisture content %
High-quality MDF	6.36 ± 0.27	596.63 ± 4.14	8.01 ± 0.12
Plywood	31.63 ± 3.41	602.95 ± 26.93	7.16 ± 0.15

is lighter than MDF, more resistant to a change in its properties due to moisture, good for bearing loads, but less affordable.

Saddles have adjustable designs that can accommodate blades of different sizes and shapes, and allowing for versatility in testing. A saddle configuration comprised of a wood-like material in contact with the blade and a steel frame has always been used in wind turbine blade tests (Jensen et al., 2006, 2008; Yang et al., 2012). Both the MDF and the plywood used for this saddle are made of several sheets aligned in the out-of-plane direction. Their mechanical properties, along with their density and moisture content were estimated with coupon tests, following the ISO 13061 standard. The compressive strength, along with density and moisture content are shown in Table 3.

The mechanical properties of the steel frame were classified by the manufacturer as an EN10210-1 grade S355NH, with an elastic modulus of 210 GPa and a Poisson's ratio of 0.30 (Eurocode Applied, 2005). The frame was assembled using 300 mm x 90 mm C-section channels. Bending was observed in these steel frames during clamping, however, no signs of structural damage were observed after the tests. This saddle part is clamped with steel rods at both ends of the frame and bolted into the hydraulic actuator, as shown in Fig. 5. The wooden cradle supports the blade and was designed to match the shape and contour of the cross-section profile, providing a secure and stable resting position during testing. A thin layer of rubber was inserted between the saddle and the blade to increase the pressure distribution (Lopez Dubon et al., 2024).

2.1.4. Clamping equipment

Two different clamping mechanisms have been tested for comparison: a torque wrench and hydraulic bolt tensioners. The torque wrench in Fig. 6a had a range of five evenly spaced steps, ranging from 27.1 N m to 81.4 N m. Initially, it was used on one side to tighten the three M24

bolts (Fig. 7), followed by the tightening of the other three bolts in the other side of the saddle before increasing a step on the applied torque. To account for the induced bolt load loss from tensioning a bolt adjacent to an already tensioned bolt, all bolts were re-tightened, effectively reducing the cross loading factor (CLF). The order in which the bolts were tightened was similar between all tests to achieve similar applied torques.

For the subsequent tightening method, the bolt tensioners shown in Fig. 6b employed were HTS20 M36 tensioners manufactured by Tension Pro (Staffordshire, UK). It had a maximum pressure of 1500 bar, a maximum load of 443 kN, a hydraulic pressure area of 2954 mm², and a maximum stroke of 15 mm. One M36 steel rod on each side was used in this method to be able to tighten them in parallel, as shown in Fig. 6, with rectangular washers to distribute the load to the same area as the previous method. The hydraulic pressure was gradually increased from 0 bar to 160 bar, achieving 47 kN of tension per bolt, which matched the design flapwise load coming from the actuator using Eq. (3). A 100 % tensioning procedure was used, thus both bolts on the saddle were tensioned simultaneously to minimise cross load losses.

2.1.5. Image acquisition

DIC images were collected and processed using the MatchID software (MatchID, Ghent, Belgium), and the setup consisted of six cameras working in pairs. These cameras are FLIR Blackfly S USB3 (BFS-U3-88S6M-C), with characteristics shown in Table 4. The first and second set of cameras pointed to the region between the root of the blade and the first saddle, named RO11 on Fig. 8(b). The third set pointed to the surface area between the first and second saddle, defined as RO12. As all sets use a stereo configuration, calibration was critical to correlate both cameras. 150 high-resolution images were taken, targeting a calibration plate with 107 dots evenly spaced by 50 mm. Calibration was processed individually for every set of cameras, and the average error was 0.03 pixels across all sets.

After calibration, the exposure of the cameras ranged from 10 ms to 13 ms, based on the lighting conditions throughout the test. For the clamping phase of the test, the DIC cameras captured images at 5 Hz. This was increased to 10 Hz for the static phase and the fatigue phase of the test. The blade was covered with 6 mm speckles in the pressure

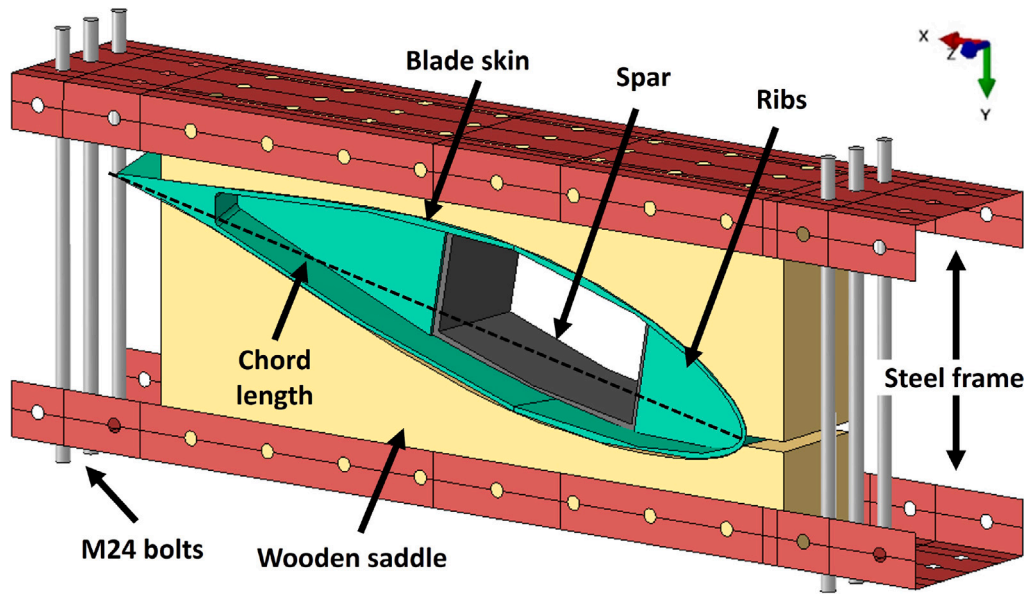


Fig. 5. Simplified blade-saddle system. Steel frame cut for visual purposes.

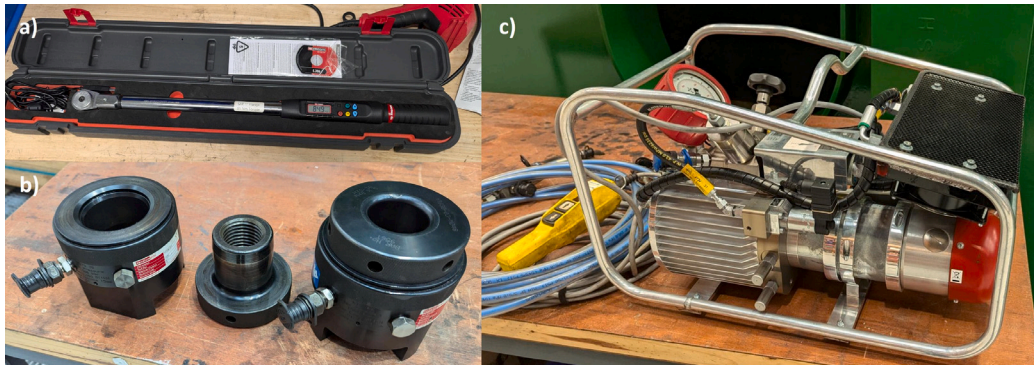


Fig. 6. Equipment used for clamping: (a) manual torque wrench, (b) bolt tensioners for M24 and M36 bolts, (c) pressure pump with parallel connections.

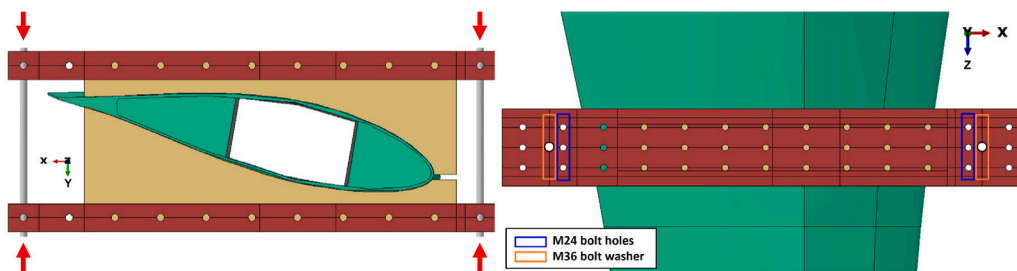


Fig. 7. Bolt locations in the system: cross-section view (left) and top view for the two different methods (right).

Table 4
DIC camera characteristics.

Feature	Value
Resolution	4096 × 2160
Frame Rate	32
Megapixel	8.9
Chroma	Mono
Sensor	Sony IMX267
Pixel Size	3.45
ADC	12-bit

side along with fiducials to define a frame of reference during the DIC processing.

2.1.6. Sensors

Data was logged throughout all tests from load cells in the actuators, extensometers, and 20 sets of strain gauges at five different positions across the span of the blade. The linear and rosette strain gauges (type FRA-3-350-11, 350 Ω) had 5 % and 3 % strain limits, respectively. As shown in Fig. 8, the rosette strain gauges were placed 0.8 m apart, at three different y-coordinate locations on the pressure side and one on the suction side. Fig. 8(a) shows the equivalent distance from the first saddle to each of the five stations in terms of a chord length ratio. The pressure side of the blade is depicted in Fig. 8(b) where only one set of

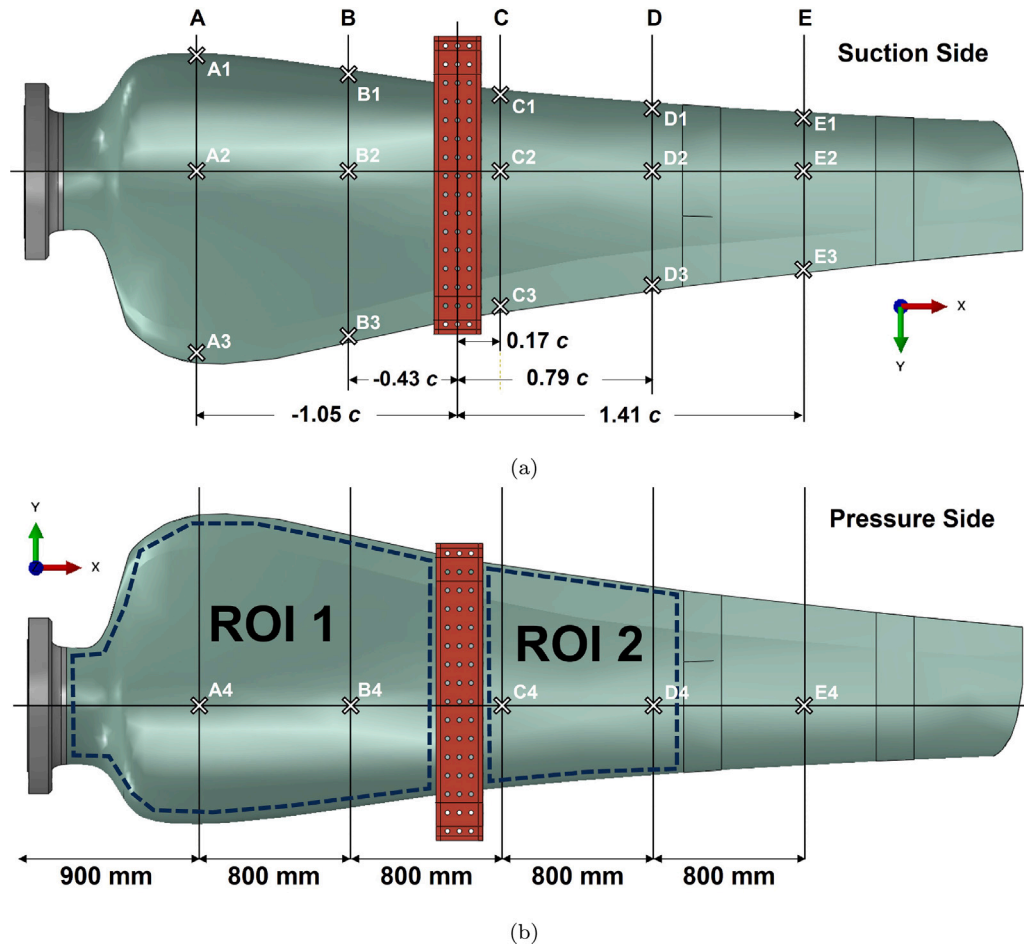


Fig. 8. Regions of interest (ROIs) for image acquisition and rosette strain gauge locations at (a) the suction side, (b) the pressure side of the blade. Total blade length: 5.25 m.

strain gauges located along the neutral x -axis is installed to minimise the interference with the DIC image acquisition.

2.2. Preload formulation

The main purpose of a clamped saddle system is to provide enough stability that the blade or the wooden saddle do not fail in bearing compression or slip during a test. The literature does not provide information regarding calculation of the clamping force required to fix the saddles. Current practices involve estimating the preload based on the target load at each saddle; thus, it does not consider the blade geometry, mechanical properties, or bolt characteristics.

However, a few different approaches can be adapted to the saddle-blade system considering a bolted joint design methodology. For the most simplified calculation, the assumption is that the preload should reach a maximum value before the clamping force damages any of the components within the system. In that case, the main limiting factor is the yield strength of the bolt connecting both sides of the saddle, as Eq. (1) shows:

$$F_{PL} = LF \cdot \sigma_y \cdot A_t \quad (1)$$

Where LF is a load factor between 0.7 and 0.9, σ_y is the yield strength of the bolt, and A_t is the tensile stress area of the bolt, which considers the area reduction caused by the threads, which is often provided by the manufacturer. As torque wrenches are the most used tools to achieve a predetermined preload, Eq. (2) is often used to estimate the torque required to meet the preload force calculated in Eq. (1) (Budynas et al., 2011):

$$T_{PL} = \left[\left(\frac{d_m}{2D} \right) \left(\frac{\tan \lambda + f \cdot \sec \alpha}{1 - f \cdot \tan \lambda \cdot \sec \alpha} \right) + 0.625 f_c \right] \cdot F_{PL} \cdot D \quad (2)$$

Here T_{PL} is the applied torque, d_m is the average of the major and minor diameters, D is the nominal bolt diameter, λ is the lead angle of the thread, f is the coefficient of thread friction, α is the thread angle, and f_c is the coefficient of collar friction. The term inside the brackets is often referred as the nut factor K_n , which typically has a value of 0.2 but varies depending on the application (Budynas et al., 2011). The existence of friction and shear between the joint parts and the bolt can reduce the accuracy of this approach by as much as 50% (Stikvoort, 2019).

The most accurate approach can be achieved by measuring the elongation of the bolt as the preload is introduced. However, given the size and conditions of this experiment, this method requires more robust equipment such as ultrasonic displacement sensors or bolt tensioners. Bolt tensioners are specialised tools that can achieve precise and controlled bolt tension. Their operation involves the application of hydraulic pressure to extend the bolt, resulting in a specific preload. A pure tensile load removes factors such as friction, shear, and lubrication from the uncertainty variability. Each tensioner manufacturer provides a relationship between the total hydraulic pressure applied and the desired bolt preload, such as Eq. (3) (Stikvoort, 2019):

$$P_h = \frac{F_{PL}}{A_h} \cdot \left(1.01 + \frac{D}{G} \right) \quad (3)$$

Where P_h is the corresponding hydraulic pressure for a desired bolt preload, A_h is the hydraulic pressure area, and G is the bolt grip length. The term in brackets is called the load transfer factor (LTF) and can be replaced by a factor which must be determined experimentally for more accurate results. Eq. (2) and Eq. (3) can be solved by matching F_{PL} with the actuator design load (95 kN), a common practice amongst the industry, to find the required applied torque T_{PL} and the hydraulic

pressure P_h . The impact of different preload values was not studied for this research as it could damage the test equipment or the blade before experimentation.

2.3. Digital image correlation (DIC)

2.3.1. Signature identification

DIC is an optical technique that provides a non-intrusive method for measuring deformation on the outer surface of a structure. It has shown accurate results on full-scale wind turbine blades (LeBlanc et al., 2013) and is currently used as a validation tool in tidal blade tests. To accurately measure displacement on a specimen's surface, the image is divided into subsets of pixels, and sum of squared difference (SSD) criteria, such as the zero mean normalised sum of squared differences (ZNSSD) chosen for this study in Eq. (4), are used to maximise the cross correlation of a subset between images (Tong, 2005):

$$C_{ZNSSD}(f(x_i, y_i), g(x'_i, y'_i)) = \sum \left(\frac{\bar{f}_i(x, y)}{\sqrt{\sum \bar{f}_i^2(x, y)}} - \frac{\bar{g}_i(x', y')}{\sqrt{\sum \bar{g}_i^2(x', y')}} \right)^2 \quad (4)$$

Where $\bar{f}_i(x, y) = f_i - \bar{f}$, $\bar{g}_i(x', y') = g_i - \bar{g}$; $f(x, y)$ are the grey values of the reference image, and $g(x', y')$ are the pixel values of the deformed image. When the correlation function C is close to one, the subset has been identified in the deformed image. Under real conditions, images contain noise coming from various sources, and the ZNSSD function has an advantage over other correlation functions as it is insensitive to both offset and scale changes of the subset intensity. However, the correlation function will never be a perfect match, and interpolation techniques must be used.

2.3.2. Strain field calculation

The strain calculation during DIC correlation is based on the definition of the deformation gradient $\mathbf{F}$ in Eq. (5). It can be expressed in matrix form for plane-stress conditions (Chu et al., 1985), as DIC cannot measure the through-thickness component $\partial/\partial Z$.

$$\mathbf{F} = \frac{\partial \mathbf{x}}{\partial \mathbf{X}} = \mathbf{I} + \frac{\partial \mathbf{u}}{\partial \mathbf{X}} = \begin{bmatrix} 1 + \frac{\partial u}{\partial X} & \frac{\partial u}{\partial Y} \\ \frac{\partial v}{\partial X} & 1 + \frac{\partial v}{\partial Y} \end{bmatrix} \quad (5)$$

Where $\mathbf{X}$ is the material coordinate system, x is the spatial coordinate system, $\mathbf{I}$ is the identity tensor, and $\mathbf{u}$ is the displacement vector. As the displacement vectors u, v are very susceptible to noise, the differentials used to solve the deformation gradient can magnify these fluctuations. Thus, a local polynomial fitting is often used to determine the displacement measurements analytically, such as the bilinear Lagrange polynomials, which turn Eq. (5) into the following:

$$\mathbf{F} = \begin{bmatrix} 1 + b_u + d_u Y_0 & c_u + d_u X_0 \\ b_v + d_v Y_0 & 1 + c_v + d_v X_0 \end{bmatrix} \quad (6)$$

The strain window previously discussed considers a discrete number of displacement data points that can be used in the spatial domain to calculate the deformation gradient $\mathbf{F}$. Furthermore, the displacement data points are a function of the subset and the step size, parameters that influence on the final resolution and noise level of the strain field. As strain magnitudes are not expected to be larger than 10 % for all the materials involved in the experiment, any of the strain formulations will show similar results. We therefore choose the Hencky strain formulation, usually referred as true strain or logarithmic strain, from Eq. (7).

$$\mathbf{H} = \ln(\mathbf{U}) = \ln(\sqrt{\mathbf{F}^T \cdot \mathbf{F}}) \quad (7)$$

Table 5

Strain gauge noise levels under no-load conditions in $\mu\text{m m}^{-1}$.

Blade Side	SG ID	Dist. c	ϵ_{xx} $\mu\text{m m}^{-1}$	ϵ_{yy} $\mu\text{m m}^{-1}$	ϵ_{xy} $\mu\text{m m}^{-1}$
Suction	A1	-1.05	5.27 ± 0.42	5.56 ± 0.45	0.96 ± 0.48
	B1	-0.43	5.49 ± 0.57	11.49 ± 0.50	0.09 ± 1.20
	C1	0.17	6.73 ± 0.96	0.69 ± 1.08	3.35 ± 1.16
	D1	0.79	10.46 ± 0.87	12.35 ± 0.52	3.14 ± 1.85
Pressure	A4	-1.05	1.71 ± 0.44	1.48 ± 0.42	1.52 ± 0.37
	B4	-0.43	2.62 ± 4.11	10.50 ± 1.46	2.71 ± 1.08
	C4	0.17	7.17 ± 1.53	2.92 ± 1.73	2.37 ± 2.11
	E4	1.41	7.75 ± 1.50	0.83 ± 0.51	2.55 ± 1.55

2.3.3. DIC performance analysis

To prove the reliability of DIC on such large structures, the DIC post-processing algorithm described in the previous section required a sensitivity analysis. This helps to define the most optimised parameters to correlate strain values from the DIC with the real values at the surface of the blade. The step size, subset size, shape function, and strain window were varied to determine the noise threshold from the image acquisition. The strain values calculated from these different parameters were grouped based on their subset size. Fig. 9 highlights three of the most optimised subset sizes for both ROIs.

The best parameters to match the strain values from the strain gauges were achieved with high VSG sizes. Q8 interpolation shows a very sparse optimisation, caused by an overfit behaviour where noise gets attenuated by the differentiation. Thus, Q8 was not chosen for further analysis as the standard deviation would be high enough to mistake the strain distribution across the surface as noise. Thus, a VSG of 183 pixels with the following parameters has been chosen for the cameras targeting ROI 1: a subset size of 21, a step size of 9, and a strain window of 19. As for ROI 2, the VSG is made of 195 pixels with a subset size of 27, a step size of 12, and a strain window of 15.

Such validation can be observed in Fig. 10a, where all three phases of the plywood saddle test are related. The DIC values were obtained by averaging the data points surrounding the location of the strain gauge. Different radius for this surrounding area were considered, of which a radius of 200 pixels was the most optimised.

3. Results & discussion

3.1. Torque wrench vs hydraulic pressure clamping method comparison

Due to safety concerns associated with the torque-wrench method, conducting a fatigue test with personnel in close proximity to the blade was deemed unfeasible. Consequently, the data collected for this particular test solely pertains to the clamping and static phase and was obtained from sensors and DIC images.

Data logging began with the saddles already clamped to the blade from previous static and fatigue tests. Subsequently, bolts were gradually loosened until a 1 mm gap became visible between the saddle and the blade. Data was collected at this stage when the blade and the clamp system are not in contact. This data was used to calculate the strain gauge noise levels, effectively defining a boundary for negligible readings for the following stages of the test. The noise levels of the most relevant strain gauges are presented in Table 5.

Bolts were then re-tightened to the maximum permissible value by the torque wrench, which in this case was 81.4 N m. Fig. 11(a) illustrates that, during this loosening process, the strain on the pressure side of the blade decreased, resulting in a strain reduction of 115 microstrain ($\mu\text{m m}^{-1}$) at a distance of -0.43 c from the saddle. The same behaviour can be observed on the other side of the saddle in Fig. 11(b), however, at a lesser extent as the maximum strain observed in strain gauge B1 of $10 \mu\text{m m}^{-1}$ is considerably low and close to the strain gauge noise threshold. This discrepancy suggests a variation in the disturbance,

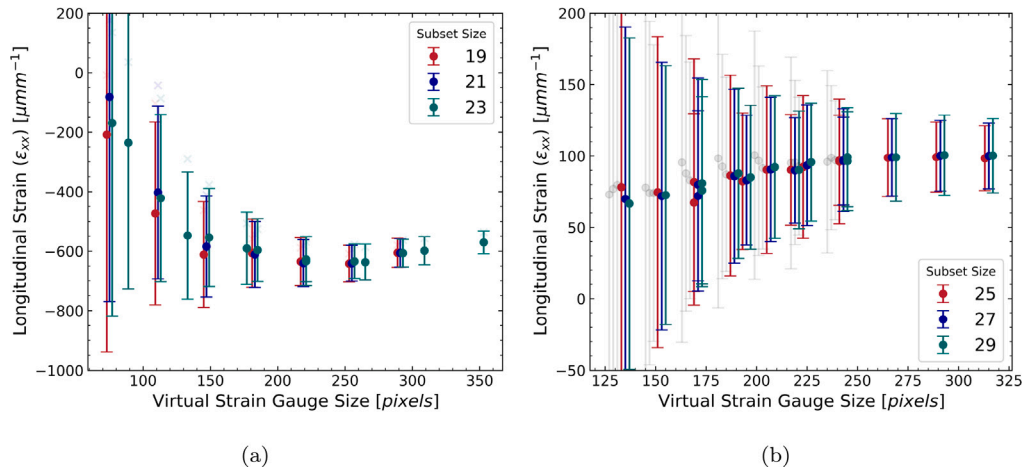


Fig. 9. DIC strain sensitivity for different subset sizes at: (a) ROI1, (b) ROI2. Grey values are suboptimal subset size candidates.

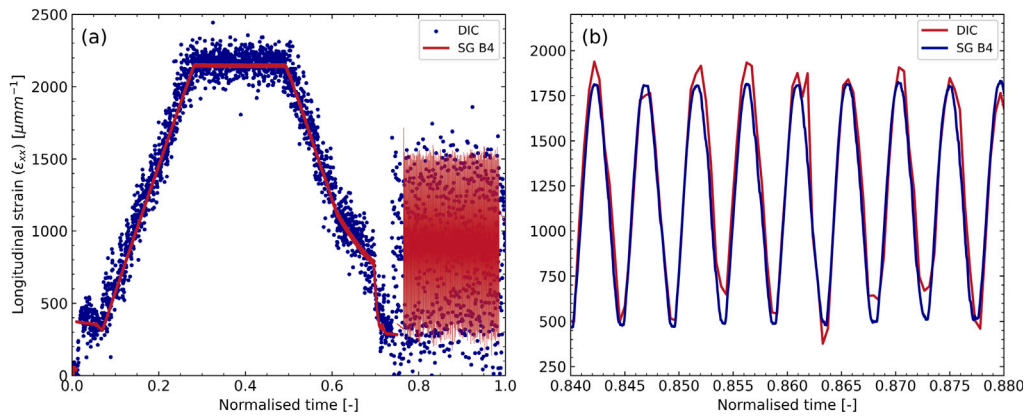


Fig. 10. Longitudinal strain correlation between DIC and strain gauge B4 located at $-0.43c$ from the saddle: (a) Full static and fatigue test (Test Case 3), (b) last 10 cycles of a fatigue test (Test Case 2).

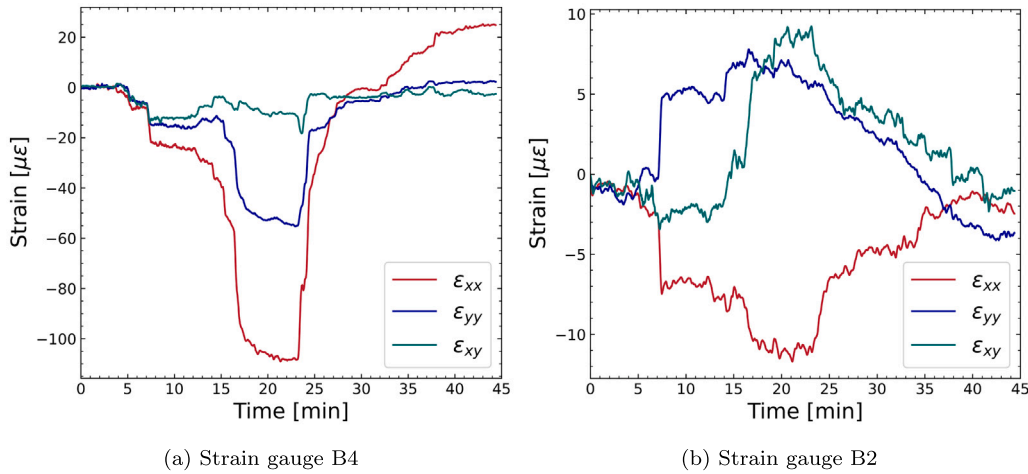


Fig. 11. Strain measurements at $-0.43c$ from the saddle for location: (a) B4, (b) B1, for the torque wrench method. At min 7, the first bolt is loosened until min 17, when the saddle loses all the pressure on the blade. Bolts are then re-tightened at min 23.

thus different values related to the chord length could be used as safe boundaries for each side of the blade independently.

When considering the overall analysis for the torque-wrench method presented in Fig. 11, it becomes evident that the final strain readings do not align with the initial balanced strain. This inconsistency could be attributed to a reduction in the clamping force over time. As a system

composed of multiple components, this decrease could be due to factors such as embedment, a creep effect on the MDF or the steel frame, or a combination of these factors.

When examining the strains at a distance of $-1.05c$ from the saddle (locations A2 and A4), the influence of the saddle becomes even more diminished. The maximum longitudinal strain values are measured at

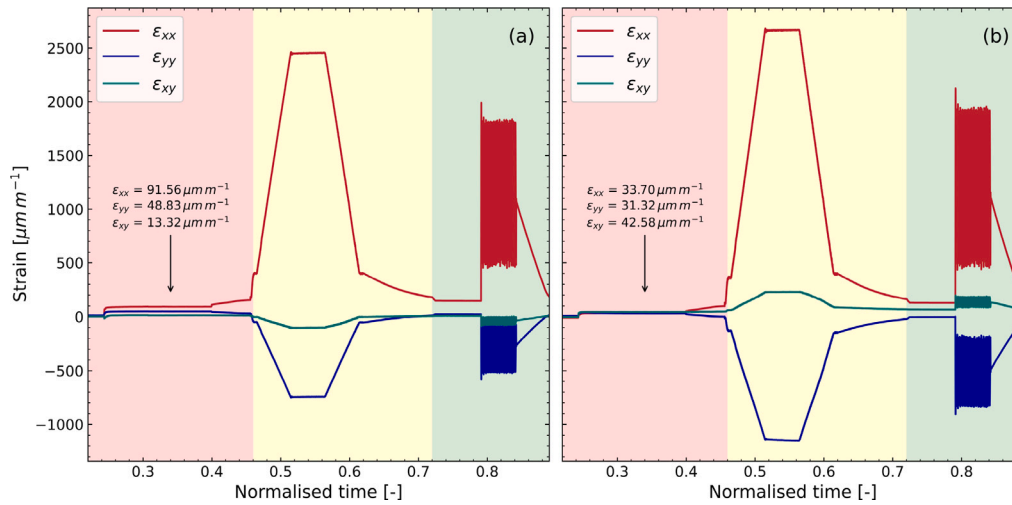


Fig. 12. Strain values on the centreline on the pressure side of the blade during: clamping (red), static test to 95 kN (yellow), and 100-cycle fatigue test (green). (a) strain gauge B4 at $-0.43c$ from the saddle and (b) strain gauge C4 at $0.17c$ from the saddle.

$(6.50 \pm 0.56) \mu\text{m m}^{-1}$ for the pressure side and $(5.30 \pm 0.38) \mu\text{m m}^{-1}$ for the suction side. These values can be regarded as negligible since they are within their noise limits from Table 5 and also account for less than 1 % of the strain values obtained at those locations during static and fatigue tests.

By referring to Eq. (2) and assuming a commonly used nut factor of 0.2, an estimation can be made that each bolt is subjected to a clamping force of approximately 16.96 kN. Notably, no signs of slip were observed during the earlier tests when a similar clamping force was applied. An estimated clamping force of 101.75 kN is acting on the system. This force corresponds to 1.07 times the design load for the first actuator during a static test, however, a more precise nut factor could equalise both forces.

As for the hydraulic method, the bolt tensioners were positioned on both sides of the top steel frame, and the hydraulic pressure was controlled in parallel. The hydraulic pressure could reach 150 bar for the clamping stage, equivalent to 84 kN, estimated using Eq. (3). This clamping phase was followed by a 95 kN static load and a short fatigue test conducted at a frequency of 0.5 Hz and a peak load of 75 kN, as shown in Fig. 12.

The maximum strain value recorded during a static test occurred at location C4, $0.17c$ away from the saddle, reaching a value of $(2667.43 \pm 3.01) \mu\text{m m}^{-1}$ in Fig. 12a. By comparing this to the maximum value observed during the clamping phase, which is approximately $33.7 \mu\text{m m}^{-1}$, it is evident that the preload represents only 1.3 % of the total load. Furthermore in Fig. 12b, for the longitudinal strain at B4 or $-0.43c$, the contribution from the clamping phase increases to 3.7 %.

After observing data pertaining to the individual clamp mechanisms, both methods were also compared in terms of displacement and strain dissipation as a function of distance from the saddle. Fig. 13 shows the mean displacement and longitudinal strain of the blade skin at different distances away from the saddle. The figure is divided in two zones, the skin zone and the spar zone. This distinction is made as the stiffness of these two zones are different underneath the surface. The spar zone can be as much as 10 times thicker than the skin zone. The two zones show similar displacement behaviour when comparing both clamping methods in Fig. 13a. The manual method shows a more sparse distribution towards the root of the blade, with high values closer to the saddle for both zones. This can be attributed by the cross loading factor induced from tightening one bolt at a time, which causes pressure to be lost on previously tightened bolts. On the other hand, the hydraulic method shows a less sparse distribution as long as all bolts are tensioned in parallel, with displacements approaching zero at a distance of $-0.45c$ onwards.

Longitudinal strains show the same overall trend of sparsity as displacements, however zones behave differently as a result of the difference in stiffness. The use of the torque wrench as a clamp mechanism results in sparse strains that are highly dependent on the order of bolt tightening. The strain measurements at location B4 fall within the range of values measured using both bolt tightening techniques from the DIC, but are more closely aligned to the tighter range of hydraulic bolt-tensioning results. The observed normalised strains indicate that the structure of the blade has a greater impact than the clamping force when considering an exclusion zone threshold value. Therefore, a single threshold value for all tidal turbine blades cannot be established. Instead, achieving a consistent pressure distribution should be the priority. Overall, bolt tensioners show a more consistent clamp pressure through different iterations, and it should be considered as the desired method to clamp a thick composite blade to the load mechanism in an laboratory testing facility.

The influence of the blade's components on the distribution of clamping pressure is further illustrated through full-field DIC maps. Fig. 14 showcases the out-of-plane displacement component, denoted as W , subsequent to the application of the target pressure on both M36 bolts. The region situated between the two shear webs experienced buckling, leading to a similar behaviour between the shear web and the trailing spar. This could lead to early skin-spar debonding causing blade failure within the exclusion zones. The presence of a rib, indicated by the horizontal dashed line in Fig. 14, exacerbates this behaviour as it constrains the displacement decay.

During a fatigue test shown in Fig. 15(a), the troughs showed longitudinal strains of $94.40 \mu\text{m m}^{-1}$ and $451.56 \mu\text{m m}^{-1}$ at $-1.05c$ and $-0.43c$, respectively. The strain experienced in those locations during clamping contributed to 2.3 % and 20.3 % of the minimum strain values in a fatigue test, respectively. The most relevant strain comparisons between the three phases of a test are displayed in Table 6. The effects on the clamping can be seen on each of the sides of the blade in Fig. 15, highlighting the fact that their regions available for analysis might differ.

The strain needed to create the first crack in the composite matrix can be used as a reference value to determine whether the blade's stiffness is altered before a test is run. Based on Garrett and Bailey (1977) and a ply thickness of 1 mm, the initial cracking strain is 0.49 %. The strain values during the clamping phase can then be normalised to this strain value and are shown in Fig. 16 for both sides of the blade. It can be observed that the pressure side of the blade experiences higher strains closer to the saddle but match the suction side values at approximately $0.75c$. This observation agrees with the conservative

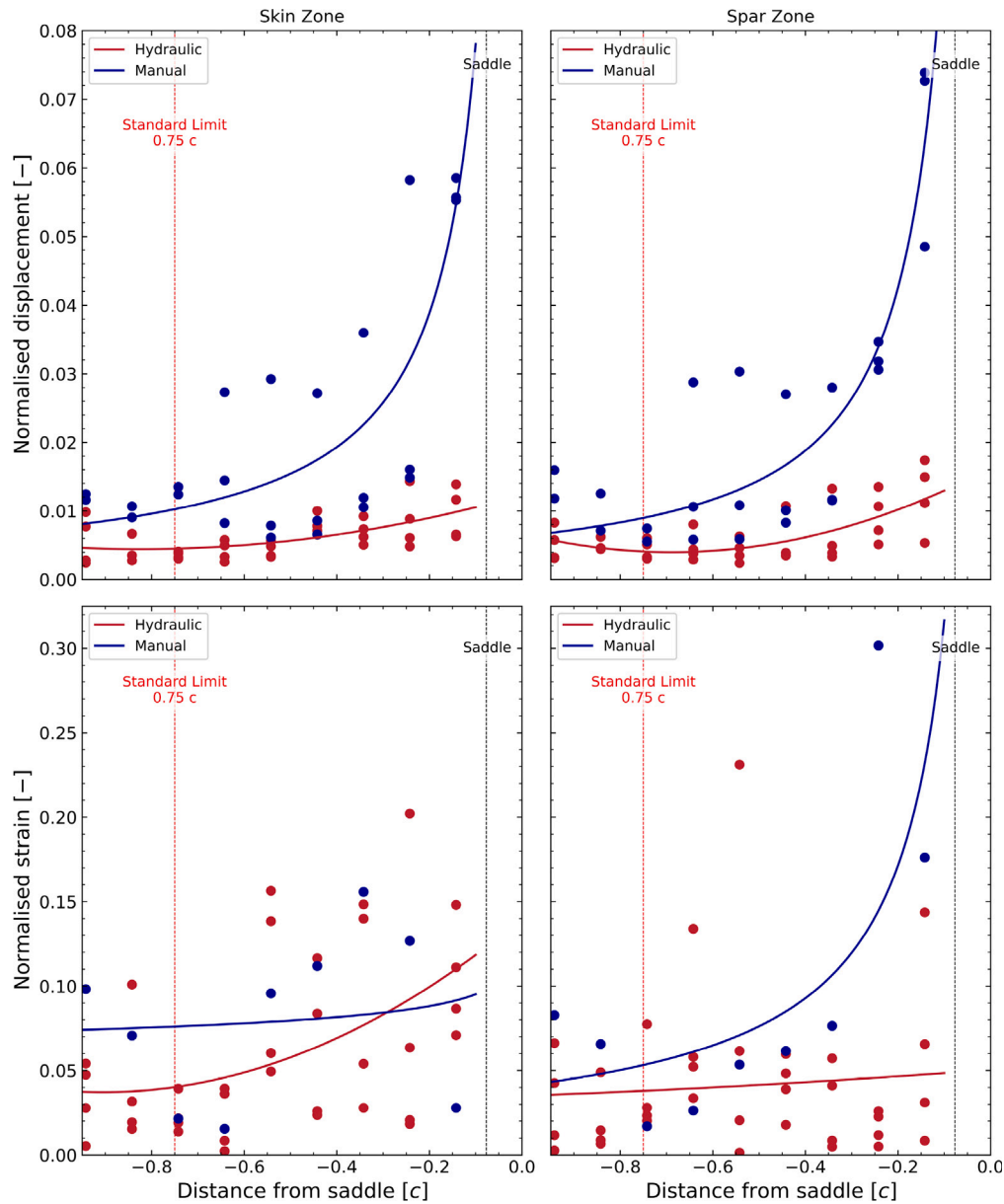


Fig. 13. Clamping method comparison at ROI1 (pressure side of the blade) for: (a) displacement magnitude (top figures) and (b) longitudinal strain (bottom figures).

Table 6

Longitudinal strain comparison between all three phases of a test using the hydraulic method and an MDF saddle in $\mu\text{m m}^{-1}$. Strain gauge D4 not available as not enough data was collected from this sensor.

Blade Side	SG ID	Dist. c	Clamp ϵ_{xx}	Static		Fatigue (Min)	
				ϵ_{xx}	%	ϵ_{xx}	%
Suction	A1	-1.05	0.9 ± 0.40	-694.7	0.7%	-118.6	0.8%
	B1	-0.43	15.6 ± 0.07	-2093.2	0.5%	-313.1	4.8%
	C1	0.17	24.9 ± 0.02	-2332.1	0.7%	-333.2	7.0%
	D1	0.79	1.7 ± 0.44	-2309.4	0.4%	-328.0	0.5%
Pressure	A4	-1.05	-0.4 ± 0.02	613.8	0.4%	94.4	0.4%
	B4	-0.43	96.3 ± 2.90	2453.2	3.7%	451.6	21.3%
	C4	0.17	41.3 ± 1.97	2667.4	1.3%	434.6	9.5%
	E4	1.41	2.3 ± 0.57	1941.7	0.1%	260.0	0.9%

approach given by the standards to consider three quarters of the chord length for the whole structure. It is noted that the values for

the pressure side are still very small, with values at $0.5c$ being below the initial cracking strain value. Reaching this reference value does not strictly correlate to failure, as it will also depend on the ply orientation.

3.2. Comparing saddle materials

MDF and plywood were tested and compared to assess the structural performance of the saddle. A hydraulic mechanism was used to tighten the bolts, followed by a static test with a peak load of 95 kN. Only three clamping tests were performed on the MDF and six on the plywood saddle, due to the high costs associated with these large-scale experiments. Fig. 17 displays the deformation behaviour observed at ROI 2, captured by the third set of cameras for both the MDF and the plywood experiments. This region is affected by the second saddle coming into contact with the blade, resulting in the overlapping of clamping forces from both fixtures. However, the presence of a rib positioned between the two saddles effectively restricts the extent of the disturbed area generated by the introduction of the load at these points. A maximum displacement of (0.83 ± 0.02) mm spreads from the

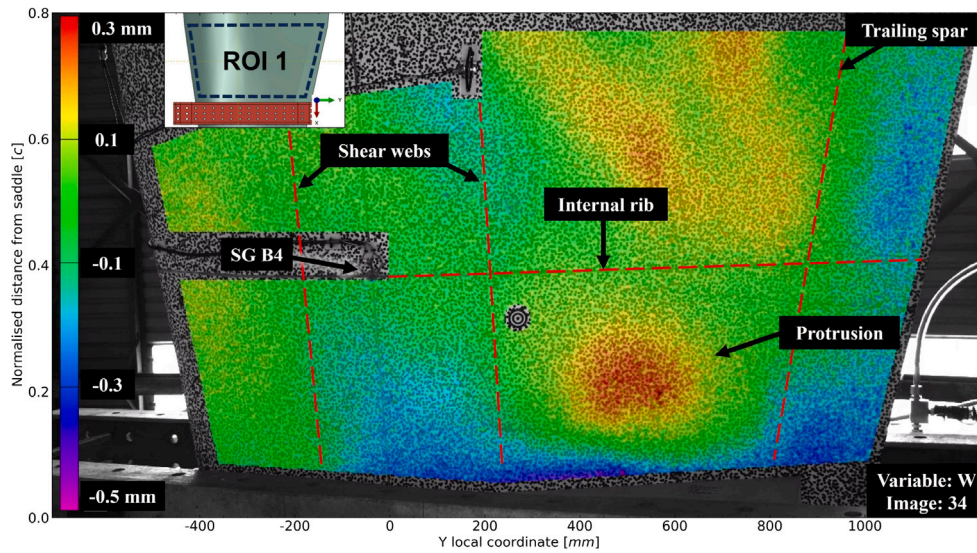


Fig. 14. Out-of-plane displacement component in ROI 1 (pressure side) after a hydraulic pressure of 150 bar is applied to the bolt tensioners, in mm.

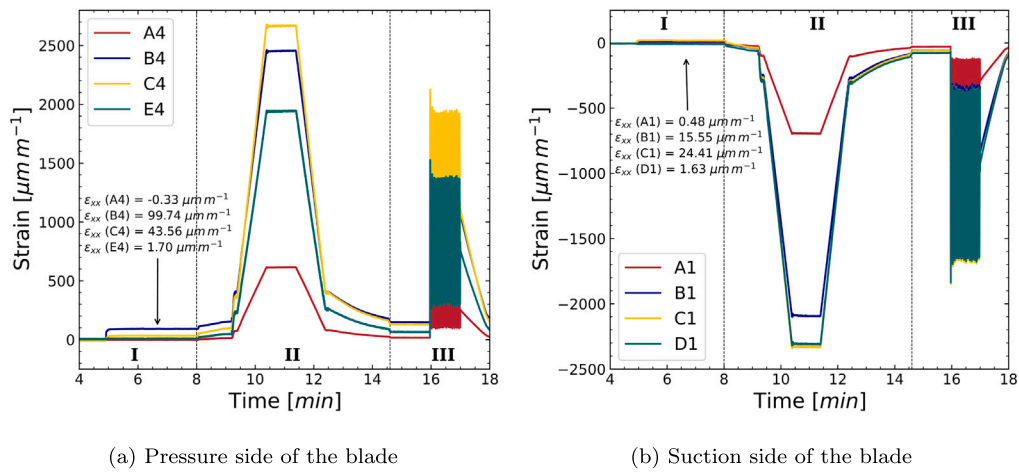


Fig. 15. Longitudinal strains at the neutral z-axis during a full test campaign: clamping (I), full static test to 95 kN (II), 100-cycle fatigue test (III).

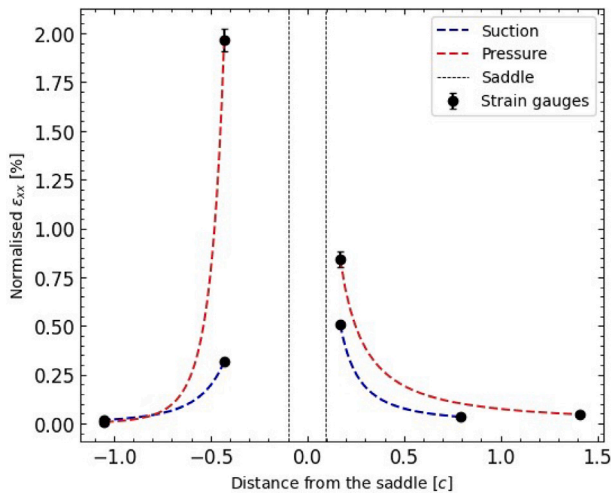


Fig. 16. Clamping longitudinal strain normalised along the neutral z-axis of the blade.

MDF saddle, and (0.48 ± 0.15) mm for the plywood saddle test. Blue and darker areas show displacements of less than 0.15 mm that are close to the noise levels. Based on these observations, the dissipation of the clamping stage preloads for ROI 2 is significantly faster than that of ROI 1. Preload effects settled between 0.25 c to 0.36 c for ROI 2.

The strain field computation was done using the DIC parameters obtained from the performance analysis, a Q4 interpolation, and the Hencky strain formulation from Eq. (7), employing a window tolerance of 50%. In Fig. 18, the longitudinal strain observed after 160 bar is applied to the saddle bolts for each of the saddle configurations, MDF and plywood. As the blade is very stiff, the range of strain values in this region is very low and clear differences between the two saddle materials cannot be identified. The MDF configuration shows an average strain value of $(5.07 \pm 3.96) \mu\text{m m}^{-1}$, whereas the plywood shows a slightly larger average of $(16.99 \pm 27.21) \mu\text{m m}^{-1}$. However, both values are within noise levels from the DIC processing algorithm.

A similar procedure was followed for the ROI 1, where the strain field better agrees with the measurements obtained from the strain gauges, as depicted in Fig. 19. The highest strain fluctuations are closer to the saddle as expected for both compression and tension longitudinal strains. The maximum strain value measured was $(540.6 \pm 71.3) \mu\text{m m}^{-1}$ and the minimum was $(-452.4 \pm 114.4) \mu\text{m m}^{-1}$. Both peaks are located no further than 0.45 c from the saddle. Other patterns can be observed

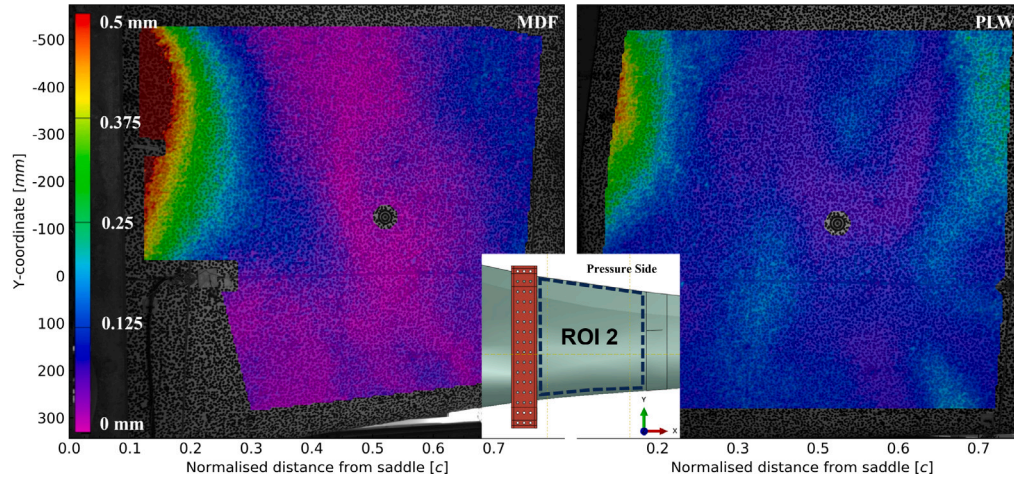


Fig. 17. Overall displacement comparison at ROI 2 on the pressure side of the blade during clamping for the MDF saddle (left) and the plywood saddle (right), in mm.

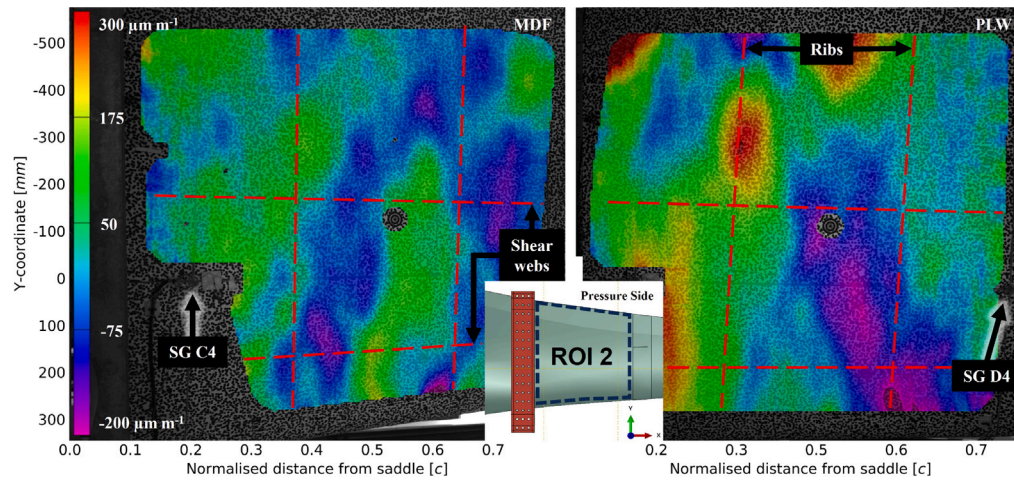


Fig. 18. Longitudinal strain comparison during clamping between the MDF saddle (left) and the plywood saddle (right) on ROI 2, in $\mu\text{m m}^{-1}$.

at 0.45 c and above, however, the strain values in this region are within noise levels.

Table 7 summarises a comparison of the preload strain contribution towards the static test between the two saddles. Strain gauges on both the pressure and suction sides of the blade show higher strain values when the MDF saddle is used. This may be due to the non-uniform pressure distribution caused by internal blade components like shear webs or ribs, which create localised stress concentrations. In these areas, the MDF deforms plastically, redistributing pressure across other regions. In contrast, the higher stiffness of plywood keeps stress concentrations more localised. The MDF saddle was also observed to have critical damage after all tests were complete, rendering them unsuitable for any future experiments. This critical damage included onset of delamination as well as compression failure on the free edges that were in contact with the blade. On the other hand, the plywood saddle had no visible signs of delamination nor crushing on any of the sheets.

4. Conclusions

Current standards lack detailed specifications and practices for the introduction of load in full-scale tidal turbine blades tests. As these standards need to be followed for blade accreditation, one of the most constraining guidelines is assessed here for the first time, by

Table 7

Normalised clamping longitudinal strain, expressed as a percentage of the peak static strain.

Location	SG ID	Distance	MDF	Plywood
Pressure	A4	-1.05 c	$0.06 \pm 0.01\%$	$0.008 \pm 0.05\%$
	B4	-0.43 c	$3.93 \pm 0.12\%$	$2.16 \pm 0.09\%$
	C4	0.17 c	$1.55 \pm 0.07\%$	$0.10 \pm 0.02\%$
Suction	A1	-1.05 c	$0.13 \pm 0.06\%$	$0.009 \pm 0.05\%$
	B1	-0.43 c	$0.75 \pm 0.01\%$	$0.59 \pm 0.03\%$
	C1	0.17 c	$1.07 \pm 0.02\%$	$0.93 \pm 0.04\%$

comparing two clamping methods and two saddle materials. For this purpose, a full-scale composite tidal turbine blade was tested where strain measurements and Digital Image Correlation (DIC) data were used to quantify the strain contribution from the clamping stage to the strain measurements during static and fatigue tests.

The two clamping methods compared were a manual torque wrench and a hydraulic bolt tensioner. Both methods revealed the impact of internal structures in the blade on the strain dissipation. This comparison also showed the different structural response between the pressure and the suction side of the blade. The cross-loading factor induced by the manual method increased the effects of the preload, whereas the hydraulic tensioners achieved dissipation 585 mm away from the saddle. Observations prompt the reduction of the exclusion zone boundary

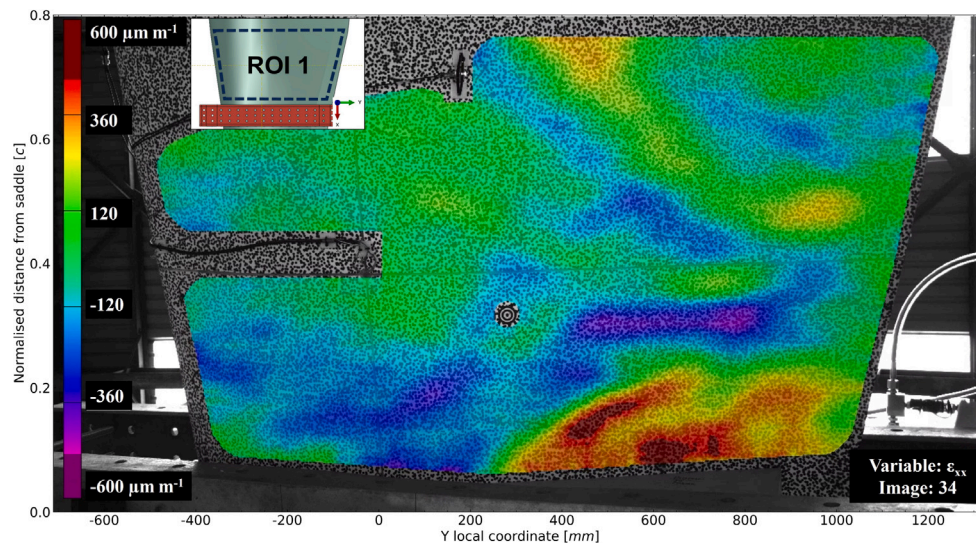


Fig. 19. Longitudinal strain at ROI 1 on the pressure side of the blade after 150 bar applied to the bolt tensioners.

from 0.75 to 0.45 times the chord length. Although this proposition would only apply to this specific tidal blade, this study opens the opportunity for other testing facilities to reduce the spacing between saddles methodically.

The material used for the saddle was compared using MDF and plywood. The MDF showed a higher strain contribution onto the blade surface, as it distributes the pressure more evenly and thus spreads to less stiff components of the blade. Plywood had a higher reliability under high static loads and proves to be a better option than MDF, which exhibited high levels of structural damage at the end of the experiments. Further investigations are needed to reduce the uncertainties, to help establish a framework for determining the required clamping force and the optimised number of load introduction points.

CRediT authorship contribution statement

Miguel A. Valdivia-Camacho: Conceptualization, Data curation, Formal analysis, Investigation, Methodology, Software, Validation, Visualization, Writing – original draft. **Sergio Lopez Dubon:** Data curation, Methodology, Project administration, Resources, Supervision, Writing – review & editing. **Fergus Cuthill:** Data curation, Investigation, Methodology, Resources, Software, Writing – review & editing. **Marek J. Munko:** Investigation, Writing – review & editing. **Edward D. McCarthy:** Project administration, Resources, Supervision, Writing – review & editing. **Parvez Alam:** Methodology, Supervision, Writing – review & editing. **Conchúr M. Ó Brádaigh:** Conceptualization, Funding acquisition, Methodology, Project administration, Resources, Supervision, Writing – review & editing.

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.

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Data availability

Data will be made available on request.

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