OTC 21621

Improved Concentric Thread Connectors for SCRs and Pipelines

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Abstract

Concentric thread mechanical connectors have been developed and qualified to ISO 21329 for offshore pipelines and risers. The connector design consists of a conical box and pin with interlocking concentric grooves and redundant metal-to-metal seals. Instead of make-up by screwing, it is designed to “snap” together under simultaneous axial clamping and hydraulic pressure in the thread annulus. After make-up, the connector cannot unscrew, back-off or loosen under dynamic loading and/or thermal cycling. Concentric thread connectors have been used in various tubular applications for more than 20 years. However, recent ISO qualification tests and modern FE analyses have been used to identify and design significant improvements. These improvements include: preloading of ID and OD abutment shoulders; elimination of a small gap at the connector ID; elimination of hot spot stresses; and lockout of any box-pin slippage due to extreme torque loading. In addition, smaller (6-5/8-in) concentric thread designs that are integral to high strength pipe are available. The improved concentric thread design has approximately 10x more fatigue life than a field welded pipeline. It is also leak-tight and stronger than the pipe body in destructive testing, i.e., bending, tension and hydrostatic pressure to failure. These improved connectors enable pipeline construction with lighter, high-strength steel, or with corrosion resistant cladding. Since connected pipe joints are completely fabricated onshore, special materials and fabrication processes have no impact on pipelay rates. The improved concentric thread connector is now qualified and project ready for deepwater Steel Catenary Risers, Oil Offloading Lines and subsea pipelines in locations such as the Gulf of Mexico, West Africa, Brazil and the North Sea. Also, pipelay with concentric thread connectors may be done faster with lower cost modular systems in lieu of mobilizing welding pipelay vessels with high cost day rates.

Introduction

A mechanical connector with concentric threads has been designed and qualified for use in SCRs, flowlines and other subsea pipeline applications. The concentric thread design consists of an internally tapered box which mates with an externally threaded and tapered pin. Instead of screwing together like helically threaded connectors, the connector is snapped together under simultaneous axial clamping and pressurization of the thread annulus. Disconnection is performed by simply reversing the connection process of clamping and pressurizing the annulus. The connector is ready for use on deepwater FPS risers in locations such as the GOM, West Africa, Brazil and the North Sea.

Mechanically Connected Pipelines and Risers

Deepwater pipelines and risers are usually welded on location and deployed from J-lay, S-lay or Reel Lay vessels. These vessels typically have day rates in the hundreds of thousands of dollars and can cost millions of dollars to mobilize. Installation with mechanical connectors would simplify the construction process and significantly speed-up the overall operation. Also, since offshore welding and NDT would be eliminated, the overall spread costs would be reduced.

The onshore fabrication of connected double, triple or quad joints can be done ahead of time and out of the critical path. Pipe joints which require high fatigue resistance can be made with double-sided welds which are ground flush and inspected to tighter NDT criteria. This would raise the fatigue rating from DNV E to DNV C classification, resulting in a fatigue life which is 10x more than field welded joints.

Connectors which are capable of meeting qualification requirements would yield several advantages over conventional pipelay such as:

- Faster pipelay which reduces lay vessel time and spread costs.
- Pipelay with pre-fabricated pipe joints - complete with connectors, coatings, insulation and nondestructive testing.
- Approximately 10x higher fatigue life of riser critical sections.
- Pipeline construction with lighter, high-strength steel or with corrosion resistant materials and/or cladding - having no impact on lay rates.
- Modular pipelay equipment on more flexible and lower cost DP construction vessels.

**Concentric Thread Design.**

The concentric thread connector is normally welded onto pipe, although it can also be machined integrally onto pipe upsets. Weld-on connectors allow easy access to the pipe ID and OD so that double-sided welding is possible. Weld neck extensions and flush-ground welds eliminate stress concentration factors, and when combined with enhanced NDT criteria, single joints can meet DNV C fatigue curves. Double, triple or quad pipe-to-pipe joints can be honed internally to match the fatigue life of the connected ends, or the use of single joints in fatigue critical locations can make this extra preparation unnecessary.

The initial stab-in position of the connector is shown in Figure 1 with contact between the box and pin threaded surfaces. The stand-off is shown in the magnified view on the right, and final make-up and preloading is achieved with an axial movement of only 0.6-in. Alignment and snap-in of the concentric threads is practically assured since the connector is about 95% made-up after stab-in.

The connector is made-up with an external clamping tool, using annulus pressure to expand the box and shrink the pin. Figure 2 shows the connector in the made-up position with a close-up of the metal nib seal forced into the sealing groove. Two nib seal locations at the inner and outer diameter of the connector provide four redundant metal-to-metal seals in the fluid leak path.

**Connector Testing and Qualification.**

A 20-in OD x 1-in WT concentric thread connector was designed was tested in accordance with ISO 21329, i.e., “Petroleum and natural gas industries —Pipeline transportation systems - Test procedures for mechanical connectors.” This standard specifically addresses connector qualification for pipeline and riser applications. The scope and order of testing is shown in Figure 3 below. (Note: Connector data and test parameters are listed under Nomenclature in this paper.)

![Figure 1. Connector in Stabbed Position](image1)

![Figure 2. Connector after Make-up](image2)

![Figure 3. Connector Testing Requirements per ISO 21329](image3)
Installation and Service Load Testing.

The test program required twelve connector sets which were subjected to: 1) make and break, 2) torsional breakout resistance, 3) installation bending and external pressure, 4) hydrostatic testing and bending, 5) cyclic operational restrained and unrestrained testing, 6) limit load testing and 7) fatigue testing to failure. The test sequence simulated connector loading history, essentially duplicating installation, commissioning, service and extreme loading, and final fatigue to failure. Descriptions of the various tests are detailed in the sections below.

Make and Break Testing.

In accordance with the ISO plan, a minimum of 5 make-ups and breakouts were performed on test articles 1 through 8. After each breakout, the connectors were inspected for galling or other damage that might interfere with the performance or leak tightness of the connector. After each breakout the connector surfaces were visually inspected, photographed and then setup for reconnection. (See Figures 4 and 5 below.)

Torque Testing.

Following make and break, test articles 1 - 8 were subjected to a static torque load of 158,233 lb-ft in accordance with equation in C.2 in Annex C in ISO 21329. The test was performed by centralizing the connected test strings within a larger pipe, fixing one end of the string to the outer pipe, and then applying a load through torque arms on the free end of the string. The test load was held for one hour, and the abutment line of each connector was marked with witness lines to detect any slippage between the individual connections.

Installation and Hydrostatic Load Tests.

As shown in Figure 6, installation cyclic load conditions were applied on two test strings containing four test articles in each string. These tests simulated installation bending and external hydrostatic pressure loads for a depth rating of 6,562 ft (2000 m). Installation loading was then followed by hydrostatic testing plus bending. This was the most highly stressed service load case, thus simulating riser commissioning and 24-hour hold periods.

Unrestrained/Restrained Load Testing.

“Unrestrained” conditions are indicative of a pipeline or riser segment that is not axially constrained, and would also have axial tension due to self-weight. “Restrained” conditions simulate pipeline axial restraint with cyclic compressive loads (i.e., no bending), with temperature cycling and constant internal pressure. A total of 20 combined loading cycles were applied for each unrestrained and restrained conditions. Unrestrained testing included cycles of bending, tension, high and low internal pressure and temperature. Testing in the restrained condition applied cycles of compression, high and low internal pressure and temperature.
Service load testing was applied to simulate loads due to operational cycling. The combined loading conditions were carried out in a 6,000 kip test bed (Figure 7). It was modified with an external hydraulic bending frame connected to the test string to apply the bending moment. Trunion swivels connected to the end of the test string allowed the tension load to be applied in a straight line, i.e., independent of the bending curvature of the test string. Induction coils were used to heat the connectors, while a refrigerated ethylene glycol mixture was sprayed on for cooling (Figure 8).

**Limit Load Testing.**
Destructive testing was done to determine the structural limits of the connector in tension, internal pressure and bending. These results were used for correlations to strain gage and finite element analysis, and are also useful for establishing the reliability limits in field conditions. In each test the connector was proven to be stronger than the pipe, and no leakage or yielding was detected in the connector.

**Tension + Pressure.**
The tension test with internal working pressure was performed in the 6,000 kip load frame. The test setup for applying simultaneous tension and constant internal pressure is shown in Figure 9.

The tension failure occurred well away from the connector in the pipe body. (See Figure 10.) No leakage or yielding was detected in the connector throughout the test.

**Bending + Pressure.**
The limit load for bending was done in a 4-point bending arrangement which was rigged into a 5,000 ton load frame. (See Figure 11.) The test consisted of applying a constant bending moment across the connector with simultaneous internal working pressure. The test unit was loaded until gross yielding was observed in the pipe. Throughout the test, no leakage or separation was witnessed in the connector.
Pressure to Failure.

The pressurization test consisted of hydro pressurizing a connected test pup with heavy bulkheads, previously tested in the operational restrained test string. The test pup was contained in a test pit and was pressurized until failure at a rate of 1,500 psi per minute. As shown in Figure 13, the test article failed by bursting with a longitudinal fracture. The connector body remained leak-tight and did not plastically deform throughout the test.

Fatigue Testing.

Six connector test articles approximately 26-ft long were fabricated for fatigue testing. The test units consisted of two connectors that were previously tested through service loading and four untested connectors. Each unit was tested at 26 Hz in a resonance rig, which applied bending stresses 360 deg around the pipe circumference with simultaneous hydrostatic pressure. (See Figures 14.) The test articles were tested at low, medium and high pipe stress ranges from approximately 17 to 28 ksi. The hydrostatic pressure in each test article was also adjusted to perform the fatigue at constant R-ratios of -0.67 and -0.19.

The pipe-to-connector girth welds in the fatigue units were prepared to increase their fatigue life as much as possible. In this case, the welds were made to DNV-RP-C203 C-curve requirements. Therefore, each weld was double-sided, ground flush in the ID and OD of the pipe, and then nondestructively inspected to DNV Level 1 requirements (Special Category). The DSAW welds in the 20-in x 1-in pipe were also ground flush in the ID and OD for a distance of about 4 ft from each connector girth weld.

During the fatigue testing of each unit, fatigue failure was defined by the detection of internal pressure loss and/or leakage. A typical fatigue failure is shown in Figure 15, where a through-crack is detected in the pipe DSAW seam. As predicted in analysis, the fatigue life of the test units were limited by the connector-to-pipe girth welds and/or the DSAW pipe welds. Failure points for the six test articles are plotted on the S-N curve in Figure 16. These data show that the welds in the connected pipes performed in accordance to the BS7608 C mean curve for welds.

Note: BS Curves were used by testing house for reference information.
Connector Qualification and Certification.

All connector testing was performed in accordance with ISO 21329 using independent testing house facilities, which were also in compliance with ISO 21329, and “ISO 9001:2008 Quality management systems — Requirements.” Testing procedures were reviewed and witnessed by a 3rd party verification agency. A Statement of Conformity (Figure 17) was issued for the test summary and confidence level listed in Table 1 below.

![Figure 17. Fatigue Failure Points vs. BS7608 B and C Curves](image)

**Table 1. Connector Testing and Confidence Summary**

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<th>Confidence Level</th>
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<td>Normal</td>
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<td><strong>Number of Test Samples</strong></td>
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<tr>
<td>Final make-up</td>
<td>All</td>
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<tr>
<td>Reverse torque</td>
<td>1 to 4</td>
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<tr>
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<tr>
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<tr>
<td>Operational unrestrained tests</td>
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<tr>
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<td>(Med. Stress)</td>
<td>3, 12</td>
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<tr>
<td>(High Stress)</td>
<td></td>
</tr>
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</table>

*Note: Shaded areas indicate confidence level and test sample numbers tested per Figure 3.*

Further Connector Improvements.

In the course of qualification testing and further analysis, it became apparent that improvements could be made to the concentric thread design. Opportunities for improvement included:

- Elimination of the ID abutment gap and preloading the connector abutment ID.
- Increasing the resistance to torsional slip (>40% of torsional SMYS).
- Reduction of stress risers and SCFs for better fatigue resistance.

**Elimination of ID Gap.**

As discussed previously, the concentric thread connector is designed to snap together via axial clamping and pressurization in the threaded annulus. Upon pressurization of the threaded annulus, the axial clamping force is increased, thereby moving the box and pin into the connected position. As the connector moves into the connected position the tapered box and pin threads slide over each other on the flat crests of the threads. This procedure, through axial clamping force and/or threaded annulus pressurization, causes the threaded diameters of the box to expand and the pin to shrink. Therefore, and according to Poisson’s Ratio of the connector materials, the lengths of the box and pin become respectively shorter and longer. When the threads snap into their made-up position, the box and pin are essentially restored to their lengths prior to make-up.

The changes in the threaded lengths of the box and pin during axial clamping and pressurization cause the radial ID abutment surfaces to move towards each other. Conversely, the radial abutment surfaces on the connector OD move apart. When snap-in occurs the box and pin lengths are restored, which results in a preload at the OD abutment surfaces. However, the ID abutment surfaces move apart, leaving a gap on the order of 0.010 in.

The design and application of a metal spacer ring at the ID abutment accommodates the over-extension due to Poisson’s Ratio during make-up, and then seals and preloads the ID abutment upon make-up. This improvement eliminates any ID gap under all connector loading conditions, including extreme bending. (See Figure 18.) It precludes crevice corrosion at the ID nib seals, and because it is preloaded after make-up, it prevents movement of the ID sealing nib. In turn, this prevents fatigue wear in the ID nib seal and premature leakage into the threaded annulus.

**Increased Resistance to Torque.**

Torsional loading is present in any non-vertical riser or pipeline which has experienced plastic bending during installation. The mechanism for generating this torque is caused by the interactions between the pipe bends and the gravitational loads on horizontal or near-horizontal sections of a pipeline. Figure 19 illustrates how plastic bending results in “dog-legs” in pipeline lying on the seafloor. The weight of the dog-leg causes torque as it tends to rotate about the pipe axis. As a result, if such a pipeline lying on the seafloor were cut, the two ends would displace laterally in the same direction due to residual bending moment, and the ends would rotate about their axes in opposite directions to relieve residual torque.

In the case of helically threaded connectors which are screwed together, the combined effect of constant torque with a large number of pressure and temperature cycles, can cause a threaded connection to “back-out” and leak. Hence, the need for “operational restrained and unrestrained testing” discussed above.

![Figure 18. FEA Cross Section of Connector ID and OD Abutments](image1)
![Figure 19. Example of a Single and Adjacent “Dog-legs” in a Pipeline](image2)
Although it has been proven that helically threaded connectors can be “locked-out” with adhesive systems, an advantage of the concentric thread connector is that it cannot unscrew or loosen by this mechanism. However, it is also desirable to prevent any relative slippage between the connector box and pin so that the metal-to-metal seals do not slide or gall.

Based on field experience and analysis of “worst case” plastic bending, a safe resistance to back-out for threaded connectors is about 40% of yield for pipelines and flowlines, and 10 percent of yield for non-vertical catenary-type risers. The torque testing requirement in ISO 21329 for the 20-in x 1-in API X65 concentric thread connector is only 158 kip-ft; whereas, 675 kip-ft would be required to reach 40% of torsional yield strength a per Equation (1) below.

\[
Tor_y = \frac{0.577SJ}{r}
\]  
(Eq. 1)

where
- \(Tor_y\) is the pipe body yield torque
- \(S\) is the pipe yield strength of test samples
- \(J\) is the polar moment of inertia and equal to \((\pi/32)(OD^4 - ID^4)\)
- \(OD\) is the outer diameter
- \(ID\) is the inner diameter
- \(r\) is the pipe outer radius

In order to reach this level torsional resistance to slip, the concentric thread connector was modified by knurling a number of the thread loading flanks on the pin OD. (See Figure 20.) In the connector made-up configuration, the knurling peaks are under significant compressive force, and the coefficient of friction between the box and pin is greatly increased, thereby increasing the resistance to slippage between the mating threads. The FE simulation shown in Figure 21 was used to determine knurling parameters for the desired torque resistance as discussed above.

Full-scale torsion testing was also performed to measure the torsional slip resistance of a knurled connector. Torque was applied to a connected pup joint until slip occurred. As shown in Figure 22, the connector slipped only after a torque greater than 700 kip-ft was applied, thus exceeding the 675 kip-ft target. As a result, this testing validated the design approach and the ability to achieve a very large resistance to torsional slippage via knurling on a concentric thread connector.

As noted, this level of torsional slip resistance is quite large due to the worst-case calculation and the large size of the connector under consideration (20-in x 1-in). Torque resistance of 40% of pipe yield strength should be more easily achieved in smaller diameter pipes, as the torque to cause pipe yielding (\(Tor_y\)) reduces by the cube of pipe OD for a given strength and pipe thickness.
It should also be noted that residual torsion can be avoided if the pipeline, flowline, or riser is installed by a well thought-out and well executed J-lay operation, in which the pipeline is never bent beyond the elastic limit. Such projects are rare, so in most cases it may be assumed that both residual plastic bending and the consequent residual torsional loads are present.

Reduction of Stress Concentrations.
Recent advances in FE analysis allow 3D bending analysis of coupled mechanical connectors with complexities like numerous contacts between mating threads. This analytical ability and actual testing results were used to identify and optimize SCFs within the connector body.

One of the locations in the connector that was significantly improved is shown in Figure 23, which compares the SCFs in bottom of the nib seal groove. This before and after view shows the box nib groove stresses due to pressurization and external bending loads. It should be noted that the connector base material has a greater fatigue life than the girth weld connections to the pipe (e.g., DNV B vs. C or E-curves). This particular SCF does not usually result in structural failure, but localized fatigue cracking can result in reduced seal life and overall sealing redundancy. Therefore, the improved radii in this sealing groove increase redundant seal reliability in cyclic bending and internal pressure.

Modular J-lay with Mechanical Connectors
Studies have shown that the overall cost of deepwater pipeline construction is mainly determined by: vessel day rates, and pipelay rates. Steel pipelines and risers are usually installed by J-lay, S-lay or reel lay vessels with day rates in the hundreds of thousands of dollars. Joining pipe segments with mechanical connectors would be much faster than offshore welding. Connected pipelay could therefore reduce construction costs by reducing the number of vessel construction days. Mechanically connected pipelay is also less complicated than welding. Operations such as line-up, multi-pass welding and weld NDT can be eliminated. Smaller construction spreads, fewer personnel and modular pipelay equipment could be used which would reduce the vessel day rate. The combination of higher productivity, lower cost equipment, and reduced crewing costs impact the main cost drivers in pipeline installation.

It can be argued that reel lay operations are not affected by offshore welding times, since the pipeline is reeled-off in lengths of about 10 km (6.2 mi). However, the reel lay vessel must transit to and from a spool base to re-load the reel. This usually takes a few days for each trip, and long stalks of pipe must still be welded together and reeled onto the vessel at the spool base. Critical path welding operations are still not eliminated as pipelines may be recovered and joined to a new spool. Also, insertions of fatigue-critical sections (TDP, TDZ, etc.), installation of strakes, fairings and SCR end terminations add days to reel lay operations because these vessels are not usually equipped to efficiently switch back and forth from reel lay to J-lay by welding. Therefore, the use of heavy duty connectors with high fatigue resistance could eliminate offshore welding in reel lay, thus cutting several days out of operations like SCR construction. Special connector designs may also be used to join stalks in reel loading operations.

Detailed design studies have been performed on modular J-lay systems which can install low-cost, more fatigue resistant pipelines and SCRs in deepwater. The resulting system is a pipe loader arm mounted on an active gimbling table over the moonpool as shown in Figure 24. The pipe handing equipment is sized for 8 to 24-in pipe, and the pipe tensioning system is a
Subsea Elevator (SSE) which clamps directly onto the pipe string. This design avoids handling the pipe tension in the loader arm, and the pipe load is carried by the SSE through the gimbal table and winches at the main deck level. The system is sized for up to 500 tons pipeline tension, and can handle triple joints 120-ft long. Added equipment modules consist of a pipe rack, pipe loading buggies and a gimbaling PLET launcher on the vessel stern.

**Modular System Operation.**

The gimbal table clamp and SSE use a “hand-over-hand” method of deploying the pipe string. The loader arm erects the triple joints, locking into the motion of the active gimbal table when approaching a near-vertical deployment angle. The triple joint is then translated downward to the gimbal table for clamping and connector make-up to the suspended pipe string. The SSE then clamps onto the lower pipe string, and the gimbal table clamp is released. Then the SSE lowers the pipe string until the clamp on the gimbal table is activated to hold the top of the newly connected triple joint. Meanwhile, the loading arm is lowered to pick up a new pipe joint, and the SSE is raised and clamped-off on the pipe string.

**Gimbal Table.**

The gimbal table is a key component which is located over the vessel moonpool. (See Figure 25.) The inner ring of the gimbal table supports the pipe string clamping system and provides a hinged support for the loader arm. The gimbal is driven by four hydraulic actuators which dynamically compensate for vessel roll and pitch motions of up to ±3 deg. The hydraulic system allows the lay angle to be set in any azimuth direction (±10 deg), so that the vessel heading can be independent of movement along the pipelay route. The heading can therefore be optimized for lowest motions.

**Loader Arm.**

The loader arm is similar to a crane boom which raises pipe joints from the deck and aligns the joints with the gimbal-supported pipe string. When the pipe loader arm is in the loading position, the pick-up clamps are aligned with the center of rotation of the gimbal table. This allows the clamps to rotate around the centerline of the pipe joint while the loader is synchronized with the vessel pitch motions. Thus, the pipe joint can be picked up from a fixed station in the same plane as the gimbaling axis.

The pipe loading sequence is detailed in Figures 26 – 28 as follows:

1. **Step 1:** The loader arm is docked in the pick-up position and fixed relative to vessel pitch motion through forward winch wires. The arm rolls with the gimbal table.
2. **Step 2:** The loader arm is partially raised and the control system drives the arm onto the arm cylinders for controlled docking.
3. **Step 3:** The arm is fully docked and perpendicular to the inner gimbal ring and winch wires are slackened. Supported on heel bearings and docking cylinders, the loader arm can now gimbaling in roll and pitch.
The loader arm is also equipped with a large A&R sheave which can be centered over the pipe string for rapid partial or full abandonment using the vessel winches.

**Subsea Elevator.**
The SSE travelling block is the main tensioning system for the deployment and retrieval of the pipe string. (See Figures 29 and 30.) It is suspended from the moonpool gimbal table by two redundant wire loops, that are reeved through multi-fall sheaves and a winch system on the main deck. This arrangement puts the tensioning loads on the gimbal table and deck winches, thereby avoiding an elevated tensioning structure above the main deck. As described above, the SSE deploys or retrieves the pipe string in 120-ft lengths by locking onto the pipe string, lifting the pipe string as the gimbal table clamp is released, and then raising or lowering the string by one joint length. The clamps on the gimbal table and SSE alternately hold the pipeline load while the SSE is moving; therefore, the description of this handling sequence is “hand over hand.” In the normal deployment mode, the SSE works between -122 and -244 ft measured from the gimbal table elevation. Depending on the outer pipeline configuration, the clamping system on the gimbal table and SSE may use friction or collar clamps.

**Pipeline Handling Equipment.**
The modular pipelay spread includes a pipe handling system and a stern gimbal table for launching PLETs and other end terminations. The handling system consists of a pipe rack capable of holding 8,200 ft of 20-in pipes in triple joint lengths. Transfer buggies move the pipe from the rack to the staging area and onto roller stands at the loader arm pick-up station. (See Figures 31 and 32.)

A second gimbal table is mounted on the stern of the vessel. This is used for connecting and launching pipeline terminations such as PLETs and flexible joints, which may be too large for handling in the moonpool. It is also used for operations such as pipeline initiation, laydown or handoff operations that may exceed the angular clearances in the moonpool. (See Figures 33 and 34.)
The table provides passive roll and pitch gimbaling up to +/-10 deg while supporting the pipe string load. It is equipped with a hydraulically powered tilt table, tilt frame and collar clamp, which are controlled locally at the stern. When in the upright position, the gimbaling clamp is locked using hydraulic cylinders, relative to the gimbal arm. This prevents motion between the loader arm and the pipe-string while both still have one degree of freedom (roll angle). When the collar clamp is locked, the pipe fitting is lowered towards the pipe string, while an adjustable clamp ensures accurate mating.

**Cycle Time and Operation.**

The cycle time of the modular J-lay system is designed to be as fast as possible by performing several operations in parallel. For example, the loader arm releases the standing pipe once it is sufficiently lowered by the SSE. The arm can then be lowered and reloaded with another triple joint, and then raised while the SSE is retrieved and clamped-off on the pipe string. As shown in Figure 35, the overall cycle time for connection and deployment of a triple joint is approximately 20 minutes. This is approximately 3x faster than welding since the connector takes only 5 minutes to make-up.
Conclusions and Summary
A concentric thread connector design has been thoroughly tested and qualified to recent International Standards for connected risers and pipelines (ISO 21329). This connection system is now ready for use on deepwater FPSO, FSO and TLP risers in locations such as the GOM, West Africa, Brazil and the North Sea. Further improvements to the connector have been made, thus increasing its reliability in deepwater applications. Summary conclusions on this development effort include the following:

- A new pipe connector has been tested and qualified to International Standards (ISO 21329) for use in offshore pipelines. This application includes pipelines on the seabed and risers connected to deepwater Floating Production Systems (FPSs), i.e., flowlines and export lines such as SCRs and Lazy Wave SCRs (LWSRs).
- Mechanically connected pipelines have met high reliability standards and can offer several advantages over traditional welded pipe joints:
  - Mechanically connected pipelines with double-sided welds and high quality requirements can meet DNV C1 or C curves, which can increase fatigue life by 10x over field-welded pipe.
  - Pipeline joints with connectors can utilize high strength steel, pipelines with exotic cladding, and other materials such as titanium or Inconel\(^1\) which are impractical for field welding. These materials can be utilized in pipelines with no impact on pipelay rates and construction productivity.
  - Connected high strength pipelines can reduce riser hang-off weight and reduce overall material costs through weight reduction, i.e., cost per unit pipe length.
- Modern FE capabilities allow the analysis of mechanical connectors and complex load cases. Further design improvements have been studied and implemented in the qualified connector. ID gaps have been eliminated, tightly sealed and preloaded. Torsional resistance to slippage and/or back-off have been increased to more than 40% of pipe SMYS. Connector SCFs have also been optimized for even greater fatigue life.
- A DP construction vessel with modular pipelay equipment can perform turnkey installation of pipelines, flowlines and risers. Modular J-lay using mechanical connectors instead of welding can increase pipelay rates and reduce overall spread costs. Connected pipelay would be less complex than offshore welding, and it can be performed on vessels of opportunity, thereby making pipelay logistics and planning more flexible.

Acknowledgements
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References

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1 Inconel is a registered trademark of Special Metals Corporation that refers to a family of austenitic nickel-chromium-based superalloys. Inconel alloys are typically used in high temperature applications. It is often referred to in English as “Inco” (or occasionally “Inconel”). Common trade names for Inconel include: Inconel 625, Chronin 625, Altemp 625, Haynes 625, Nickelvac 625 and Nicrofer 620.


**Nomenclature**

3D = 3-Dimensional  
API = American Petroleum Institute  
A&R = Abandonment and Recovery  
deg = degrees  
double = 2x standard pipe length (80ft)  
DNV = Det Norske Veritas  
DP = Dynamically Positioned  
DSAW = Double-sided Arc Weld  
Eq. = Equation  
FE = Finite Element  
FPS = *Floating* Production System (FPSO, FSO, TLP, Semi-sub, etc.)  
FPSO = *Floating* Production Storage and Offloading  
FSO = Floating Storage and Offloading  
\( ft = \) foot  
GOM = Gulf of Mexico  
Hz = Hertz (cycles per second)  
ISO = International Organization for Standardization  
\( J = \) Polar moment of inertia (Eq. 1)  
\( in = \) inch  
\( kg = \) kilogram  
\( kip = \) 1000 pounds-force  
\( kN = \) kilo Newton  
\( ksi = \) 1000psi  
\( lb = \) pound force  
\( LWSR = \) Lazy Wave Steel Riser  
\( m = \) meter  
\( min = \) minutes  
\( MPa = \) 10\(^6\) Pa  
\( N = \) Number of cycles  
\( OD = \) Outside Diameter  
\( OOL = \) Oil Offloading Line  
\( NDT = \) Nondestructive Testing  
\( Pa = \) Pascal  
\( PLET = \) Pipeline End Termination  
\( psi = \) pound-force per square inch  
\( quad = \) 4x standard pipe length (160ft)  
\( r = \) Pipe OD radius (Eq. 1)  
\( S = \) Pipe yield strength (Eq. 1)  
\( SAE = \) Society of Automotive Engineers  
\( SCF = \) Stress Concentration Factor  
\( SCR = \) Steel Catenary Riser  
\( Semi = \) Semi Submersible  
\( SMYS = \) Specified Minimum Yield Strength  
\( S-N = \) Fatigue Stress-Number of Cycles Curve  
\( SSE = \) Subsea Elevator  
\( TDP = \) Touch Down Point  
\( TDZ = \) Touch Down Zone  
\( TLP = \) Tension Leg Platform  
\( Tor_y = \) Pipe Torsional Yield Strength (Eq. 1)  
\( triple = \) 3x standard pipe length (120ft)  
\( WT = \) Wall Thickness  
\( x = \) times, multiplied
Unit Conversions.
(US Customary = SI Equivalent)

Length
1.0in = 25.4mm
1.0ft = 0.3048m

Mass
1lb = 0.4536kg

Force
1lbf = 4.448N
1kip = 4.448kN
1ton = 8.896kN

Stress
1psi = 0.069bar
1ksi = 6.895MPa

Temperature
\(^{\circ}F = \frac{\text{\circ}C + 32}{1.8}\)

Connector Configuration and Test Parameters.
Test Article Configuration.
Box and pin weldments with the following dimensional characteristics:

- \(D_i\) = (specified pipe inside diameter) = 18-in
- \(D_o\) = (specified pipe outside diameter) = 20-in
- \(E_p\) = (Young’s Modulus of pipe) = \(29 \times 10^6\) psi
- \(L_p\) = (length of test sample between supports) = 46.8-in minimum for load testing
- \(L_g\) = (grip length of pipe) = variable depending on test configuration.
- \(L_s\) = (length between scribe mark and coupling on test sample) = variable
- \(t\) = (specified wall thickness) = 1-in
- \(t_{\text{min}}\) = (minimum wall thickness accounting for manufacturing tolerances) = 0.920-in


- Maximum and minimum rated pressures and temperatures are calculated per material strengths of the 20-in Connector (SAE 4130, 90 SMYS) and test pipe segments (API 5L X65, 20-in OD, 18-in ID):
  
  - \(THT\) (maximum test temperature) = 120\(^{\circ}\)F (49\(^{\circ}\)C)
  - \(TLT\) (minimum test temperature) = 39\(^{\circ}\)F (4\(^{\circ}\)C)
  - \(PH\) (nominal upper pressure for service load tests) = 2,250 psi (155 bar)
  - \(HT\) (depth upon which external pressure is based) = 6,562 ft (2,000 m)
  - \(FCMU\) (Connector make-up force, max and min) = 850 to 550 kips (3,781 to 2,447 kN)
  - \(PCMU\) (Connector make-up pressure, max and min) = 6,000 to 3,750 psi (414 to 259 bar)
  - \(FCBO\) (Connector break-out force, max and min) = 2,500 to 3,500 psi (241 to 172 bar) (This is the clamping force applied prior to connector break-out)
  - \(PCBO\) (Connector break-out pressure, max and min) = 6,300 to 5,700 psi (448 to 393 bar) (An annular break-out pressure exceeding 6,500 psi will cause yielding of the connector material)
  - \(Fun\) (axial tension force for unrestrained operation) = 839.3 kips (3,733.0 kN)
  - \(Fre\) = (axial compression force for restrained operation) = 1,038.1 kips (4,617.4 kN)
  - \(Fat,\) = (axial force for limit load test, 80% of the pipe tensile rating, 65 ksi) = 3,104 kips (13,807 kN)
  - \(K =\) (load multiplication factor) = 1.08
  - \(K_{cc}\) = (ratio actual to specified min yield stress of critical Connector material) = 1.16
  - \(K_p\) = (ratio actual to specified min yield stress of pipe body material) = 1.08
  - \(KSCF\) = (stress concentration factor) = (A calculated value which is variable according to mean stress and connector location.)
  - \(Min\) = (bending moment for installation) = 1,010.7 ft-kips (1,370.3 kNm)
  - \(Mhy\) = (bending moment for hydrostatic pressure test) = 1,327.8 ft-kips (1,800.3 kNm)
  - \(Mun\) = (bending moment for unrestrained operation) = 1,120.7 ft-kips (1,519.5 kNm)
  - \(Mcb\) = (cyclic bending load for fatigue testing) = 17 – 28 ksi (117.2 – 193.0 MPa)
  - \(pd\) = (design pressure) = 2,250 psi (155 bar)
  - \(pex\) = (external hydrostatic pressure at HT = 6,562 ft) = 2,916psi (201.1 bar)
  - \(pL\) = (minimum internal test pressure) = 73 psi (5.0 bar)
  - \(pop\) = (operating pressure) = 2,250 psi (155 bar)
\( pr = \) (manufacturer’s rated pressure) = 2,250 psi (155 bar)
\( pt = \) (hydrostatic test pressure) = 3,375 psi (232.8 bar)
\( RPRI = \) (limit load tests - rate of increase in pressure equivalent to 15,000 psi/min rate of increase in pipe hoop stress) = 1,500 psi/min
\( RMRI = \) (limit load tests - rate of increase in bending moment equivalent to 15,000 psi/min rate of increase in bending stress) = 338 ft-kips/min
\( RARI = \) (limit load tests - rate of increase in axial tension equivalent to 15,000 psi/min rate of increase in tension stress) = 895 kips/min

R-ratio = \( \sigma_{min} / \sigma_{max} \) = min. fatigue stress/max. fatigue stress
SL = (lowest fatigue stress range) = 17 ksi (117.2 MPa)
SM = (medium fatigue stress range) = 21 ksi (144.8 MPa)
SH = (highest fatigue stress range) = 28 ksi (193.0 MPa)
\( \sigma_{ax re} = \) (restrained axial stress) = 9,592 psi (66.1 MPa)
\( \sigma_{ayc} = \) (actual yield stress of the Connector critical component) = 104.8 ksi (723 MPa)
\( \sigma_{syc} = \) (specified minimum yield stress of the Connector critical component) = 90 ksi (620 MPa)
\( \sigma_{ayp} = \) (actual yield stress of the pipe body material) = 70.3 ksi (485 MPa)
\( \sigma_{syp} = \) (specified minimum yield stress of the pipe body material) = 65 ksi (448 MPa)

\( T_{dmax} = \) (maximum design temperature) = 120°F (49°C)
\( T_{dmin} = \) (minimum design temperature) = 39°F (4°C)
\( T_{max} = \) (maximum rated temperature) = 120°F (49°C)
\( T_{min} = \) (minimum rated temperature) = 39°F (4°C)
\( T_{opmax} = \) (maximum operating temperature) = 120°F (49°C)
\( T_{opmin} = \) (minimum operating temperature) = 39°F (4°C)
TR = (torque resistance value from Eq. C.2) = 158.23 ft-kips (214.53 kN-m)