Bending Fatigue Tests on Full-scale Drill Pipe Connections Used for Oil Drilling

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The fatigue strength of drill pipe connections is a critical issue for oil drilling, especially for horizontal extended-reach wells, where the drillstring accumulates fatigue cycles while rotating at the dog-leg. Drillstring buckling is also a reason for bending fatigue, because the bent configuration continuously changes while the drillstring rotates. More than 50% of drilling failures are caused by fatigue, and recovery procedures for failures at the drilling sites can be very costly and time-consuming.1-3 Many factors can play a role in determining the fatigue strength of the drill pipe connection, such as the tool joint make-up torque, the metallurgical condition of the drill pipe material, the quality and state of the surface and the residual stresses at the thread root. Simple small-scale testing is not reliable enough to determine the fatigue strength of the drill pipe connection; full-scale drill pipe connection fatigue testing is required instead.4,6 A three-point bending frame can be used to test the drill pipe connection under rotating fatigue.4,6 The loading condition can be defined as quasi-static, i.e. the inertia of the specimen is not involved, and the bending distribution can be solved as a beam-static scheme while the specimen rotates. The full-scale drill pipe testing approach developed in the Mechanical Department of the University of Pisa is dynamic instead of static. A controlled rotating eccentric mass induces a vibration in the drill pipe connection specimen. Near the specimen resonance, the vibration amplification is so high that the cyclic bending load is above the bending fatigue limit.

Types of Connection Tested
Two different connections were tested:

- a drill collar tool joint rotary shouldered connection (RSC) (see Figure 1A); and
- an aluminum drill pipe to steel tool joint connection (ADP-STJ) (see Figure 1B).

In the RSC connection, the fatigue crack primarily nucleates at the pin last engaged thread (LET) root or at the box LET (see Figure 1A). The weakest point of the ADP-STJ connection is not at the thread roots: the aluminum body pipe experiences fretting fatigue at the edge contact of the steel tool joint thread-free portion (see Figure 1B).7 The minimum length of the RSC to be tested is the thread length, while for the other connection it is the entire length of the two assembled tool joints.

Test Rigs
Two different test rigs were designed and manufactured, because a single solution was not possible due to the connection lengths and masses being different.

Rotary Shouldered Thread Connection Test Rig
Figure 2 shows a photograph and a schematic of the RSC test rig. The specimen is gripped by two arms and the system is supported by weak springs. Two couples of counter-rotating eccentric masses are placed at the ends of the bending arms. The motion of the rotating masses is in plane with the axis of the specimen. Cyclic in-plane bending (instead of rotating bending) is applied to the specimen. The system allows for the phase angle to be shifted between the two couples of rotating masses. When the two couples of masses rotate in-phase (phase angle 0°), the system just oscillates and no deformation is induced in the specimen. When the couples of masses rotate out-phase (phase angle 180°), the bending vibration is maximised. The inertia of the rotating masses would produce a very small bending load in the specimen, assuming the specimen and the bending arms are rigid. By contrast, the induced vibration is strongly amplified near the system resonance because of the large inertia of the oscillating bending arms.

A simple and effective inertia–stiffness model can be formulated as follows:

\[ \sigma_n = \frac{2R m l_s \omega^2}{W_s} \left( \frac{1}{1 - \xi^2} \right) \]

where \( \sigma_n \) is the nominal bending stress at the outer diameter of the connection, \( m_e \) is the eccentric mass, \( R_e \) is the mass eccentricity, \( l_s \) is the bending arm length, \( W_s \) is the connection bending modulus, \( \omega \) is the working frequency imposed to the rotating masses and \( \xi = \omega \omega_n \) is the ratio between the working frequency and the first natural frequency of the system (\( \omega_n \)) that results from the connection bending stiffness (\( k_L \)) and the bending arm moment of inertia (I):

\[ \omega_n = \sqrt{\frac{k_L}{I}} \]

It is evident that the bending stress would be indefinite at the resonance, since no damping has been introduced. A zero damping model gives a reasonable representation of the dynamics of the system even for working frequencies very near to the resonance, up to \( \xi = 0.95 \). The dynamic model shown in equation 1 is not accurate enough to determine the actual imposed bending stress because some of the geometric details of the system are approximated. During each fatigue test, a strain gauge couple connected as a half-bridge configuration was used to measure the actual bending stress, obtaining double amplification and eliminating the temperature effect.

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It is worth noting that the specimen was supported by the test rig by weak springs only. The load required to impose the cyclic bending was generated by the inertia of the bending arms and the eccentric masses. The test rig did not support any fraction of the bending load applied to the specimen. This is a remarkable advantage in terms of structural design compared with a quasi-static bending test rig solution such as the three-point bending scheme, where the entire structure of the rig is heavily loaded.

The natural frequency of the system (and the working frequency during the test) was $f_n = \frac{4m_1}{2\pi^2} = 25\text{Hz}$. The high speed was also a clear advantage in terms of the time needed to perform the tests.

**Aluminum Drill Pipe to Steel Tool Joint Connection Test Rig**

Figure 3 shows pictures of the ADP-STJ test rig and the deformed shape of the specimen during the test. This solution is more suitable for a long connection. There are two (fixed) masses – one at each end of the long specimen – instead of the heavy bending arms. The excitation is generated by a single rotating mass placed at one end of the specimen. The vibration induced is rotating bending, because the vibration shape rotates synchronously with the eccentric mass, while the specimen orientation does not change. The bending moment distribution is not uniform along the specimen axis. The lengths of the specimen ends and the masses were calibrated in order to have equal bending moment (or bending nominal stress) at the critical sections of the connection, i.e. at the edge of the contact between the steel tool joint and the aluminum drill pipe, where the fatigue crack nucleates (see Figure 1B).

The test rig designed for the RSC, as described above (see Figure 2), required only one strain gauge couple because the inertia is concentrated in the bending arms and the imposed bending is (almost) uniform along the specimen. By contrast, the inertia of the ADP-STJ connection specimen is distributed, which is the reason for the non-uniform bending stress along the specimen axis. The bending stress distribution can be calculated numerically by means of finite element linear dynamic harmonic analysis. Assuming that the material damping is low, the actual bending distribution is expected to be accurately reproduced by the simulation. Three strain gauge couples were placed: one at the first body pipe, one at the tool joints and one at the second body pipe. During the tests, the readings of the three strain gauge couples were in agreement with the finite element simulation.

As with the RSC test rig, the ADP-STJ test rig supported the weight of the specimen, while the bending load was produced by the inertia of the specimen, the fixed masses at the ends and the eccentric mass. This test rig was also very fast; indeed, the first natural frequency of the specimen was 30Hz.

**Bending Stress Monitoring During the Test**

Figure 4 shows the possibilities for controlling the bending stress cyclic amplitude during the test. The test frequency is chosen in a range near the resonance condition, where the bending stress slope, as a function of the working frequency, is high. An effective way of controlling the imposed bending stress is by changing the working frequency in the sub-resonance range. Another possible way to modify the bending stress amplitude is to change the eccentric mass(es), as is evident from equation 1. The test rig dedicated to the RSC offers a further way of controlling the
bending stress: by changing the phase angle from the out-phase condition towards the in-phase condition, the resultant mass eccentricity reduces from maximum to zero. This is equivalent to changing the mass eccentricity, and can be performed while the test is running.

During each test the imposed cyclic bending stress amplitude must not be changed. So, the test rig configuration would not be modified during a test. However, when the fatigue crack nucleates and reaches a certain size, the system stiffness reduces because of the section discontinuity introduced by the crack. For this reason, the natural frequency of the system reduces as the crack propagates, and then the resonance condition moves towards the left (see Figure 4). To keep the imposed bending stress constant, even during the fatigue crack propagation, the test rig was controlled by a LABView routine that continuously read the strain gauge signals and operated on the mass phase angle for the RSC test rig, or on the working frequency for the ADP-STJ test rig.

**Test Results**

Fatigue S-N curves were produced for both of the connections (see Figure 5). The RSC was an API NC50 connection (geometry details are
reported in the API standard\(^b\), while the ADP-STJ was an ISO 147x13 connection (geometry details are reported in the ISO standard\(^b\)). The fatigue (endurance) limit (\(S_{\text{ef}}\)) was 55 MPa for the RSC at 10x10^6 number of cycles to failure, while it was 50 MPa for the ADP-STC connection at 20x10^6 number of cycles to failure. A typical fatigue fracture surface of the ADP-STJ connection is shown in Figure 6, in which the fretting nucleation, first propagation and subsequent unstable fracture surface are evident.

### Conclusions

This article reports the experimental activity undertaken in the Mechanical Department of the University of Pisa. Two test rigs were designed and manufactured to test under bending fatigue two different connections used for oil drilling. For both test rigs, the approach was to excite the specimen near its natural frequency by means of eccentric rotating masses. The test rig structures did not bear the cyclic bending loading acting on the specimens, and the test was relatively fast, since the natural frequencies could be controlled by means of specimen side lengths and masses. The proposed tests were performed at 25 and 30 Hz. The fatigue crack nucleated at the pin LET root of the steel tool joint connection, while for the other connection the failure mode was fretting fatigue that nucleated in the aluminum body pipe at the edge contact of the steel tool joint thread-free portion. The fatigue limit of the drill collar RSC NC 50 was 55 MPa at 10x10^6 cycles to failure, and 50 MPa for the ISO 147x13 ADP-STJ connection at 20x10^6 cycles to failure.

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