Engineering Failure Analysis 140 (2022) 106590

Contents lists available at ScienceDirect

Engineering Failure Analysis

journal homepage: www.elsevier.com/locate/engfailanal

Comparative study on loosening of anti-loosening bolt and standard bolt system

Øyvind Karlsen [a,b,*], Hirpa G. Lemu [a]

[a] Faculty of Science and Technology, University of Stavanger, Norway
[b] Bondura Technology AS, Norway

ARTICLE INFO

Keywords:
Self-loosening
Bolt-loosening
Pre-loaded bolt
Anti-loosening
Fasteners
Vibration

ABSTRACT

In general, bolted connections are exposed to vibrations or repeated over-loads that could lead to self-loosening due to loss of preload. Pre-tensioned bolts in ring flanges are critical parts in Offshore Floating Wind Power Systems, and normally a certain percentage of the installed bolt connections are checked and re-tightened every year. This re-tightening is often done at a high cost and a short weather window due to strong winds and high waves. In this paper, three bolt dimensions (M20, M30 and M42) of the anti-loosening bolt system have been tested. The M30 and M42 bolt systems were preloaded and exposed to transverse oscillating loading, and the loss of preload as a function of load amplitude and number of cycles were measured and compared to standard bolts of HV type, exposed identically. The tested novel bolt system has shown superior capacities to withstand self-loosening, compared to standard bolts.

1. Introduction

Hundreds of billions of fasteners are produced around the world every year [1], and in almost any mechanical connection, there will be solutions for how to transmit power from one part to the other, in the most efficient way, and the transmitted power will often be of mechanical type. There exists a wide range of technical solutions for how to transmit such power, and for mechanical systems, several methods can be mentioned including: riveted and interference fitted pins, pre-tensioned, open-hole bolts and expanding pin solutions [2-4] as standard choices, depending on type of exposure, loads, sizes, and preferences. In general, increasing transmitted power in a mechanical system will result in increased exposure of the mechanical connections to higher loads, with increased probability for damages or malfunctions as a result, especially at the contacting surfaces.

Relative movements between connected parts, typically because of vibrations or over-loads, can result in fretting related problems, like fretting wear, corrosion, and fatigue [3]. Many studies have been performed regarding fretting, and how to prevent the fretting related problems in joints [5-11]. The most important factor in relation to reliability and functionality of bolted joints is the clamping force, where both friction and torque are important factors [12].

Vibrations in mechanical systems can result in loss of clamp forces in preloaded bolts, whose effect depends on vibration type and level, preload level, contact surface and size, reduction and loss of asperities, thread type and size, thread pitch, lubrication of threads, etc. Loss of clamp force in bolts in a bolted connection can easily result in failure and damages of both the bolts, the connection, and the mechanical system itself. After fatigue, self-loosening of bolts is one of the most common causes of failure in bolted connections.

* Corresponding author at: Faculty of Science and Technology, University of Stavanger, Norway.
E-mail address: oyvind.karlsen@uis.no (Ø. Karlsen).

https://doi.org/10.1016/j.engfailanal.2022.106590
Received 25 March 2022; Received in revised form 27 June 2022; Accepted 28 June 2022
Available online 3 July 2022
1350-6307/

Grabon et al. [12] indicated that surveys from the USA show that up to 23% of all car service problems were related to loosening fasteners, and even for new cars around 12% had this problem.

Standard preloaded bolts, with bolt head, shank and one or two nuts, are often applied when connecting huge flanges like the ring flange connections in wind turbine towers, illustrated in Fig. 1, and in many different constructions, heavy machineries, flange connections of tubes, etc. Such standard preload bolt solutions have a relatively low cost and are normally easy to install. Generally, there are two methods of generating the required pretension in bolts: (1) by torquing the nut directly by hand torquing tool or torquing tools with external energy source, or (2) by hydraulic tensioning of the shank [13,14] followed by a light torquing of the nut, pressure release and disconnection of the hydraulic tool thereafter. In the first case, an additional second nut can be applied and torqued. In both cases, with direct torquing or hydraulic tensioning, the nuts are exposed to vibrations and possible fatigue and fretting problems over time, which again can lead to self-loosening. The two nuts in such a situation would normally be working at the same identical bolt threads, with same shape, size and pitch and surface conditions. This means that the two nuts could loosen their grip together, and turn together, without the second nut having any specific locking effect. If the two nuts are identical, the locking nut would not have any specific locking effect or locking capabilities that the main nut doesńt have, and in addition it can be argued that the locking nut is weakening the locking effect of the main nut by reducing the contact pressure at the main nut threads.

Standard bolts and nuts are often made in strength class 8.8, 10.9 or 12.9 with typically zinc plated, hot dipped galvanized (HDG) surface protection, but also coatings with zinc and aluminium flakes like Geomet500®, Dacromet®, or Delta® are in use. All surface treatments will have a certain amount or level (height) of asperities that must be taken into consideration when analysing self-loosening behaviour of the bolt-nut system, in addition to relaxing and creep effects due to structural changes in the material, over time.

Re-tightening of standard bolts at critical positions, for instance, for floating Offshore Wind Power systems has become a high-cost activity that the system owners and operators both want and need to minimize. For safety reasons, all critical pre-tensioned bolt positions are normally checked yearly by re-tightening 10% of the bolts. These activities require a great number of resources, but at a limited time window due to operational limitations as weather and wave conditions, as investigated by Gintautas and Sørensen [15].

The first author of this paper has designed a new anti-loosening bolt solution where all three parts; bolt, main nut and locking nut, are designed to work together to reduce any self-loosening effects of the bolt assembly, and by that reduce or prevent the need for re-tightening pretensioned bolts in critical positions. Such a reduced need for re-tightening bolts would have a great cost-impact, especially at Offshore Wind Power sites, where the cost for re-checking and re-tightening is substantial.

Today's Offshore Wind Power systems require bolts up to M72 size on the ring flanges, and bolt sizes for future systems will reach M100 or bigger, typically of 10.9 or 12.9 quality class, typically with HDG surface protection. It is a known "problem" that the bolts must be checked or re-tightened after a few weeks or months due to plastic deformation of asperities and creep of the metal, where both effects could lead to turning of the nut and loss of tightening force consequently. Vibrations and oscillating loads in both axial and transverse directions will often lead to self-loosening of the bolted systems, and therefore a yearly re-tightening schedule of a certain amount of the bolts, typically 10%, is normally introduced. Vibrations tend to deform and reduce the height of asperities at the contact surfaces, and a high pre-load could lead to material yield at the first engaged threads. Both lead to reduced clamp force and increased risk of nut turning, which generate further clamp force loss and possibly total failure of the bolted system.

The anti-loosening bolt system, Vibralock® [16], illustrated in Fig. 2, contains a bolt with a bolt head, a shank, and threaded parts with two different diameters, and hence with different thread pitches. Two nuts, a main nut, and a locking nut, are used on the same

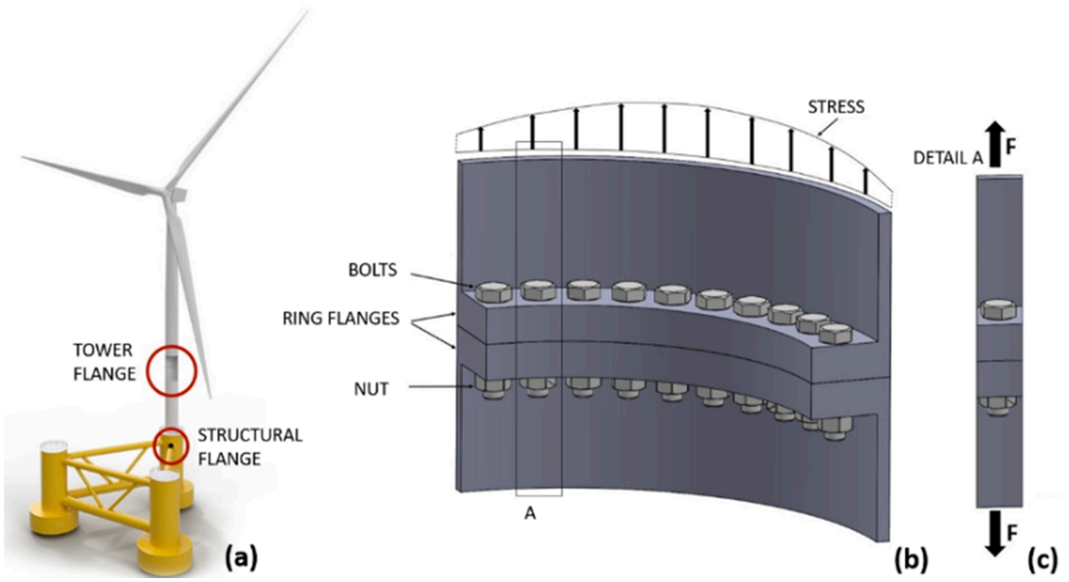

Fig. 1. Wind turbine tower, (a) floating system, (b) flange details, and (c) force directions.

bolt where the bigger diameter is for the main nut and the smaller diameter is for the locking nut. The two nuts will therefore have different thread pitches, with fine pitches on the smaller diameter, and the contact surface between the nuts has a conical contact area. These two nuts can never loosen and turn together, given the differences in pitches, where due to larger pitch the main nut (inner nut) would move faster axially in loosening direction than the locking nut, and therefore increase the contact pressure on the conical contact surface between them. This prevents the locking nut from turning. Due to the conical contact area between the two nuts, the locking nut will not be exposed to transversal loads and displacements relative to the main nut.

The aim of this study is to investigate how the anti-loosening bolt concept reacts and resists on external axial and transverse loads including loads due to vibrations. To conduct the study, two different experiments and one Finite Element Analysis (FEA) were conducted, all on different bolt sizes. The two experiments are designated as Test 1 and Test 2, where:

- Test1: M20 bolts were used, but without the bolt head, to make it possible to install and test two bolt systems at the same time. The aim of the test is to investigate whether the nuts were exposed to self-loosening during the *axial* dynamic loading, and if there were dimensional and surface changes at the contact surfaces, like the threads and nut cones.
- Test 2: M30 and M42 bolt systems were exposed to *traversal* dynamic loads and any loss of clamp load were measured. The FE analysis was performed to analyse the structural integrity of the experimental test, but on bigger pin diameter, M72.

2. Literature study

Many investigations have been conducted to get more knowledge about self-loosening of bolted joints, often based on studies by Junker [17] regarding criteria for self-loosening of fasteners under vibrations. Zhu et al. [18] proposed torque-preload formulas to estimate anti-loosening and control preload in threaded fasteners, and Zhang et al. [19] studied the roles of thread wear on self-loosening behaviour. Some of the studies have focused specifically on axial excitation on bolted joints [20-22], and others on transverse vibrations [1,19,23], but also torsional and prying loads are important, which all can lead to vibration-induced bolt loosening.

In general, it is commonly accepted practice that up to 85–90% of the input-torque in threaded fasteners is converted into heat when overcoming the two main friction torque components, and down to 10–15% contributes to increased preload in the bolt [12,18], if not sufficient or correct lubrication is applied. The two components that the friction torques need to overcome are: (1) between the bolt head/nut or bearing surface and (2) between male and female threads. Croccolo et al. [24] provided an experimental methodology to determine the friction coefficient in bolted joints, and Zhu et al. [18] proposed torque-preload formulas to estimate anti-loosening performance, and control preload.

In general, as Liu et al. [21] indicates, there are two basic loosening mechanism: (1) fastener elongation beyond its elastic limit at the threads, and (2) microslip on the thread and bearing surfaces. Vibration will increase the risk of wear on the threads and then also increase the loosening of the bolted joint, with risk of losing completely the clamp load. By experimental and numerical studies of bolted joints subjected to axial excitation, they found that the clamp force decreases rapidly at a *first stage* because of the cyclic plastic deformation, and at a *second stage* a slower decrease due to fretting wear [20,23,25-27]. The same conclusion was made by Zhang et al. [19].

At the first stage of self-loosening, there is no relative rotation between the bolt and the nut, but there could be a local cyclic plasticity near the engaged thread roots, typically the three first threads. Earlier studies by Guan et al. [28] and others show that the first engaged thread is the most loaded, and will therefore be the most damaged, with about 30% of the total load on the first threads, which results in a stress redistribution in the bolt, and consequently a gradual loss of clamping force. The second stage is characterized by relative rotation of the nut to the bolt. It has been found that, by increasing the pre-load, decreasing the load amplitude of the axial excitation, and lubricating with MoS2, the damages on threads will be smaller and the anti-loosening effect will increase [20,21]. Furthermore, finer threads or threads with smaller pitches are more resistant to loosening than coarse threads or larger pitches [18]. At higher tightening torques, which create higher preload/clamp load, the slippage at contact areas will be reduced and the sticking effect increase and thereby increasing the anti-loosening effect. It is a near linear relationship between the preload and the self-loosening

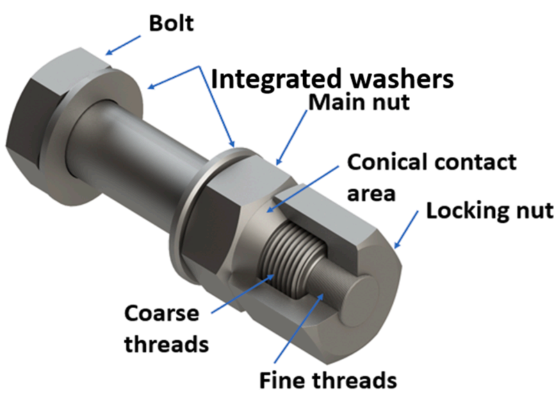

Fig. 2. Vibralock® anti-loosening bolt-nut system.

resistance, according to an experimental study performed by Jiang et al. [23]. The plastic deformation of asperities is large under high excitations amplitude, which results in reduction and possibly complete loss of clamp force over time. There exists an endurance limit for the self-loosening curve, much like the one for steel and other metals, and larger preload results in a larger endurance limit, but a large preload can also result in an increased risk of fatigue failure [23], especially in combination with high excitation amplitudes.

Fretting wear is a type of wear damage typically induced by a short amplitude sliding motion between two loaded surfaces [3] and can reduce the fatigue life substantially and is therefore one of the main reasons for failure. This microscopic dynamic process involves deformed structures, cracks and oxidation processes which include oxide debris between contact surfaces, heat generation and possible cold welding of contact surfaces. Fretting can be understood as three interrelated problems: (1) Fretting wear, (2) fretting corrosion and (3) fretting fatigue, where fretting wear is typically a wear damage due to a fretting problem, and important for prediction of fretting fatigue.

In the study reported by Karamiş and Selçuk [29], it was concluded that improvement of the surface roughness is vital in the joint reliability. The friction at the contacting surfaces depends not only on the contact area, but also strongly on the surface quality and level of asperities. The real contact area can be described as the sum of the contact areas for each asperity, which in total is much smaller than the apparent or nominal contact area. As a result of relative movements between the contacting surfaces due to typically axial or transverse excitation, the asperities will be plastically deformed, and the real contact area will increase. Such a deformation of asperities results therefore in loss of clamp load in the bolted connection, and possibly self-loosening of the bolts.

It has been found that a given set of system parameters that change in preload can result in changing from loosening to tightening of the nut [22,30]. Such behaviour involves nonlinear dynamic interaction of both friction and vibration, frequency, and amplitude of vibration, contact stiffness, pre-load, and mass of the clamped components. By taking all these factors into consideration, it could be possible to tune the input factors to exhibit loosening, tightening or no twist at all, of the nut.

Several studies have concluded that transverse loading of a bolted joint is the most severe when it comes to self-loosening [17,26]. Pai and Hess [31,32] were the first to propose and concluded that a complete slip is not an absolute condition for self-loosening of bolts, and that a localized slip can be sufficient. They concluded that there are four possible processes for loosening.

(1) localized slip both at threads and head at the same time,
(2) localized head slip in combination with complete thread slip,
(3) localized thread slips in combination with complete head slip, and
(4) complete slip both on thread and head at the same time.

The complete slip can be a result of an accumulation of localized slips in the form of elastic deformation over time. The analysis also indicated that the loosening processes are mainly independent of the frequency but depend highly on the applied load amplitude.

Already in 1973, Junker [17] had showed the relationships between the relative displacement between clamped parts, the transverse load, and the axial preload for a bolt, using a new kind of machine (at that time), now called the “Junker machine” [26].

For a bolt or nut to rotate, it must overcome the resisting friction torques at the bolt head and threads. Zadoks and Yu [1] showed that for transverse excitation, the relative motion between bolt head and flange is the most important when it comes to self-loosening because of a larger friction radius than on the threads. Housari and Nassar [26] studied the effect of thread and bearing friction coefficients on the transverse vibration-induced loosening of threaded fasteners, and both the torque and the thread friction force magnitude are affected. They showed the relationship between thread friction coefficient and loosening rate, for various bearing friction coefficients, and the relationship between bearing friction coefficient and loosening rate, for various thread friction coefficients.

Loss of preload in flanged bolted connections is a major concern in most industries, as in the relatively new industry of Offshore Wind, and several factors are involved in the process. Braithwaite et al. [33] performed a sensitivity analysis of friction and creep deformations on preload loss in bolted connections for Offshore Wind turbine systems. They concluded that the higher the friction coefficient between the contacting surfaces are the lower the preload into the bolt is. In addition, preload relaxation can come from

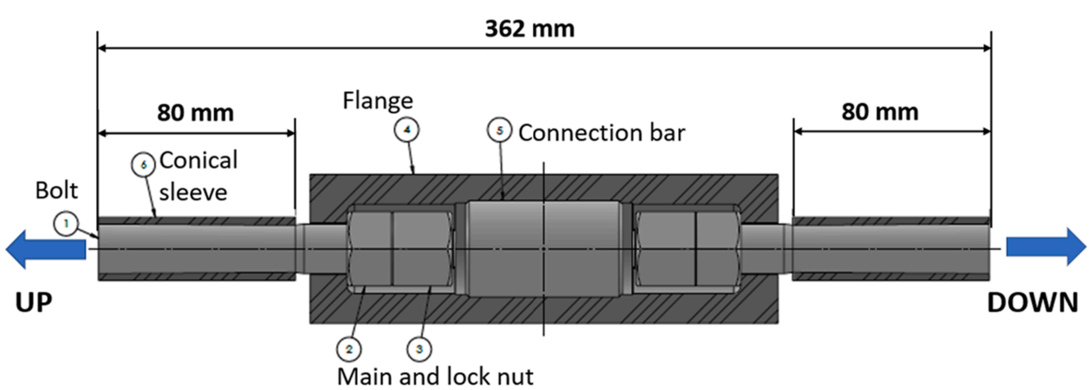

Fig. 3. Test setup for the anti-loosening M20 bolt system.

material creep, but normally not at low temperatures if the pretension stress stays below 75% of the material yield limit.

It can be concluded, based on the literature study, that many investigations have been performed on self-loosening of bolts, over many years. This is because self-loosening of bolts represents a major problem in many industries and is heavily related to high costs and safety issues. Transverse loads in oscillation represent the type of external exposure that generate most self-loosening issues in preloaded bolt system, where elongation beyond the elastic limit of the threads and microslip on the thread and bearing surfaces are the two basic loosening mechanisms.

3. Experimental methods and materials

3.1. Experimental setup for test 1

Fig. 3 shows the test setup of the anti-loosening bolt system with M20 bolts, where the bolt system is installed vertically in the fatigue machine. The bolt heads are removed for this test to ease the clamping connection with the fatigue machine, and the test setup allows two different anti-loosening bolt systems to be tested simultaneously, with axial loading and oscillation. Three bolt system setups were considered:

1) Bolt system A: lubricated, exposed to 498 $\times 10^3$ cycles
2) Bolt system B: unlubricated, exposed to 498 $\times 10^3$ cycles
3) Bolt system C: lubricated, exposed to 668 $\times 10^3$ cycles

Each bolt system, A, B and C, was loaded with 100 kN axial tension, and oscillating (amplitude) load of $\pm$15 kN, which gives a utilization level of approximately 60% to material yield, at 10 Hz frequency and without washers. The bolts are made of material quality 10.9, and the locking nut was torqued with 200 Nm against the main nut. The axial force from the test machine was intended to simulate the preload from torquing the main nut, or hydraulic tension of the bolt. All threads on bolts A and C were lubricated with Molykote P74. The threads in contact between bolt and nuts, and the nutś coned contact surface were inspected with Stereo (binocular) microscope (Olympus SZX16) and scanning electron microscope (SEM) (Zeiss Supra 35 VP) to discover any surface damages due to the imposed vibrations. In addition, the bolt thread pitches and nuts were measured with a coordinate measuring machine (CMM)(Zeiss Contura) both before and after loading. This was done to investigate any dimensional changes, and it was checked whether any of the nuts had turned loose, by comparing photos before and after loading. The fatigue test was conducted using MTS 809 Axial/Torsional Test System, Model 319.25.

3.2. Experimental setup for test 2

To conduct this experiment, the following test and measuring equipment were used whose specifications are also provided:

- Fatigue test machine: MTS 809 Axial / Torsional Test System, Model 319.25.
- Displacement measurement unit: LVDT – linear variable differential transformer.
- Clamp force loss measuring unit: Instrumentation / 2 strain gauges at each bolt

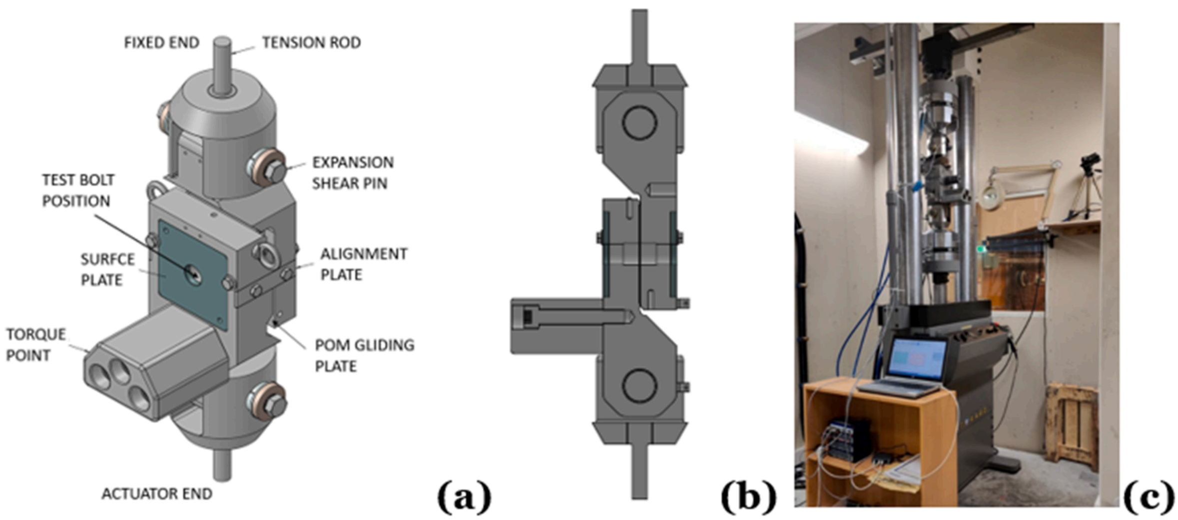

Fig. 4. (a) Test jig schematics, (b) cross-section view, and (c) jig installed in hydraulic test rig.

The test setup and test jig are shown in Fig. 4 and implemented in a 260 kN dynamic standard test rig (MTS Series 809), which achieves the load, amplitude and frequency needed for the applied Junker test. The load jig was designed for this scenario specifically with two main structure plates where one is moving by connection to the hydraulic test machine cylinder and the other is stationary. The jig is connected to the hydraulic test machine by a tension rod and pivot pin connection on the top and bottom. A Polyoxymethylene (POM) gliding plate was located between the two structure plates to reduce friction and provide a smooth and even movement of the jig. Replaceable surface steel plates were used under the bolt head and the nut to ensure the same surface conditions for each test. Alignment plates were used for the installation of bolts to centre the test samples. Molykote® G-Rapid Plus was used as thread and assembly paste to maintain a consistent friction coefficient between threads and surfaces for the duration of the test.

The calculations of the preloads, effective cross-section area and required torque for the tests of the two bolt dimensions are given by the German guidelines of the VDI 2230:2014 [34], which is the standard reference for calculating highly stressed bolted joints with one cylindrical bolt. The calculated pre-tension for a M30 and M42 bolts of grade 10.9 was 353 kN and 706 kN, respectively, which represent 70% of the material yield strength and achieved by using (a) a tensioning tool, and (b) a torque tool. The displacement amplitudes to be achieved for the M30 and M42 tests are $\pm$ 1.0 mm and 1.5 mm, respectively, and was defined by DNV (Germanischer Lloyd) based on earlier similar tests. The M30 bolt has 3.5 mm pitch on the main nut and 1.5 mm on the locking nut, and for the M42 bolt the pitch values are 4.5 mm and 2.0 mm, respectively. The design requirements for the bolt systems are shown in Table 1.

The actuator forces and displacements are logged by the MTS test machine control unit and computer, the data from the strain gauges and displacement transducers are logged by a HBM QuantumX data acquisition device via Catman AP. The test frequency of amplitude is 1 Hz with data logging of 100 Hz.

Tension and torque tools were used to achieve the preload specified to test both methods. Some test runs of the reference bolts were found necessary in the start to determine the correct amplitude to be used for the testing, where the reference HV bolts were calculated to loose more than 90% of their preload in the range of 200–400 cycles, and afterwards the test was repeated 3 times with the determined amplitude. The VIBRALOCK® assemblies were tested for verification of the system with a minimum of 3000 cycles for each test samples, with the same loads and dicplacement amplitudes as for the reference bolts. The referance bolts were both of type HV, produced according to NS-EN 14399-4 [35] of material quality 10.9. The following parameters were logged; amplitude, clamp load at start, clamp load at end, number of cycles, while the loss of preload over the duration of the test was calculated. All loads were given in kN and the loss of preload was in addition given in percentage values.

3.3. Finite element analysis

The finite element analysis was performed prior to experimental Test 2, and the ANSYS 19 R2 program was used with 3D elements. Both linear and non-linear models (with quadratic element order for the 3D elements and linear element order for axisymmetric model) were checked to verify the structural integrity of the bolt system. The friction coefficient between threads was set to 0.12, while the friction between the two nuts was set to 0.35. The conical contact between the nuts is 30° to the bolt axis and includes a fabrication tolerance of 0.5°.

Two different analyses were done, whose finite element model is shown in Fig. 5(a). The first analysis was conducted using linear material model, with non-linear geometry (large deformation theory) and contacts. First stage of this analysis was done by pre-tensioning the bolt shank up to 2 180 kN, and the second stage was done by tightening the locking nut up to pre-tension of 930 kN. The second type analysis was the same as the previous analysis, but with non-linear material model. The analysis was conducted with the following conditions regarding geometry and material, boundary, contact and mesh.

Boundary conditions: Identical boundary conditions were used for both analyses. The locking nut was torqued until required pre-tension was achieved. As illustrated in Fig. 5, the washers were set as fixed outside the jig bore. The bolt head and the nut end centers were locked against rotation and displacement in the YZ plane, which were placed perpendicular to the bolt axis.

Geometry and material: The analysis was performed on a bolt system with coarse threads M72 $\times$ 6 and fine threads M48 $\times$ 3 (Fig. 6(a)). The clamp length between the washers is 157 mm. For the linear elastic model, Youngś modulus of 210 GPa, and Poissońs constant of 0.3 were used, while bilinear isotropic hardening was assumed for the non-linear material model.

Contact conditions: Normal Lagrange formulation was employed on all frictional surfaces for the linear analysis, as it allows no contact surface penetration. In addition to previous friction coefficients, the coefficients between washers and nuts/bolt are set to 0.2.

Table 1
Design requirements for the anti-loosening bolt systems.

General requirements for the bolt system		NO-EN 14399-4:2015 & DASt-Richtlinie 021
Threads on bolt	Tolerance	6 g
	Standards	ISO 261 & ISO 965-2
Threads on main and lock nut	Tolerance	HDG – 6AZ*
	Standard	ISO 261 & ISO 965-5, ISO 261 & ISO 965-1
Mechanical properties for bolt and nuts	Property class	10.9
	Standard	ISO 898-1/-2
General tolerances on bolt and nuts	Product grade	C / B
	Standard	EN ISO 4759-1
Finish - coating	Standard	EN ISO 10684 & DSV-GAV guideline for the manufacturing of hot-dip galvanized screws.

* Internal threads of the nuts are first coated (HDG) and then the threads are cut.

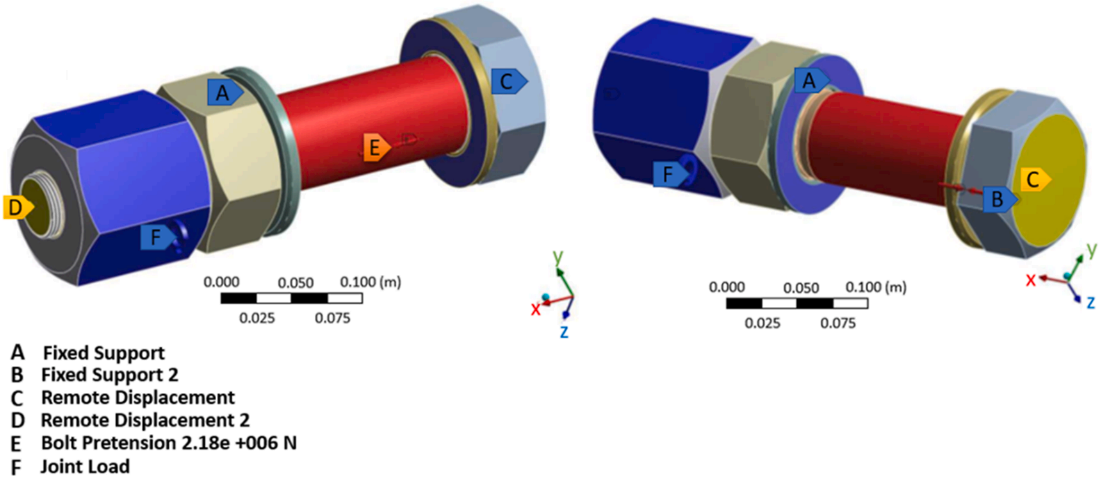

Fig. 5. Boundary conditions.

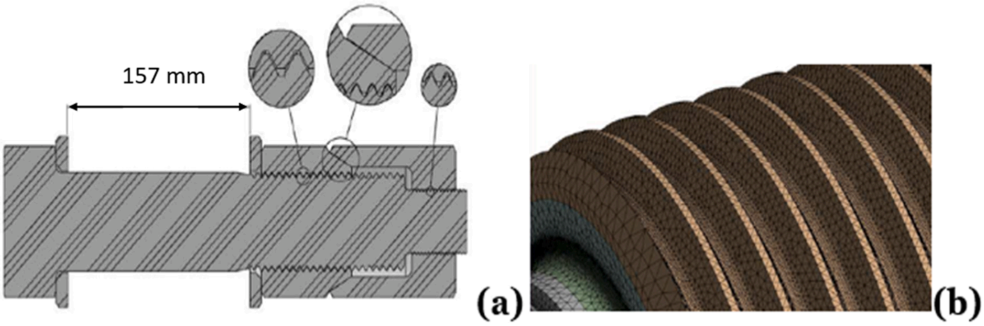

Fig. 6. (a) Bolt assembly, and (b) Meshed model of bolt.

For the non-linear analysis, the conditions were the same, but with the exception that the Augmented Lagrange formulation was used instead of the normal. This is acceptable if the penetration is insignificant.

Mesh: Linear and full integration elements were used for the linear analysis (Fig. 6 (b)), and the meshed model has a total number of 1 157 297 nodes and 5 792 650 elements. For the non-linear analysis the number of nodes and elements reduced to 532 513 and 2 460 415, respectively. The models were meshed using the default element sizes of 0.0022 mm and 0.001 mm for the linear and quadratic element models respectively.

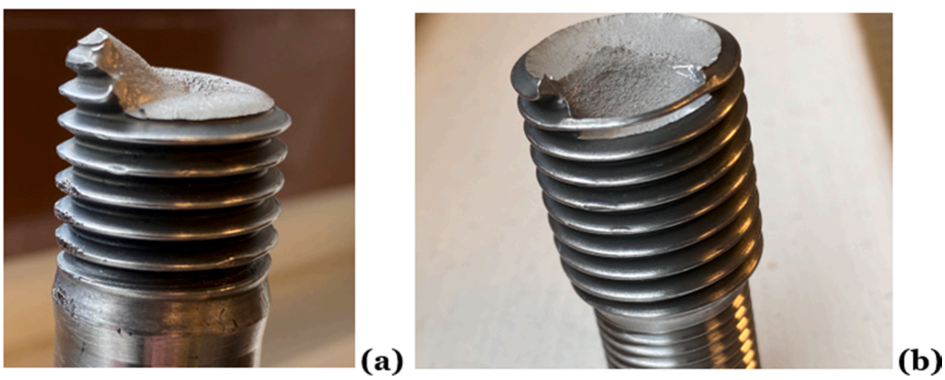

Fig. 7. Breakage of bolt B, (a) threads outside the main nut, and (b) threads within the main nut.

4. Discussion of results

4.1. Results of test 1

The study results reported in this paper indicate that the new bolt locking concept shows a superior locking, or anti-loosening capacity compared to the standard HV bolt-nut system. The HV bolt system had a preload loss of 86–92% within 200–400 transversal load cycles, while the novel locking bolt system had a loss of only 3–9% within 3000 cycles. Both systems were preloaded to 70% of yield with the same forced displacement amplitude in transverse direction. The preload loss in the tested anti-loosening bolt system did not occur due to turning of the nuts, which indicates that the registered preload drop is mainly due to an immediate reduction of asperities, and probably not due to any material creep effects since the pretension stress in the bolt is maximum 70% of material yield, and the temperature is low, 20–25 °C.

Test 1 was conducted on M20 bolt size with minor shank size M16 subjected to an axial oscillational excitation and the test contained 3 bolt systems, designated as bolt A, bolt B and bolt C. Bolt B was not lubricated nor the most loaded, and the break initiated at the first engaged thread into the main nut towards the flange. Fig. 7 show the broken bolt images and Figs. 8 and 9 show the SEM images of this bolt. The images indicate that first threads are taking the highest loads. Although it is not possible to conclude with certainty, this may be due to lack of thread lubrication, which damaged the surface and initiated the breakage. The locking nut and the main nut did not have relative rotation during the testing, for neither of the test bolts.

4.2. Results of test 2

This test was conducted on M30 bolt size with minor shank size M16, and M42 with minor M24, both with transverse oscillation excitation. The test procedure was developed by DNV GL Oil&Gas in accordance with Junker vibration test method. The bolts were tested at alternating transverse loading to determine their resistance against loss of preload and consequently loosening.

All the HV bolts, tensioned and torqued, showed an immediate and almost linear loss of preload from the start of each test, until the nut came completely loose. The Vibralock® bolts showed generally an immediate minor loss of preload, and thereafter practically no further loss until the end of the test.

The tests of the M30 and M42 bolt locking systems showed a superior resistance to self-loosening compared to the HV reference bolts of same sizes, see Figs. 10 and 11. The HV reference bolts lost 86–92% of their preload before 400 cycles, while the bolt locking systems had a loss of 3–9% before 3000 cycles, for the same tests. The test requirements were specified by DNV, for a possible approval of concept at a later stage, and therefore assumed to correctly represent the comparison technique between the two concepts, the locking bolt and standard HV bolt concepts.

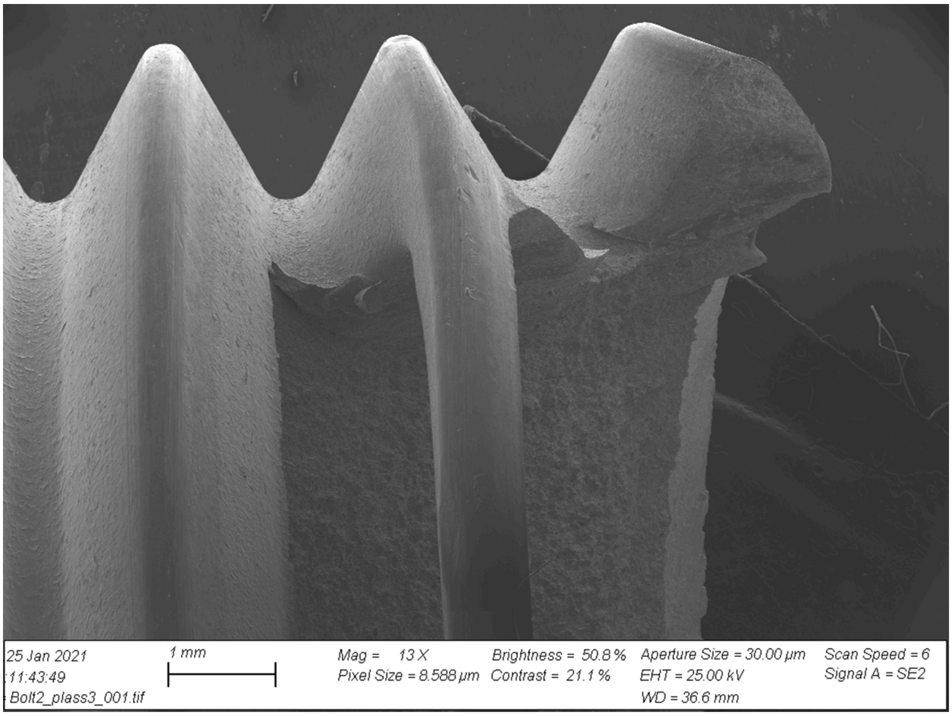

Fig. 8. SEM image of broken bolt B at first engaged threads, with Mag = 13X.

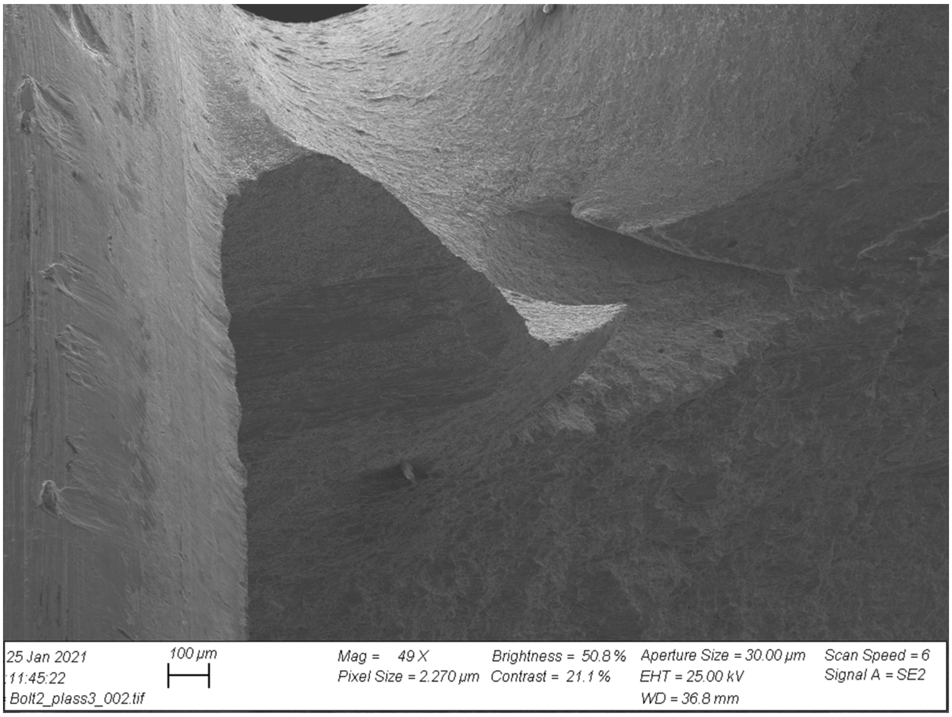

Fig. 9. SEM image of broken bolt B at first engaged threads, with Mag = 49X.

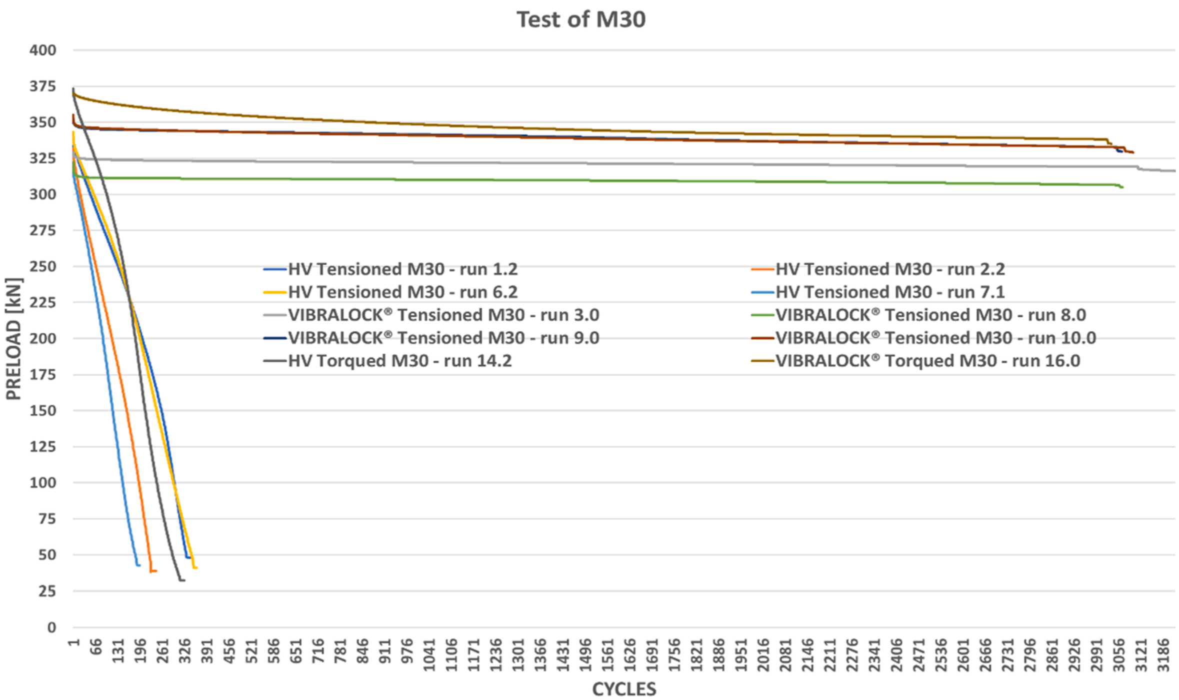

Fig. 10. Loss of preload Vibralock® versus HV bolts for bolt size M30.

4.3. FE results

The FE analysis was performed by an external company on a M72 bolt, which is bigger than the bolts tested physically, i.e., M20, M30 and M42. The linear FE analysis was performed mainly to investigate the structural integrity of the threads in the bolt and nut

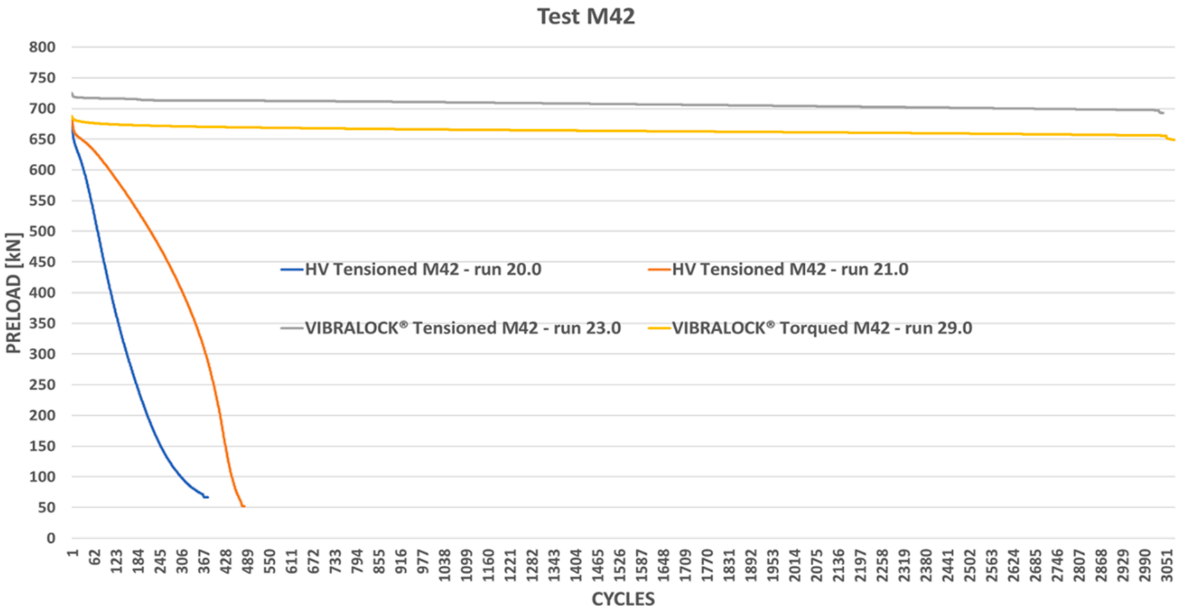

Fig. 11. Loss of preload Vibralock® versus HV bolts for bolt size M42.

system. The analysis was done based on the predefined clamp loads in both bolt diameters, whose values are 2 180 kN in the main shank and 930 kN in the minor shank. The torque value result from ANSYS to reach the locking nut preload of 930 kN is 29 967 Nm, which is almost identical to the value obtained by manual calculation using the formula in Eq. (1) [34], which gave calculated preload value of 29 440 Nm. In the non-linear analysis, the corresponding preloads required to reach the locking nut preload of 930 kN were 32 105 kN from the FE analysis and 32 890 kN as calculated from Eq. (1).

$$T = \frac{F_p}{2}\left[\frac{(\mu_n \times d_n)}{\sin 30^\circ} + 1.155 \times \mu_t \times d_t + \frac{p}{\pi}\right] \tag{1}$$

Where the symbol definitions and values are as given in Table 2 and the Sin 30° value is due to the conical shape of the nut contact surfaces.

Figs. 12 and 13 show the von Mises distribution of the FEA results from the linear analysis and non-linear analysis, respectively. In the linear analysis, a fabrication tolerance of 0.5° was introduced between the nut angles where the contact point between the nuts where at the lower conical area with an average contact diameter of 80 mm. In the non-linear analysis the conical fabrication tolerance was set to 0°, which resulted in a wider contact area between the nuts and increased contact diameter, up to 90.6 mm.

When comparing the two models, it can be seen that the stress pattern is almost identical, as illustrated in Fig. 11(a) and 12(a). The linear analysis resulted in the maximum stress level of 820 MPa at the main nut contact area against the locking nut, and at the first engaged coarse bolt threads, in addition to the first engaged fine bolt threads. In the non-linear analysis, the over-all stress level is approximately the same as for the linear analysis, also at the most stressed areas, where the stress level is in the range of 788 – 900 MPa. A very small portion of the main nut to locking nut contact area shows an increased stress level of 900–1350 MPa.

Though the simulation was done on a bigger bolt size, the plots in Fig. 12 (c) and 13 (c) coincide well with the fracture condition of bolt B in Test 1, which occurred at or close to the first engaged thread against the main nut (Figs. 7 - 9) and reflects the description from literature [36]. In general, the combination of physical tests and FEA, and the combination of 4 different bolt dimensions is meant to give a wider *proof of concept* of the bolt locking system.

Bolted connections are widely used in many different applications and machines, and it is commonly known that the initial bolt preload will be reduced over time, due to vibrations. Xu et al. [37] performed dynamic analysis and loosening evaluation of bolted connections in machine tools, exposed to vibrations. They introduced a new experimental design method called *The quadratic rotary*

Table 2
Symbol definitions and values used in Eq. (1).

Parameter and symbol	Linear analysis	Non-linear analysis
Required bolt pretension, F_p [kN]	930	930
Friction coefficient at nut bearing surface, μ_n	0,35	0,35
Effective contact diameter of nut face, d_n [mm]	80	90,6
Friction coefficient at threads, μ_t	0,12	0,12
Effective mean contact diameter of threads, d_t [mm]	45,925	45,925
Pitch, p [mm]	3	3

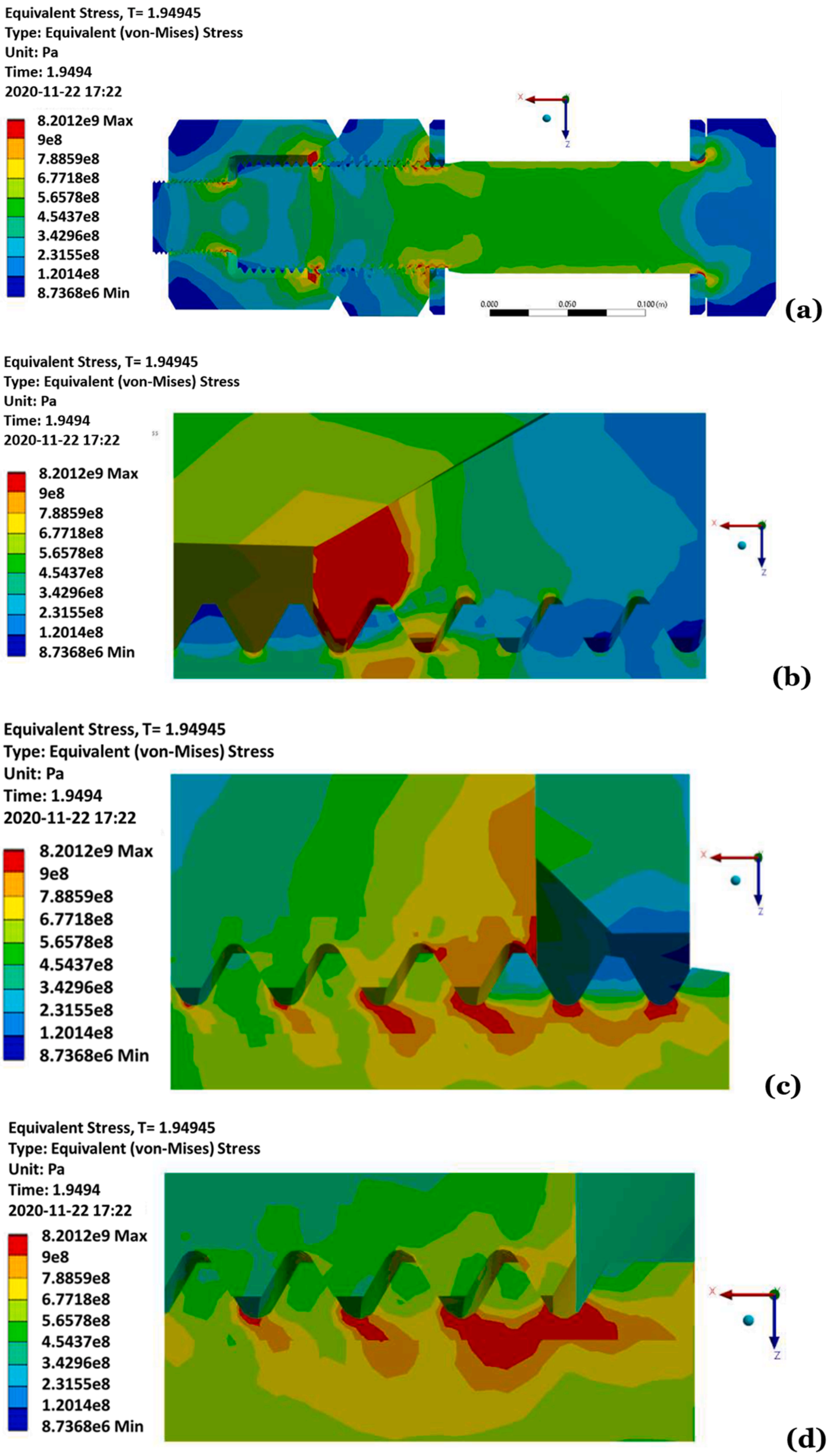

Fig. 12. von Mises stresses from the linear FE analysis, (a) overall view, (b) conical contact area, (c) coarse threads, and (d) fine threads.

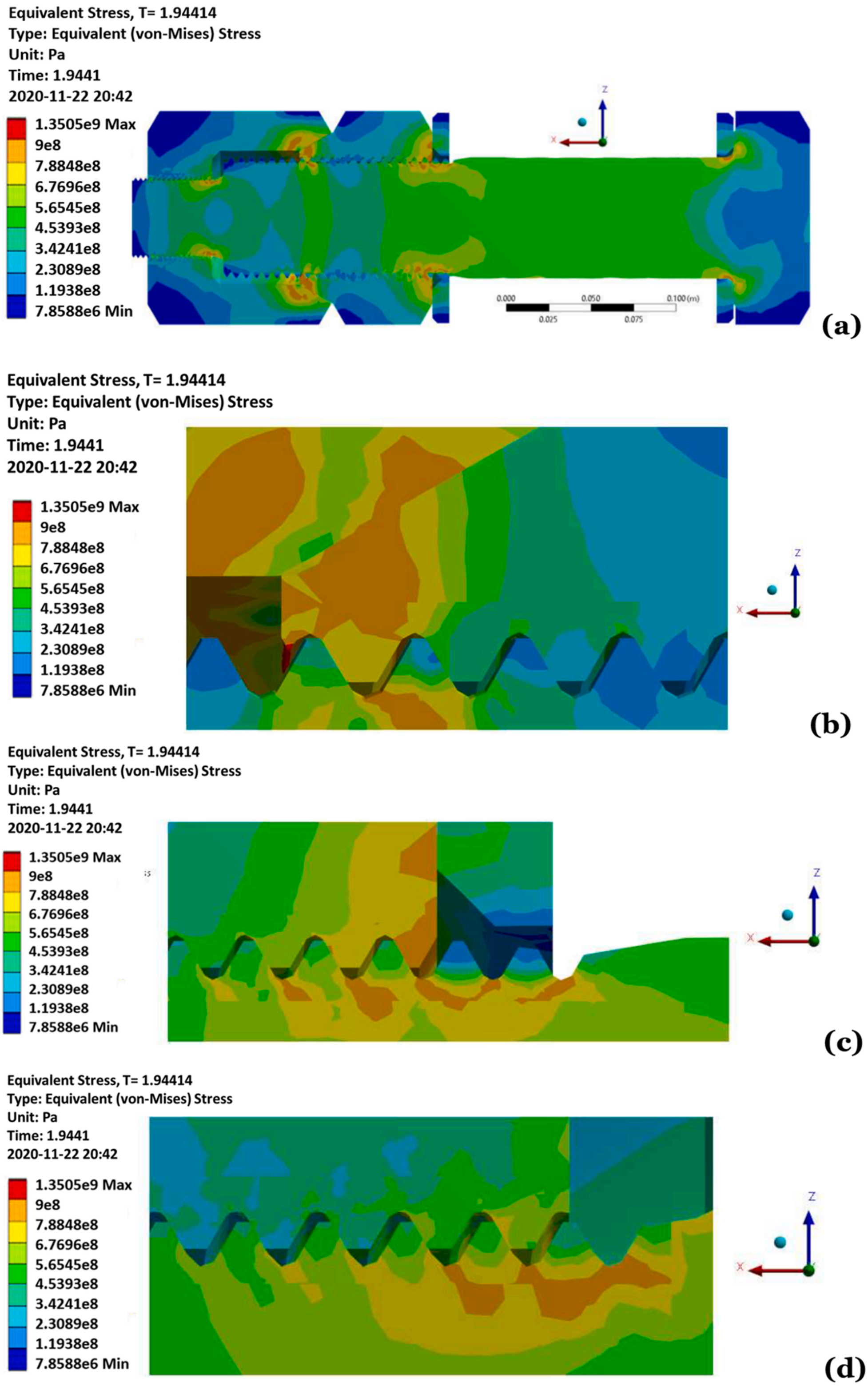

Fig. 13. von Mises stresses from the non-linear FE analysis, (a) overall view, (b) conical contact area, (c) coarse threads, and (d) fine threads.

unitized design, from where the principal factors affecting the preload loss can be obtained, such as preload, working load (cyclic), frequency, temperature, bolt form and diameter, and surface state, such as coefficient of friction. Based on the obtained results, the authors concluded that the preload attenuation rate decreases with increased preload but increases with increasing cyclic load and increased frequency. Vilela et al. [38] presented a numerical simulation by finite element method (of ANSYS software) of a model with one bolt only connecting two and three plates. Bolts subjected to tension, shear, and a combination of the two were modeled with two, three and two plated, respectively. The unitary model includes all necessary and required considerations such as contact between all friction surfaces involved, bolt preload, parallel and perpendicular forces compared to plate contact surfaces, including slip and prying effects. The authors concluded, based on all the analyses developed, that the methodology used in the unitary model is capable of simulate with great accuracy the behavior of a single bolt subjected to tension, shear, and a combination of the two.

5. Conclusions

The new bolt locking system can be used in any joint, location or mechanical system where self-loosening is real or potential problem, or where the consequences of loose bolts are high. This could be in wind power systems (offshore and onshore), huge valve connections, tube connections exposed to high pressure and/or chemical fluids, bridges, or other constructions, and many more.

Based on the test and simulation results, the following main conclusions can be made regarding the new bolt locking system:

- The bolt system is superior to standard bolt-nut systems when it comes to anti-loosening resistance
- It experiences a minimum of preload loss over time.
- No turning of nut relative to bolt takes place.
- It is easy to preload and disarm, if required.

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.

Acknowledgements

The authors are grateful for all help and service given by Mr. Andreas Sigvardsson at Stavanger Engineering AS for the support provided in running the Finite Element Analysis for this project, Teo Teknikk AS and Torq/Lite Europe AS for providing their services within strain gauges and torque & tensioning, and DNV for designing the test procedures.

Funding

The research is conducted as part of Industrial PhD study funded by the Norwegian Research Council and Bondura Technology AS, Grant nr. 283821. This financial support is highly acknowledged.

References

[1] R.I. Zadoks, X. Yu, An investigation of the self-loosening behaviour of bolts under transverse vibration, J. Sound Vib. 208 (2) (1997) 189–209, https://doi.org/10.1006/jsvi.1997.1173.

[2] I. Berkani, Ø. Karlsen, H.G. Lemu, Experimental and numerical study of Bondura® 6.6 PIN joints, IOP Conf. Ser.: Mater. Sci. Eng. 276 (2017) 012028, https://doi.org/10.1088/1757-899X/276/1/012028.

[3] Ø. Karlsen, H.G. Lemu, Fretting fatigue and wear of mechanical joints: Literature study, IOP Conf. Ser.: Mater. Sci. Eng. 700 (1) (2019) 012015, https://doi.org/10.1088/1757-899X/700/1/012015.

[4] Ø. Karlsen, H.G. Lemu, On modelling techniques for mechanical joints: Literature study. Int. Workshop of Advanced Manufacturing and Automation (IWAMA 2019). Springer, Singapore, 634 (2020) 116-125. https://doi.org/10.1007/978-981-15-2341-0_15.

[5] I.R. McColl, J. Ding, S.B. Leen, Finite element simulation and experimental validation of fretting wear, Wear. 256 (11–12) (2004) 1114–1127, https://doi.org/10.1016/j.wear.2003.07.001.

[6] V. Sabelkin, S. Mall, Investigation into relative slip during fretting fatigue under partial slip contact condition, Fatigue Fract. Eng. Mater. Struct. 28 (9) (2005) 809–824, https://doi.org/10.1111/j.1460-2695.2005.00918.x.

[7] J.J. Madge, S.B. Leen, P.H. Shipway, The critical role of fretting wear in the analysis of fretting fatigue, Wear 263 (1–6) (2007) 542–551, https://doi.org/10.1016/j.wear.2006.11.021.

[8] A.T. Kasarekar, F. Sadeghi, S. Tseregounis, Fretting fatigue of rough surfaces, Wear 264 (7–8) (2008) 719–730, https://doi.org/10.1016/j.wear.2007.06.016.

[9] C. Song, M.X. Shen, X.F. Lin, D.W. Liu, M.H. Zhu, An investigation on rotatory bending fretting fatigue damage of railway axles, Fatigue Fract. Eng. Mater. Struct. 37 (1) (2014) 72–84, https://doi.org/10.1111/ffe.12085.

[10] M. Luke, M. Burdack, S. Moroz, I. Varfolomeev, Experimental and numerical study on crack initiation under fretting fatigue loading, Int. J. Fatigue 86 (2016) 24–33, https://doi.org/10.1016/j.ijfatigue.2015.09.022.

[11] X. Mi, Z.B. Cai, X.M. Xiong, H. Qian, L.C. Tang, Y.C. Xie, J.F. Peng, M.-H. Zhu, M-h. Zhu, Investigation on fretting wear behavior of 690 alloy in water under various temperatures, Tribol. Int. 100 (2016) 400–409, https://doi.org/10.1016/j.triboint.2016.05.012.

[12] W.A. Grabon, M. Osetek, T.G. Mathia, Friction of threaded fasteners, Tribol. Int. 118 (2018) 408–420, https://doi.org/10.1016/j.triboint.2017.10.014.

[13] Enerpac, https://www.enerpac.com, [Online; accessed 15-March-2022].

[14] Torq/Lite Europe, https://www.torqlite.eu, [Online; accessed 15-March-2022].

[15] T. Gintautas, J.D. Sørensen, Improved methodology of weather window prediction for offshore operations based on probabilities of operation failure, J. Mar. Sci. Eng. 5 (2) (2017) 20, https://doi.org/10.3390/jmse5020020.

[16] Bondura Technology AS, https://www.bondura.no/, [Online; accessed 15-March-2022].

[17] G.H. Junker, Criteria for self-loosening of fasteners under vibration, Aircr. Eng. Aerosp. Technol. 45 (1) (1973) 12–17, https://doi.org/10.1108/eb03498.
[18] L. Zhu, J. Hong, X. Jiang, On controlling preload and estimating anti-loosening performance in threaded fasteners based on accurate contact modelling, Tribol. Int. 95 (2016) 181–191, https://doi.org/10.1016/j.triboint.2015.11.006.
[19] M. Zhang, L. Lu, W. Wang, D. Zeng, The roles of thread wear on self-loosening behavior of bolted joints under transverse cyclic loading, Wear 394 (2018) 30–39, https://doi.org/10.1016/j.wear.2017.10.006.
[20] S. Basava, D.P. Hess, Bolted joint clamping force variation due to axial vibration, J. Sound Vib. 210 (2) (1998) 255–265, https://doi.org/10.1006/jsvi.1997.1330.
[21] J. Liu, H. Ouyang, J. Peng, C. Zhang, P. Zhou, L. Ma, M. Zhu, Experimental and numerical studies of bolted joints subjected to axial excitation, Wear 346 (2016) 66–77, https://doi.org/10.1016/j.wear.2015.10.012.
[22] J. Liu, H. Ouyang, Z. Feng, Z. Cai, X. Liu, M. Zhu, Study on self-loosening of bolted joints excited by dynamic axial load, Tribol. Int. 115 (2017) 432–451, https://doi.org/10.1016/j.triboint.2017.05.037.
[23] Y. Jiang, M. Zhang, T.-W. Park, C.-H. Lee, An experimental study of self-loosening of bolted joints, J. Mech. Des. 126 (5) (2004) 925–931, https://doi.org/10.1115/1.1767814.
[24] D. Croccolo, M. de Agostinis, N. Vincenzi, Failure analysis of bolted joints: Effect of friction coefficients in torque–preloading relationship, Eng. Fail. Anal. 18 (1) (2011) 364–373, https://doi.org/10.1016/j.engfailanal.2010.09.015.
[25] S.A. Nassar, P.H. Matin, Clamp load loss due to fastener elongation beyond its elastic limit, J. Press. Vessel Technol. 128 (3) (2006) 379–387, https://doi.org/10.1115/1.2217971.
[26] B.A. Housari, S.A. Nassar, Effect of thread and bearing friction coefficients on the vibration-induced loosening of threaded fasteners, J. Vib. Acoust. 129 (4) (2007) 484–494, https://doi.org/10.1115/1.2748473.
[27] A.M. Zaki, S.A. Nassar X, Yang, Effect of thread and bearing friction coefficients on the self-loosening of preloaded countersunk-head bolts under periodic transverse excitation. J. Tribol. 132(3) (2010) 031601. https://doi.org/10.1115/1.4001621.
[28] Z. Guan, J. Mu, F. Su, T. Bian, Y. Huang, Z. Li, Pull-through mechanical behavior of composite fastener threads, Appl. Compos. Mater. 22 (3) (2015) 251–267, https://doi.org/10.1007/s10443-014-9404-5.
[29] M.B. Karamiş, B. Selçuk, Analysis of the friction behaviour of bolted joints, Wear 166 (1) (1993) 73–83, https://doi.org/10.1016/0043-1648(93)90281-P.
[30] D.P. Hess, S.V. Sudhirkashyap, Dynamic loosening and tightening of a single-bolt assembly, J. Vib. Acoust. 119 (3) (1997) 311–316, https://doi.org/10.1115/1.2889725.
[31] N.G. Pai, D.P. Hess, Experimental study of loosening of threaded fasteners due to dynamic shear loads, J. Sound. Vib. 253 (3) (2002) 585–602, https://doi.org/10.1006/jsvi.2001.4006.
[32] N.G. Pai, D.P. Hess, Three-dimensional finite element analysis of threaded fastener loosening due to dynamic shear load, Eng. Fail. Anal. 9 (4) (2002) 383–402, https://doi.org/10.1016/S1350-6307(01)00024-3.
[33] J. Braithwaite, I.G. Goenaga, B. Tafazzolimoghaddam, A. Mehmanparast, Sensitivity analysis of friction and creep deformation effects on preload relaxation in offshore wind turbine bolted connections, Appl. Ocean Res. 101 (2020) 102225, https://doi.org/10.1016/j.apor.2020.102225.
[34] VDI 2230:2014, Systematic calculation of highly stressed bolted joints – Joints with one cylindrical bolt, Available at: https://www.vdi.de/en/home/vdi-standards.
[35] NS-EN 14399-4:2015, High-strength structural bolting assemblies for preloading-Part 4: System HV - Hexagon bolt and nut assemblies, Available at:https://www.standard.no/en/webshop/productcatalog/productpresentation/?ProductID=747126, [Online; accessed 15-March-2022].
[36] W. Wang, K.M. Marshek, Determination of the load distribution in a threaded connector having dissimilar materials and varying thread stiffness, J. Eng. Ind. 117 (1) (1995) 1–8, https://doi.org/10.1115/1.2803273.
[37] W. Xu, Q. Cheng, C. Yang, Y. Li, Dynamic analysis and looseness evaluation of bolted connection under vibration of machine tools, Int. J. Adv. Manuf. Technol. (2021) 1–10, https://doi.org/10.1007/s00170-021-07615-0.
[38] P.M.L. Vilela, H. Carvalho, L.F. Grilo, P.A. Montenegro, R.B. Calçada, Unitary model for the analysis of bolted connections using the finite element method, Eng. Fail. Anal. 104 (2019) 308–320, https://doi.org/10.1016/j.engfailanal.2019.06.001.