OYSTER 800 PROJECT

THREADED FASTENER DESIGN GUIDELINES

OY800-TN-0080

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GENERAL BACKGROUND

Aquamarine Power Ltd (APL) was a wave energy company responsible for developing a wave energy converter called Oyster. The Oyster system consisted of a Wave Energy Converter (WEC) located in shallow water close to the shore, with a bottom-hinged flap which oscillated due to wave action. Double acting pistons on each side of the WEC pumped water through a high-pressure pipeline back to shore, where high pressure water drove a Pelton wheel turbine connected to an electrical generator. The flow from the Pelton wheel discharged to a header tank and returned to the WEC via a low-pressure return pipeline.

APL deployed a full-scale 315 kW Oyster 1 system at the European Marine Energy Centre (EMEC) in August 2009, followed by a second-generation machine rated at 800kW, Oyster 800, in August 2011. APL ceased trading in November 2015, and the intellectual property was acquired by Wave Energy Scotland (WES), who propose to share relevant documents and information acquired with developers in the WES programme and the wider sector.

This document outlined internal guidelines adopted for the design and maintenance of bolted connections and fasteners on Oyster 800 and future WEC designs. These were informed by standards such as VDI 2230, learning from Oyster 1 and threaded connection failures experienced during initial phases of operation on Oyster 800 in 2012 and 2013.

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1. INTRODUCTION

Oyster 800 was installed at EMEC during 2011 and 2012, and subsequently underwent several years of commissioning and operating trials. One recurring area of concern was the long-term integrity of threaded fasteners on the machine. Operational experience showed that these connections are prone to loosening and corrosion. A presentation summarising some threaded connection failures is supplied separately [1].

During the design of Oyster 800, the criticality of securing fasteners was recognised in the light of experience and learning from Oyster 1. A guidance note was therefore prepared outlining recommendations maintained from the design of Oyster 1, and from new experiences gained directly from Oyster 800 or through other learning avenues. These include engagement with external specialists (notably Bolt Science and Technip Offshore Wind Ltd), specialist training (ASME PCC-1 course on bolted flange joint assembly), supplier engagement and literature review.

The only way to ensure that bolted connections will work effectively over the life of a project is through careful design engineering. Good working practices during component manufacture and joint assembly are also necessary, but if it is difficult (or impossible!) to implement the design intent correctly on site, the chances of joint failures increase significantly. It is therefore essential that the engineer understands how the joint will be made in practice and ensures the design is fit for this purpose.

The purpose of this document is to summarise Aquamarine Power's learning in the form of guidelines for good design practice in threaded fastener connections. For the purposes of this note, bolted connections have been generalised into a small number of discrete application categories. It is intended that common approaches may be adopted for design of bolted connections within each of these categories, which are described below.

1.1 Structural Connections

These are applications in which threaded fasteners are used to transfer loads and moments across a stationary physical interface. They are typically used where it is not feasible to make a welded connection, possibly due to reasons of material compatibility, or because there is a need for disassembly during the operational life of the equipment. The fasteners are tightened to create a compressive preload in the joint, which then carries operating loads through a combination of cyclic compressive stress and friction. The size of fasteners used for structural connections in Oyster 800 varied from small (e.g. M6) connections attaching instrument housings to very large (e.g. M64) fasteners holding major structural components such as the flap and bearing shafts together.

The design of conventional bolted connections in structures is covered by DNVGL-OS-C101 and Eurocode 3 (BS EN 1993-1-8:2005). These standards are considerably less conservative than the approach described in Section 2.2 of this guideline in the majority of cases. Their approach is only considered appropriate for joints in which the loads are predominantly non-reversing (e.g. gravity loaded structures). Very few such connections exist on offshore elements of an Oyster project, although there may be examples of this loading condition on onshore structures.

1.2 Hydraulic Flange Joint Assemblies

These are applications in which pressure retention is achieved through a joint which is comprised of two flanges, a gasket and a number of threaded fasteners. The purpose of the fasteners is to compress the gasket to provide a seal, and also to preload the joint to withstand the internal (hydrostatic) separating forces, as well as any external structural loads on the system.

Most commonly, these are specified as standard (e.g. ASME) connections, although a range of different standards were specified on Oyster 800, as well as at least one proprietary connection. The size of fasteners used on the hydraulic flanges ranged from M16 (on the installation ballast system) up to M39 (on the 14" HP lines), and also combined metric and imperial sizes of fastener.

There were also a number of hub-type hydraulic connections on Oyster (e.g. Grayloc and Techlok) but these are not considered explicitly in this revision of the guidelines.

1.3 Retaining Fasteners

In applications where relatively modest structural loads need to be transmitted, threaded fasteners are sometimes used without necessarily preloading the joint. The fasteners therefore transmit loads directly in shear and tension. This sort of connection is generally inadequate to resist significant reversing loads, but may be used in areas such as control and instrumentation, certain secondary steelwork and in connections where there are alternative load paths for the significant loads, such as cylinder pins. These bolts can be of any size.

1.4 Other Threaded Connections

Jacking screws are a special application in which a threaded fastener is used in compression to position or release one component with respect to another. They are usually loaded only temporarily, either as a one-off operation or intermittently, for example during maintenance. These can be almost any size. For example, the largest jacking screws used on Oyster 800 were M110 to retain the machine on the piles during grouting. Jacking screws are not explicitly considered in this guide, but will typically follow the same material guidelines as removable structural screws.

Rock bolts are a special case of structural connection, in which one half of the joint is made with the ground, typically either the seabed (offshore) or a concrete slab (onshore). These connections are not covered in detail in this document, but are the subject of a separate technical guidance note [2]. It should be noted that this sort of joint can suffer from extreme relaxation because the epoxy adhesive used to secure the fasteners into the rock or concrete may be susceptible to creep and local crushing of the seabed can result in large embedment losses. There may also be alignment issues if the hole drilled on site is not completely parallel to the structure to be attached. It can therefore be difficult to make a high quality structural connection with rock bolts without investing significant effort in both design and installation.

A number of hydraulically actuated valves were used on Oyster 800. A synthetic ester fluid was distributed at pressures of up to 210bar through a series of hoses and fittings of between ¼" and ½" in size. A variety of different connection types were used, many of which relied on threaded connections for their mechanical integrity. Some of these fittings were intended to be permanent connections and some needed to be periodically disconnected for maintenance.

There were also some small hydraulic fittings such as bungs and needle valves connected to the main Oyster 800 hydraulic system for commissioning and maintenance purposes, which proved particularly troublesome in service. Hydraulic fittings are also the subject of a separate guidance note [3].

2. DESIGN OF STRUCTURAL BOLTED CONNECTIONS

The first question to ask before designing a bolted connection is whether there is a better alternative available. If possible, the engineer should seek to eliminate the need for a joint at all. Bolted connections are not normally efficient methods of transmitting loads through a structure in terms of the amount of material used or the effort required for assembly. Careful detailing is required by the design engineer to ensure the connection can perform as intended in service and there always remains a risk that even the best designed bolted joint can be manufactured and/or assembled incorrectly, which can lead to failure in service.

If a connection is to be permanent then the first-choice assembly method should be to weld. Welds are typically much more efficient in their material usage than bolted joints and well-established quality controls exist to ensure the final result will meet the design engineer's specifications. Of course, there may be good reasons why this is not possible. For example, in fatigue-critical areas a bolted connection may be more efficient (particularly for secondary steelwork) and some materials (or material combinations) are very difficult to weld, such as iron castings. In some cases, it may actually be more difficult to assure the quality of a weld than a bolted connection, for example if the assembly is to be done subsea, or in a remote location without easy access to certified welders or NDT facilities. Design of welded connections is addressed in a separate guidance note [4].

The most common reason for using bolted connections rather than a welded construction is to allow for assembly and/or disassembly in service. In such cases, **a good alternative to consider is a pinned joint.** Carrying loads through a shear pin is often more efficient than using a bolted connection. Pins are also less susceptible to incorrect assembly. However, connections which are expected to see significant load reversals require pins with very small clearances to prevent wear and high local stress concentrations. It may be difficult to achieve adequate tolerances for a pinned connection and in such cases the difficulty of assembly will increase significantly. Design of pinned connections is addressed in a separate guidance note [5].

If a bolted joint is still considered to be the most appropriate connection method, then it is essential that the assembly is designed appropriately to avoid failure in service. The following sections outline the steps which should be taken as a minimum to try and achieve this.

2.1 Design Concept: A bolt is a spring

Best practice in bolted connection design ensures a rigid compression load path, with a relatively flexible tension load path through the bolts. This ensures that load variations are carried through the stiffer compression load path, preventing stress cycling in bolts which can lead to fatigue failures. One important step in creating a flexible joint is to use long grip lengths. A good rule of thumb is that the thickness of the elements to be clamped should be at least five times as long as the bolt diameter if possible. This is not essential and can sometimes be very difficult to achieve in practice, but any connections which have grip lengths less than three bolt diameters should be treated with additional caution.

2.2 Fastener Group Design Method

The process starts by defining the loads which must be carried through a connection, including an appropriate load factor (e.g. from DNVGL-OS-C101). It is important to consider a range of representative load cases and analyse all of those which may drive the design of the joint, including not only operational loads but also abnormal loading conditions such as those associated with fault scenarios, installation, transportation, etc.

Next, define an initial joint geometry and then subject this to appropriate engineering analyses to distribute loads around the joint to assess design loads on individual bolts. It is important to determine both axial and shear forces for every bolt in each load case. **To distribute moments is normal practice to assume an elastic distribution of load within the joint**. Under this assumption, the force in a bolt is proportional to its distance from the centre of rotation. The centre of rotation can usually be assumed to be the centroid of a bolt pattern, unless physical features provide an obvious alterative pivot. **Forces can usually be considered to be distributed evenly between all bolts in a pattern**.

Axial loads can be considered as being carried through the joint as deviations from the compressive preloads introduced at each bolt position during installation. Under fatigue and ultimate limit states, the appropriate failure criterion to consider is that there should be no separation of the joint – in other words, tensile axial forces should remain smaller than the minimum calculated bolt preload in all fasteners at all times. If joint separation is occurring regularly, cyclic stresses will be carried through the threads of the fasteners and fatigue failures can be expected. However, it is generally acceptable for separation to be seen in the accidental limit state, provided bolt loads remain below the ultimate capacity of the fasteners.

Shear loads should be designed to be resisted by friction across the interface between the two halves of a joint. The normal force which generates this friction is the bolt preload, minus any tensile load being carried by the joint. Compressive loads increase the friction capacity in shear. Under fatigue and ultimate limit states, the appropriate failure criterion to consider is normally that there should be no slip of the joint because this will lead to fastener loosening (see Section 4.2). However, it is generally acceptable for slip to be seen in the accidental limit state, provided shear loads in the fasteners remain below their yield capacity and bearing stresses in any of the joint plies are not exceeded.

When considering a joint in combined shear and bending, tensile components of bending resistance should not normally be subtracted from bolt preload when calculating the frictional shear capacity of joint. The reason for this is that any loss of compression on one side of joint is compensated by increase in compression on other side of joint and so there is no difference to the overall capacity.

2.3 Individual Fastener Design Method

Once the loads in a bolting pattern have been resolved and the most heavily loaded fasteners identified, a detailed check on capacity should be undertaken. The preferred method for this check has been implemented in Aquamarine Power’s internal calculation template OY800-CAL-0099 [6]. It consists of the following steps:

- Define fastener size
- Define fastener material and lubrication
- Define installation method (this sets the tightening factor – see Section 3)
- Set a target percentage yield for the fastener – this sets the target bolt tension
- Calculate the torque most likely to achieve the required bolt tension
- Calculate the maximum and minimum stresses in expected in the bolt for the given torque value and tightening method
- Subtract embedment losses from the minimum bolt force (see Section 4.1)
- Check the resulting minimum preload gives an acceptable margin against separation and slipping (see Section 2.2)
- If the joint fails (either due to over stress in the bolt, or failure of the bolt to carry the required loads) change the assumptions made and iterate until the joint works.
- For non-standard bolts and nuts, thread stripping should also be checked as a failure mode.

Note that especially critical connections should be checked against the full method in VDI 2230 [7].

The material factor for fasteners designed using this method should attract a material factor of unity (γ_{Ms} = 1.0) because uncertainty in bolt preload is explicitly addressed. Similarly, the hole clearance factor should also be taken as unity (k_s = 1.0) when calculating the design slip resistance of a friction grip connection based on this bolt preload.

3. FASTENER INSTALLATION

3.1 Small Fasteners

Small fasteners can be fully torque tightened by hand using a suitable tripping torque wrench. This is the method which the design engineer should normally assume will be used for tightening bolted connections. In VDI 2230, when combined with a theoretical friction coefficient this method of assembly attracts a “tightening factor” of 1.8. This means that the actual bolt tension achieved for a given torque could vary by as much as ±40% from a nominal value. This has implications for the efficiency of the joint.

Hand tools can realistically be used to apply torques of up to about 700Nm. However, achieving torques this large requires a long wrench, a strong fitter and a firm foundation to react the applied load. It can be very difficult to generate this much force in the subsea environment, as divers have relatively little effective weight subsea and they are working in congested areas where large torque wrenches can be difficult, or indeed impossible, to accommodate. These factors can combine to severely limit the torque which can be generated. Unless it can be demonstrated that there is space to use a large torque wrench, an upper limit between 300-400Nm should be assumed for the subsea environment before hydraulic tooling is required.

3.2 Larger Fasteners

For larger fasteners, torque multipliers may be used. Torque multipliers are effectively gearboxes which allow larger torques to be created with hand tools by providing additional mechanical advantage. Torques in excess of 1000Nm can be generated, although the tools are quite bulky and not suitable for repeated subsea use (they require significant maintenance after use to avoid corrosion of the internal gears). They also attract the same tightening factor as the torque wrench of 1.8.

A better solution for large diameter bolts is to use hydraulic tools. These are generally either tensioning tools (specialist, hollow hydraulic jacks which directly stretch a bolt or stud) or torque tools (which simply apply more torsion to the nut than hand tools are capable of). There are pros and cons of both systems, but **the system which should be selected under normal circumstances is the hydraulic torque wrench**.

3.2.1 Hydraulic Torque Wrench

The main reason for the hydraulic torque wrench being the preferred system to select is for safety. A large amount of energy is stored in hydraulic tensioning tools whilst in use. A mechanical failure within either the tool or the fastener can result in this energy being transferred (without warning) into either a nut or the head of the bolt, which will then fly off at high speed along the axis of the bolt. The risk of injury in the event of this failure can be theoretically mitigated by always approaching a hydraulic tensioning tool from the side, but this can be very difficult to achieve in practice where space restrictions exist and many tools are in use at the same time. This is especially true for divers, who also have umbilical management issues to consider.

A secondary reason for not selecting bolt tensioning tools is that they require excess bolt lengths to be provided for the tool to work with, which is not helpful, especially for subsea applications (see Section 5.1.1).

A hydraulic torque wrench is a compact tool which can deliver very large torques with a good level of precision. This allows the tightening factor to be reduced to 1.6, allowing more efficient use of the fasteners.

It is essential that any design which requires high fastening torques incorporates appropriated features to allow these large moments to be reacted. For example, it is no use placing an M36 fastener in the middle of a flat plate which requires 2000Nm of torque to be applied – this will not be possible. It is common practice to resist the tool’s reaction force through adjacent fasteners, especially on flange joint assemblies. However, on bespoke mechanical designs it may be necessary to provide dedicated tool reaction points.

3.2.2 Other Tightening Methods

To make even more efficient use of the fasteners, better methods of tightening are available. These include the use of experimentally-determined torque values (ideally, by directly measuring both torque and tension on a sample of the actual fasteners to be tightened), which would allow a tightening factor of 1.4 to be used. Better again would be the use of a calibrated means of directly measuring bolt elongation or tension, for example with RotaBolt inserts. These are not cheap, but allow a tightening factor of 1.2 to be used, making best possible use of the fastener and so might be appropriate for highly critical bolted connections, or where space is limited.

It is worth noting that the **use of an impact wrench to tighten fasteners is not recommended**. Their use should be restricted to nothing more than a rapid method of running nuts into place because their accuracy is so poor. This is illustrated by the fact that VDI assigns a calibrated torque wrench a tightening factor of 4.0.

Details of common tools are contained in a separate guidance note [8]. It should be demonstrated that there are sufficient clearances for use of appropriate tools at the design stage.

3.3 Fastener Maintenance

As discussed in Section 2, one of the major reasons to use fasteners to make a joint is that this allows the connection to be deliberately broken in service for maintenance.

After a prolonged period of service, the friction in a threaded fastener is unlikely to be the same as when first installed. In almost all cases, the coefficient of friction will have increased due to loss of lubricant and formation of corrosion products or calciferous deposits from cathodic protection systems. This can potentially mean that more torque is required to unscrew a bolt than was originally used to install it.

Disconnecting an intact joint is often best achieved using an impact wrench. This compact tool uses percussive force to impart a relatively high (but poorly controlled) torque for a short duration on the fastener but without any need for any static torsion reaction. Thus, for the same reasons that mean it is not suitable for controlled tightening of fasteners, it can assist in breaking out stubborn bolts. It also allows the nuts to be rapidly wound back up the threads, saving time.

Reuse of fasteners is generally allowed, unless there is reason to believe that the bolts may have experienced forces beyond yield. The most common causes of this are uncontrolled tightening during installation or excessive in-service loads. It is possible to test whether bolts have been subject to yield by checking that a nut will spin freely down the threads and past the point of final engagement.

Any fasteners which show signs of misuse should be discarded. If bolts are believed to have been torque tightened without adequate lubrication they should also be replaced. Fasteners made of materials which are prone to galling should not be reused. Any reused fasteners must be thoroughly cleaned and re-lubricated before torque tightening if good control of tension is required.

On flange joint assemblies, ensuring consistency of bolt forces applied is more important for robust sealing performance than achieving the correct tension in any give bolt. For this reason, if it is necessary to replace one or more fasteners, it is recommended that all bolts are changed at the same time. If this is not possible (for example due to schedule constraints), it is important that any new fasteners are distributed evenly around the flange, to avoid gross asymmetry in gasket stresses.

4. SECURING OF FASTENERS

4.1 Non-rotational Loosening

Although loose fasteners are often blamed on vibration, the reason for them losing preload may be nothing to do with this mechanism. Three major causes of non-rotational loosening exist, although only two of these are typically expected to be an issue with Oyster technology.

1. Relaxation: a reduction in the stretch of the bolt due to a change in thickness of the bolted joint. This can be a result of many different mechanisms such as embedment, wear, creep or corrosion.

2. Joint misalignment, which is probably the most serious cause of loosening and can be especially problematic with bolted flange joint assemblies. Guidelines are given in ASME PCC-1-2013 [10] on allowable misalignments of flange joints which should be referred to.

3. Loosening due to thermal issues, such as differential thermal expansion and stress relaxation, as well as brittle failure. None of these mechanisms should be a problem for Oyster and so are not considered in this document.

Bolts are generally made of steel. Steel has a very high modulus of elasticity and so even relatively high forces will result in only a small amount of elongation in most common fasteners. This is best illustrated using a simple example:

Bolt: M20, property class 8.8, free length = 60mm (i.e. 3 x diameter – appropriate)

Tighten to 50% of Yield (0.5*660MPa = 330MPa)

Elastic Modulus of steel = 210GPa, therefore strain @ 50% of yield = 330/210000 = 0.0016

Total length being stretched = 60mm + 0.8*20mm = 76mm (takes into account some stretch in the nut threads and bolt head)

Total elongation = 0.0016*76 = 0.119mm or 119 microns

In absolute terms, 119 microns is a very small amount of extension and it can be seen that this could easily be lost through joint relaxation.

The typical thickness of a galvanised layer is about 40 microns. If the zinc under head of bolt and the nut dissolve, most of tension will be lost. If there's also a galvanised washer in the joint, total tension loss can occur without any rotation.

If used subsea, cathodic protection may prevent this, but it is not possible to be confident which anode will dissolve first and so the preference is not to use galvanised fasteners subsea (see Section 5.3.2).

A typical subsea paint specification is Norsok M501, where the minimum coating thickness of epoxy paint is 350 microns (or 600 microns in the splash zone). Coatings will typically be even thicker at edges (like bolt holes) – resulting in at least 500 microns per painted interface. If paint is under both bolt and nut, total paint thickness = 1000 microns. Thermoset polymers such as epoxy are susceptible to creep at ambient temperatures. Just 5% strain creep in this thickness of paint would be 50 microns – resulting in loss of nearly half of the bolt preload. **It is therefore essential that thick paint layers are excluded from the compression load path in bolted joints.**

4.2 Rotational (vibration) loosening

A Junkers test rig simulates a lateral vibrational loosening mechanism and allows the effectiveness of different securing methods to be determined. Many traditional methods of bolt securing have been shown to be useless in these tests.

Based on the above example of an M20 bolt, with a thread pitch of 2.5mm, all tension can be lost by a rotation of just (0.119/2.5)*360 = 17 degrees, or about one quarter of a flat on a hex head bolt. This shows how sensitive fasteners can be to small amounts of rotation and why some of the traditional securing methods are not effective.

4.3 Effective Fastener Securing Methods

The primary method of securing bolted connections against loosening is to ensure good practice in the bolted connection's design and assembly. Provide sufficient pre-load to prevent any joint movement in service. Maintain this pre-load in service by using long grip lengths (5 bolt diameters if possible), align the joint correctly, and use effective tightening methods.

However, in spite of best endeavours there is always the risk that a joint will see unexpected conditions in service. Therefore, on critical or difficult to maintain fasteners, one or more additional fastener securing mechanisms should be provided. These should be thought of as an insurance policy rather than a substitute for good design.

4.3.1 Nord-Lock Washers

Nord-Lock washers are the first choice of fastener securing method against vibration loosening. These have been demonstrated to work effectively against the vibration loosening mechanism simulated by a Junkers test rig (lateral slip of the bolted joint). They are relatively expensive to purchase, but easy to fit and can be reused if undamaged.

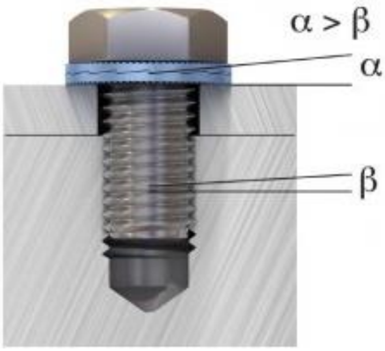

Figure 1: Nord-Lock Washer Geometry

Nord-Lock washers rely on bolt preload to be effective. If bolt preload is inadequate (e.g. due to low initial tension) or lost (e.g. through non-rotational loosening) a Nord-Lock washer will not prevent bolt loosening. This is because rotation at the interfaces between the washers and the other components is prevented by little teeth which bite into the other parts (see Figure 1). Rotation between the two halves of the washer is prevented because the angle of the wedges is greater than the pitch angle of the thread and so any rotation at this interface effectively tightens the joint faster than it can loosen.

The amount of bolt preload required to ensure the washers are effective depends on the materials in the joint. For standard (property class 8.8) bolts, the hardest material in the joint is likely to be the nut (property class 8). In this instance, the washers are only guaranteed to be effective if the bolts are preloaded to at least 30% of their yield stress. Note, that the washers may still be effective at lower levels of preload than this, but this should be considered the best practice lower bound stress for design.

4.3.2 Prevailing torque nuts

Prevailing torque (e.g. Nyloc) nuts generally don't ensure that a joint will retain much preload under vibrational loading. However, in most circumstances they should prevent nuts from falling off completely after loosening has occurred. They are therefore good as a last line of defence against full collapse of bolted joints in the short term. In the longer term, it is likely that wear or fatigue mechanisms will cause total failure of a bolted joint which experiences lateral joint movements if it is only secured with Nyloc nuts.

Prevailing torque fasteners often offer the only effective method of improving legacy joints subsea where fundamental issues exist with the joint (e.g. poorly designed joints with paint in the interface). Note that using this sort of fastener will increase the difficulty in making up the joint.

Other types of prevailing torque nuts are available, including all-metal varieties. However, these are more likely to damage the threads of the bolt, especially if these are coated. For this reason, nuts with polymer inserts are preferred. Repeated use of these nuts will reduce their effectiveness because the inserts become damaged, reducing the prevailing torque they provide. They should therefore be replaced after use (although their primary function will be unaffected).

4.3.3 Thread locking adhesives

Anerobic thread locking compounds (e.g. Loctite) can be effective at preventing vibration loosening. However, these adhesives need to be applied in a controlled fashion to be reliable, which can be difficult to ensure. Unless special measures are taken to assure the quality of the application, adhesives should be treated with caution.

That said, experiments carried out by Aquamarine Power demonstrated that certain Loctite compounds can provide additional security to joints made subsea (based on an increase in measured breakaway torque after curing). The compound must be applied to the threads in the dry, but the fasteners can then be assembled underwater.

Care must be taken to prevent wash-off of the Loctite before the threads can be assembled. In practice, this has been achieved by transporting the fasteners from the surface to the working area inside a flooded plastic box. Both Loctite 243 and 278 have been demonstrated to work in this way.

This approach has been adopted as a standard method of securing hydraulic fittings, where none of the other effective securing approaches discussed are feasible. However, due to the quality control issues and time taken to do this job properly, it is not recommended for normal bolted connections.

4.3.4 Welding

Welding to secure fasteners can be quite effective, and was used on the majority of large structural fasteners on Oyster 800. However, this approach has a number of drawbacks. The heat applied during welding has the potential to damage the fastener material. During the fabrication of Oyster 800, some welds between nuts and the main structure cracked due to the poor weldability of standard fasteners. Experience gained during offshore maintenance has shown that taking apart joints where the fasteners have been welded can take a very long time, leading to substantially higher maintenance costs as a result of this increased dive time. This method should therefore not be used on any joint where maintenance in service can be anticipated.

4.4 Ineffective Fastener Securing Methods

Helical spring washers are worse than using nothing at all in applications associated with WECs, and their use is not recommended.

Jam nuts/double nuts are generally ineffective unless used in a very specific way. This method required is too complicated to be reliable, and their used is not recommended.

Wire locking is an expensive operation which requires skill to perform correctly. During the Oyster 800 build, examples were seen of poorly executed and even counter-productive (i.e. reverse wound) attempts at wire locking. Even if done well, it will be ineffective at preventing significant loss of preload (see above example for how little rotation causes substantial loosening in typical connections). The primary purpose of wire locking is to prevent total loss of nuts if they do come loose. As such, it is not recommended to use for applications associated with WECs.

Tab washers are generally not strong enough to prevent fastener rotation because they have to be weak enough to bend with small hand tools. The forces at work in a bolted joint are normally significantly larger than this, and so the soft metal is easily bent back out of shape. The sharp edges of the tabs can also be a cause of injury, especially whilst bending them into place. As such, their use is not recommended.

5. FASTENER SPECIFICATION

5.1 Fastener Forms

5.1.1 Bolts or Studs?

For equipment which may need to be maintained, bolts are generally the better choice and so should be selected preferentially. This is because when disassembling equipment, it is not always possible to control which nut will wind off a stud. This can make the job much more difficult, especially subsea.

However, bolts are slightly more restrictive in how they can be inserted and removed, and so access around obstructions such as pipe bends needs to be more carefully considered. Note that there is no strength advantage to either studs or bolts – both can do the same job. Studs may also offer certain cost and lead time advantages for procurement, especially if the design calls for unusually long fasteners, or for diameters above about M36, and so may be used if there is a good reason to do so. However, be sure to consider through life costs when making this decision – increasing dive time to slightly reduce the capital costs of fastener purchases is unlikely to save money overall.

Irrespective of whether bolts or studs are selected, it is essential to specify the correct length. Too short is not simply good enough, but too long can be nearly as bad, particularly if these fasteners are ever going to be used subsea because of the amount of dive time wasted winding nuts up/down the excess length. It also limits the choices of tools available for tightening the nuts.

Don't add extra length to allow the components to be drawn together during assembly – this requirement is better addressed by specifying some threaded bar as a dedicated installation tool.

If using bolts, make sure that the threaded length is sufficient. Unless specified otherwise, bolts will come with standard threaded lengths, as defined in BS EN ISO 4014:2011 [9]. A selection of standard threaded lengths is reproduced in Table 1 for easy reference. If these lengths are inadequate, it is possible to specify fully threaded bolt shanks (also known as machine screws). However, note that in connections designed to withstand significant shear loads it is bad practice to include threads within the shear interface.

Bolt Size	Standard Thread Lengths (BS EN ISO 4014)		
	Bolt ≤ 125mm	Bolt 125 – 250mm	Bolt ≥ 250mm
M12*	30	36	49
M16	38	44	57
M20*	46	52	65
M22*	50	56	69
M24	54	60	73
M27*	60	66	79

M30*	66	72	85
M36	-	84	97
M48	-	108	121

Table 1 – Standard threaded lengths of bolts (sizes marked * are non-preferred, see Section 5.2)

5.1.2 Hex Head or Socket Head?

Hex head bolts are preferred to socket head cap screws. By standardising on one type, this allows the use of common tooling where possible and the hexagon bolt head will be the same size as the nut used on the same fastener.

Another reason not to use cap head screws is that once painted, it can be very difficult to clean out the socket to get a good interface for a hex drive, making them more difficult to maintain.

Finally, socket cap screws are quite commonly supplied in material property class 12.9 (or higher), which is inappropriate for subsea use because its high hardness makes it susceptible to hydrogen induced stress cracking when exposed to standard cathodic protection electrical potentials. The risk of using inappropriate materials is therefore decreased by avoiding these components wherever possible.

However, there can be space benefits to using cap screws and so they may have a place in some areas of the design. They are often seen on bought out assemblies. If they are to be used, care must be taken not to over-paint the sockets and the material grade used on bought out items should always be checked; property class 8.8 should normally be the maximum allowed bolt strength (see Section 5.3).

5.1.3 Tapped holes or nuts?

In almost every case, **nuts should be used rather than tapped holes.** Tapped holes can be difficult to manufacture, resulting in a risk of broken tools, lost time and rework. Blind tapped holes are especially poor details which are even more difficult to make and can cause particular difficulty with assembly and subsequent corrosion if a joint is assembled subsea. If a joint has to be taken apart in service, problems can occur if studs wind out of tapped holes rather than nuts coming off the stud.

Finally, if a bolt is wound into a tapped hole and the fastener seizes, it can be almost impossible to take the joint apart again. However, there may be certain circumstances where other requirements such as the need for single-sided access or robust sealing dictate that tapped holes are required.

Note, particular manufacturing issues have been experienced when trying to cut internal threads in titanium. This material's high strength and tendency to work harden and gall, combined with its low modulus of elasticity (resulting in high spring-back and "grabbing" of the tool) make traditional tapping of holes especially difficult. If internal threads need to be cut in titanium, it is strongly recommended that a CNC thread milling approach is considered.

5.1.4 Plain washers or not?

Plain washers spread bolt loads, reducing the bearing stress under bolt heads and nuts. They are generally only required for this purpose in lower strength materials (e.g. plastics, cast iron, etc.) or where slotted holes are specified. Their use can prevent damage to the base material.

Hardened plain washers also provide a smooth surface which helps to reduce friction and maintain the torque-tension relationship more consistently than may be the case if fasteners are tightened directly onto base materials. This makes them especially useful for flange joint assemblies where consistency of bolt load is important for gasket performance.

Note that soft washers are unlikely to provide either of the above benefits. It is important that plain washers are specified as through hardened (not just surface hardened). A manufacturing specification for through hardened washers can be found in ASME PCC-1-2013 [10].

Plain washers should never be used in addition to Nord-Lock washers (see Section 4.3).

Nuts and bolts can be bought with integral flanges which increase their diameter, removing the need for separate washers. These are very commonly used in applications such as automotive OEM equipment. If washers are required in the joint using flange head fasteners instead can save time during assembly due to the reduced part count. Eliminating one interface in the joint also means that embedment losses in service are reduced.

The only technical drawback to using flange head fasteners is that a slightly increased torque is required to achieve same bolt tension, due to increased radius at which head friction acts. Flange headed fasteners are only easily available in relatively small sizes (e.g. up to M16), although larger diameters could be produced as a special order. Flanged fasteners *can* be used in combination with Nord-Lock washers.

5.2 Standard Sizes

A wide range of fastener sizes were specified for use on Oyster 800. In future, it would be very helpful to reduce this variety. Unless there are strong technical reasons to use different sizes, the following guidelines should be followed:

Preferred fastener size: M16 x 2

These are large enough to be easily handled by a diver wearing gloves and the threads are robust enough to withstand a small amount of damage or corrosion without risking failure. However, they are small enough to be effectively tightened using standard hand tools.

Smallest fastener to be exposed to offshore or onshore environments: M12 x 1.75

M16 fasteners are preferred for applications where divers need to handle the bolts, but when space is tight and loads are small, a slightly smaller fastener is acceptable. Even smaller bolts may exist in bought-out items but, where possible, their inclusion should be challenged.

Preferred fasteners for structural connections: M24 x 3; M36 x 4; M48 x 5

If M16 fasteners are insufficient to resist the applied loading at a joint, a larger size of fasteners should be chosen from the above list wherever possible. Note that these bolts will generally require hydraulic tooling to tighten effectively, especially if they are to be assembled subsea or where space to wield large torque wrenches is not available.

The above standard sizes are guidelines only, and there may be good reasons to make exceptions. For example, in permanent and/or very heavily loaded structural connections, larger fasteners (e.g. M64) may be required which should be considered on a case by case basis. Intermediate bolt sizes may also be required on hydraulic flanges to conform to standard designs.

Note that these standard sizes are all specified with metric coarse threads to BS 3643-1:2007. In the oil and gas industry it is common to find unified threads specified, especially when dealing with pipe flanges. Unless there is a clear reason to use these (for example, to interface with a threaded hole in a bought out component) use of the metric equivalent fastener is recommended.

5.3 Materials

5.3.1 Base Materials

Low alloy steels are most commonly used for making threaded fasteners. In general, the higher the strength of the material used, the more efficient and robust the connection design becomes. However, above a certain strength, fasteners used in seawater with cathodic protection are known to be susceptible to stress corrosion cracking (SCC).

Property class 8.8 fasteners are the highest strength bolts that should be used in applications that have (or may occasionally get) cathodic protection. They should be used with matching (property class 8) nuts. Common comparable grades of fasteners used in the oil and gas industries which are also acceptable for subsea service under cathodic protection are B7 (ASTM A193) or L7 (ASTM A320) bolts with 2H (ASTM A194) nuts.

Higher strength fasteners can be considered for use in special circumstances where the additional strength has significant advantages. However, special attention must be given to reducing the risk of SCC in these fasteners, which is outside the scope of this document.

Washers are usually harder than the recommended maximum for use with cathodic protection (approximately 34 HRC or 300-350 HV). However, the stresses in these components are entirely compressive. In the absence of tension there is no risk of cracks developing even if the materials become brittle, and so this is not considered to be a problem.

Stainless steel fasteners have a number of disadvantages compared to low alloy steels. They cost more, they are lower strength and they have a significantly higher risk of galling during assembly and disassembly. Furthermore, standard “marine” grade austenitic stainless-steel fasteners “A4” are not sufficiently corrosion resistant to use reliably in submerged applications without cathodic protection. Super austenitic stainless steels are more highly alloyed than standard austenitic grades. 254SMO is one such alloy, which has been widely used on Oyster because it is a standard material option for Nord-Lock washers.

Duplex stainless steels suffer from most of the drawbacks of conventional austenitic stainless steels, plus they are more susceptible to SCC. However, they have the advantage being inherently resistant to corrosion when used in cold seawater. Note that Norsok M-001 specifically recommends against using Duplex stainless steel as a bolting material due to the risk of crevice corrosion. However, advice received from Aquamarine Power’s specialist materials consultant suggested that this risk was small. They were therefore used on Oyster 800 in some areas where cathodic protection could not be assured (such as the splash zone and inside assemblies where CP may not penetrate). No failures of these fasteners have been reported at the time of writing (Jan 2015).

The offshore oil and gas industry typically uses more corrosion resistant materials such as super duplex stainless steel, 6Mo austenitic stainless steel or Inconel 625 fasteners in cold sea water applications. Other highly corrosion resistant copper, steel and nickel alloys are also available. However, the cost of these fasteners is generally prohibitive for wave energy devices, and so they have not been used on Oyster.

Titanium fasteners have occasionally been used on Oyster 800. Titanium is immune to corrosion in seawater and common alloys such as Grade 5 (Ti-6Al-4V) offer a combination of high strength and low elastic modulus which makes for robust bolted connections. However, titanium can suffer from SCC under cathodic protection and so should only be used if it can be guaranteed to not be electrically connected to any sacrificial aluminium or zinc anodes. It also is susceptible to galling and so care must be taken to prevent this by using a suitable lubricant. Large titanium fasteners can be expensive and time consuming to source, but small components (up to about M12) are mass manufactured and readily available.

5.3.2 Coatings

Coatings can be applied to fasteners to modify or enhance their performance. Many different coatings are available and a wide selection has been trialled on Oyster 800.

Bright Zinc Plating (BZP) offers limited corrosion protection to low alloy steels. It is generally suitable to protect fasteners from corrosion during storage and for a limited time on the quayside, but it offers relatively little protection once in service. However, this is sufficient to ensure a reasonably consistent torque/tension relationship is maintained during tightening, provided the fasteners have been properly lubricated. Long term corrosion protection needs to be provided by alternative means such as via connection to an external cathodic protection system. The coating is conductive, which means that this is relatively simple to ensure. The plating should preferably be no more than 8 microns thick, which is thin enough so that no base material needs to be removed from the fasteners to enable assembly. To reduce potential tension loss due to corrosion, a thinner coating is preferred. The purchasing specification for zinc electroplating of 5-micron thickness with a clear passivate is FE/ZN5/A to BS 7371-3:2009.

Galvanized coatings provide much thicker layers of zinc than electroplated finishes. They therefore offer significantly improved corrosion resistance and have been effective at protecting exposed fasteners on the onshore site at Billia Croo. This coating is also specified by some oil and gas companies for subsea application on flange joint assemblies where cathodic protection is available. However, as discussed in Section 4.1, there is a risk that loss of this thick coating due to galvanic corrosion may still occur even with an effective CP system, resulting in unacceptable loss of preload in joints. Galvanizing is therefore not recommended for use on subsea joints.

Note also that due to the thickness of the galvanizing zinc coating, base material has to be removed from the fasteners prior to coating. This removal of material has two potential adverse consequences, which are:

1. An increased risk of thread stripping during tightening because the strength is compromised.
2. A risk that galvanized fasteners bought from different sources may be incompatible.

It is normal practice in the UK to remove twice the anticipated zinc coating thickness from the nut before galvanizing. This provides the most mechanically robust solution because the nut is stronger than the bolt. However, it means that the coated bolt threads will be over-sized and so normal nuts will no longer fit and they will not screw into standard tapped holes. For this reason, some suppliers may remove material equal in thickness to one zinc coating from both the nut and the bolt (this is more common in some parts of continental Europe). It is therefore good practice to **always buy galvanised nuts and bolts as matched pairs from the same supplier and check that they spin freely over each other** (without excessive play) as part of any goods inward inspection.

A technology which has been used quite extensively in the offshore industry are Xylan coatings. These are coating systems in which thin layers of high performance polymers are applied after a base treatment with low to moderate corrosion resistance such as phosphating. When done properly, these coatings offer an excellent combination of corrosion resistance (via the polymer barrier) and inherent low friction properties (the top coat usually consists of a fluoropolymer). The coatings are thin enough so that there is no need to remove additional material from the threads, which is also good for fastener integrity and interchangeability. However, the polymer coatings used are electrically insulating, which means that cathodic protection will not be provided to fasteners if this coating remains intact where the fasteners contact the structure. These thin polymer coatings are prone to damage in service (often at fastener extremities and where torque tightening equipment has been used), which then potentially leaves parts of the steel fasteners exposed to free corrosion. There will also be a small loss of bolt tension due to creep in the polymer coatings under the high local stresses used in bolts. Their use has been trialled by Aquamarine Power on some offshore flanges with reasonable success. However, this is not a technology which is preferred for new designs.

Other coating technologies exist which can offer different advantages and disadvantages. Some of these might prove useful in specific circumstances. The main issues to consider when deciding whether to use a given coating are as follows:

- Coating thickness:
    - Does base material need to be removed (compromising strength and introducing compatibility issues)?
    - Could the coating thickness change in service (e.g. due to creep or corrosion) leading to a change in preload?
- Coating conductivity: will cathodic protection be available to the fastener if it is to be used submerged?

- Coating friction: does the coating affect the friction conditions assumed in the bolt load calculation?
- Coating toxicity: some coatings with otherwise good properties may be hazardous to either operators or manufacturing personnel, or to the environment (e.g. cadmium coated fasteners).

5.3.3 Standard Materials

The following material combinations have been selected as the base case for standard applications on future Oyster designs. Deviations from these standards may be required for specific applications, but the following should be adopted where possible to ensure good performance and commonality of spares.

Offshore connections with cathodic protection

- Low alloy steel, property class 8.8 bolts with Bright Zinc Plating (FE/ZN5/A to BS 7371-3)
- Property class 8 nuts with a polymer prevailing torque insert (e.g. Nyloc)
- Stainless Steel 316 Nord-Lock washer pairs under both the nut and the bolt head
- Bolt threads to be lubricated with Molykote 1000 paste before assembly
- The components to be connected must have bare metal on the compression mating faces and under the Nord-Lock washers to ensure electrical continuity throughout. This should be demonstrated by electrical continuity testing wherever practical.

Note, very small fasteners (i.e. the non-preferred sizes smaller than M12) should be made of stainless steel grade A4 without any coating. The loss of even a small thickness of zinc on such small fasteners can make a significant difference to bolt performance, and so avoiding this issue means that the disadvantages of using stainless steel are outweighed. It is extremely important to lubricate these fasteners carefully due to the risk of galling, and they should not be reused for the same reason.

When connecting low-strength base materials, it may be necessary to reduce the stresses under the head of the fasteners. In this case, flanged nuts and bolts should be selected, along with oversized Nord-Lock washers (provided sufficient preload can be achieved to activate the washers).

Offshore connections without cathodic protection

- Duplex stainless steel (a 22% Cr alloy such as F51, UNS 31803, S32205, etc) bolts and nuts with no coating
- 254SMO Nord-Lock washer pairs under both the nut and the bolt head
- Bolts threads to be lubricated with Molykote 1000 paste before assembly

Onshore connections

- Low alloy, property class 8.8 bolts and class 8 nuts, both with galvanised coating to (ZnG to BS 7371-6)
- Galvanized, through-hardened washers should be used under both bolts and nuts in most applications to reduce stresses under the fastener heads and to provide a suitable surface to maintain a consistent torque/tension relationship
- Bolt threads and the surface underneath the bolt head and nut must be lubricated with Molykote 1000 paste before assembly
- Always buy galvanised nuts and bolts as matched pairs from the same supplier and check that they spin freely over each other

Note, very small fasteners (i.e. the non-preferred sizes smaller than M12) should be made of stainless steel grade A4 without any coating. Galvanising on such small fasteners could result in an unacceptable loss of base material and compromise to strength. It is extremely important to lubricate these fasteners carefully due to the risk of galling, and they should not be reused for the same reason.

6. REFERENCES

[1] OY800-PPT-0013 - Selected Threaded Connection Failures - Rev A1

[2] OY800-TN-0185 – Design & Installation of Resin Bonded Rock Anchors

[3] OY801-TN-0033 - Standards for Hydraulic Fittings

[4] OY800-TN-0148 – Static Analysis of Welded Interfaces using DNVGL-OS-C101

[5] OY800-TN-0187 – Pinned Connections Design Guidelines

[6] OY800-CAL-0099 – Bolt Torque Calculator Template

[7] VDI 2230-1: 2015 – Systematic calculation of high duty bolted joints: Joints with one cylindrical bolt, Verein Deutscher Ingenieure (The Association of German Engineers). See also VDI 2230-2:2014 - Systematic calculation of high duty bolted joints: Multi bolted joints.

[8] OY800-TN-0191 – Tooling Guidance

[9] BS EN ISO 4014:2011 – Hexagon head bolts. Produce grades A and B, BSI

[10] ASME PCC-1-2013 Guidelines for Pressure Boundary Bolted Flange Joint Assembly

APPENDIX A: SUMMARY OF MAJOR RECOMMENDATIONS

Connection Design

Best practice in bolted connection design ensures a rigid compression load path, with a relatively flexible tension load path through the bolts. It is essential that thick paint layers are excluded from the compression load path in bolted joints.

Assign forces to individual bolts within a joint based on the assumption of an elastic distribution of load.

Assess the load on each individual bolt using OY800-CAL-0099.

Fastener Forms

For equipment which may need to be maintained, bolts are a better choice than studs. Ensure sufficient threaded length is provided. Hex head bolts are preferred to socket head cap screws. Nuts should be used rather than tapped holes. Threads should generally be metric coarse profile to BS 3643-1.

Standard sizes

- Preferred fastener size: M16 x 2
- Smallest fastener to be exposed to offshore or onshore environments: M12 x 1.75
- Preferred fasteners for structural connections: M24 x 3; M36 x 4; M48 x 5

Base case offshore connections with cathodic protection:

- Low alloy steel, property class 8.8 bolts with Bright Zinc Plating (FE/ZN5/A to BS 7371-3)
- Property class 8 nuts with a polymer prevailing torque insert (e.g. Nyloc)
- Stainless Steel 316 Nord-Lock washer pairs under both the nut and the bolt head
- Bolt threads to be lubricated with Molykote 1000 paste before assembly
- The components to be connected must have bare metal on the compression mating faces and under the Nord-Lock washers to ensure electrical continuity throughout. This should be demonstrated by electrical continuity testing wherever practical.

Base case offshore connections without cathodic protection

- Duplex stainless steel (a 22% Cr alloy such as F51, UNS 31803, S32205, etc) bolts and nuts with no coating
- 254SMO Nord-Lock washer pairs under both the nut and the bolt head
- Bolts threads to be lubricated with Molykote 1000 paste before assembly

Base case onshore connections

- Low alloy, property class 8.8 bolts and class 8 nuts, both with galvanised coating to (ZnG to BS 7371-6)
- Galvanized, through-hardened washers should be used under both bolts and nuts in most applications to reduce stresses under the fastener heads and to provide a suitable surface to maintain a consistent torque/tension relationship
- Bolt threads and the surface underneath the bolt head and nut must be lubricated with Molykote 1000 paste before assembly
- Always buy galvanised nuts and bolts as matched pairs from the same supplier and check that they spin freely over each other.