

Research Article

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Fatigue Damage of the Pipeline in Deep-Water S-Lay Operation



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Abstract

In deep-water S-lay operations, the short pipe joints are welded continuously and transported down along the stinger. Cyclic loadings are induced in the pipeline due to the dynamic pipeline-stinger interaction, which can cause fatigue damage when the pipeline is exposed to too many loading cycles, especially if the installation process is temporarily stopped for some reasons, such as unexpected accident, pipeline reparation and in-line structure welding. In this condition, accumulated fatigue damage may be the limitation to govern the installation operation. Rational methods are needed to be developed to calculate the fatigue damage and maximum allowable temporary stop time in the installation process. To solve this problem, this paper firstly investigates the dynamic pipe-stinger interaction and obtains the installation-induced cyclic loading history of the pipeline based on a combined finite element simulation and model test method. And then, the dynamic cyclic pipeline loading history is used to evaluate the accumulated fatigue damage and allowable short stop/standby time based on the S-N curve and Miner's rule. The results confirm that dynamic installation process leads to cyclic loadings and causes obvious fatigue damage in the pipeline, which may governs the allowable limit sea state and temporary stop/standby duration in some circumstances.

Keywords: Pipeline; Stinger; S-lay; Cyclic loadings; Fatigue

Introduction

Submarine pipelines are usually laid onto the seabed by the S-lay method schematically illustrated in Figure 1, whose name comes from the configuration of the pipeline suspended between the pipe lay vessel and the seabed. Short sections of the pipe joints are welded on the pipe laying vessel firing line to make a continuous pipeline. It then passes through the tensioner and slides over the curved stinger downward the seabed [1,2]. In the process, pipeline bends from horizontal to an almost vertical departure angle where the pipeline leaves the stinger, large dynamic cyclic loading is induced due to the S-configuration and vessel motions [3]. These dynamic cyclic loadings will cause the pipeline to accumulate fatigue damage. This is specially the case if the installation process is temporarily stopped for reasons of unexpected accident, pipeline reparation and in-line structure welding, etc. The pipeline will be held on the stinger and exposed to more dynamic loading cycles until reaches the fatigue criterion. At that time, the pipeline will have to be abandoned to the seabed.

Fatigue life estimate of the pipeline has been widely studied for the operation phase, such as VIV-induced fatigue [4,5] and operation temperature/pressure variation-induced fatigue [6,7], but the installation-induced fatigue was studied only by a few researchers. DNV-OS-F101 splits the pipeline fatigue life between installation, as-laid and normal operation conditions for respectively 10%, 10% and 80%, but the calculation method of the installation fatigue was not mentioned in the standard [8]. Morten studied the fatigue performance of pipeline installed by S-lay, but mainly focused on the welding influence of the installation process, such as influence of misalignment, weld defect size and weld residual stresses [9]. Poberezhnyi discussed the fatigue damage of the pipeline after the S-lay and J-lay method based on both the micro and macro level. The study confirms that the laying process affects the residual life of the pipeline [10]. Bruschi confirmed the SCR fatigue damage exists during the installation phase, and designed a procedure to monitor the real-time fatigue damage during the

installation period [11], but more detailed results and analysis methods are needed to be validate by future work.

The scope of this paper aims to solve two problems. The first is to study the dynamic cyclic loading history of the pipeline caused by the installation process, especially when the installation work is temporally stopped. The second is to evaluate the accumulated fatigue damage of the pipeline induced by the aforementioned

dynamic cyclic loading history. These objectives are firstly pursued by a finite element analysis and a model test to obtain the pipeline cyclic loading. The cyclic loading history was then used to calculate the fatigue damage of the pipeline based on the S-N curve and Miner's rule. The findings confirm that the installation process lead to obvious dynamic cyclic loadings in the pipeline, which will cause fatigue damage and limit the installation operation (Figure 1).

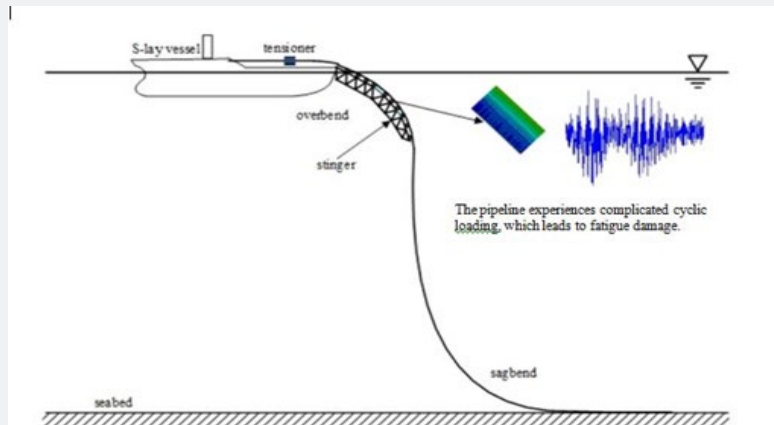


Figure 1: S-lay configuration.

Finite Element Analysis of the S-Lay Installation Process

The finite element analysis presented in this paper is based on OrcaFlex [12], and then verified by a large scale (1:20) model test. The whole laying system including the “vessel-stinger-pipeline-seabed” is built in the software. The dynamic pipeline cyclic stress history at each node is calculated and exported. The main characteristics and results of the simulation model are described herein.

Metocean data

Ocean waves are irregular by nature, which can be described by wave spectrum that indicates the distribution of the wave energy within the sea. A typical ocean wave spectrum JONSWAP spectrum is used in the simulation, which represents the real sea conditions and used to determine the motion responses of a vessel. The general characteristics of the spectrum are specified by significant wave height (H_s), peak period (T_p) and peak enhancement factor (γ). The parameters are listed in Table 1 (Figure 2).

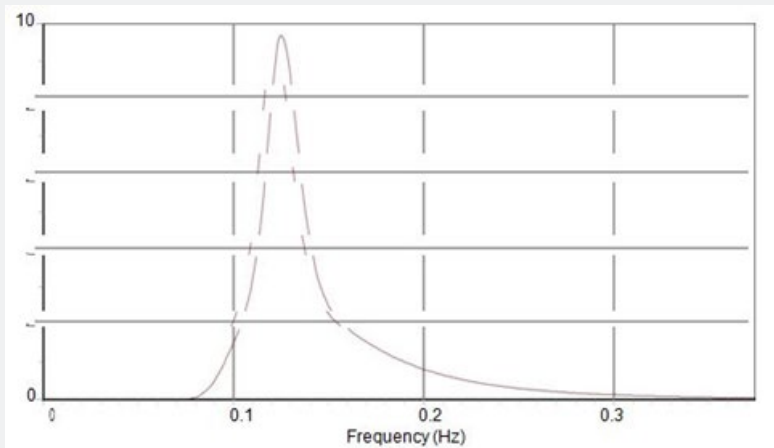


Figure 2: JONSWAP wave spectrum.

Table 1: Environment parameters.

Hs: Significant Wave Height; Tp: Peak Period; γ : Peak Enhancement Factor; Vw: Wind Speed.

Hs(M)	Tp(S)	γ	Vw(M/S)	Wave Direction	Pipe Load Condition
2.5	8	3.3	16	90°	100%

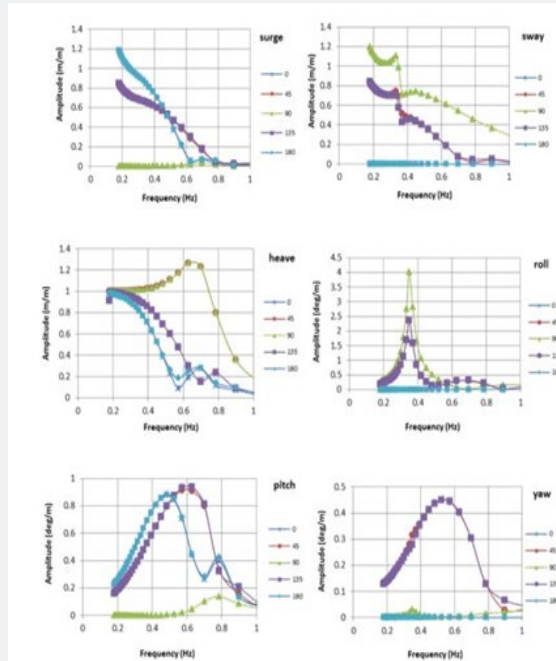


Figure 3: RAOs of pipelay vessel 'Ocean Oil 201'.

The Pipelay vessel motion

The 'OCEAN OIL 201' pipelay vessel is used in the analysis. It is a dynamically-positioned vessel measures 204.65m in length and 39.2m in breadth. The stinger is hinged in the stern of the vessel,

which is 88m long. The vessel motion can be described by the displacement response amplitude operators (RAOs). The RAOs data of the vessel consists of a pair of amplitude and phase for each degree of freedom, including surge, sway, heave, roll, pitch and yaw. The RAOs of "Ocean Oil 201" is shown in Figure 3.

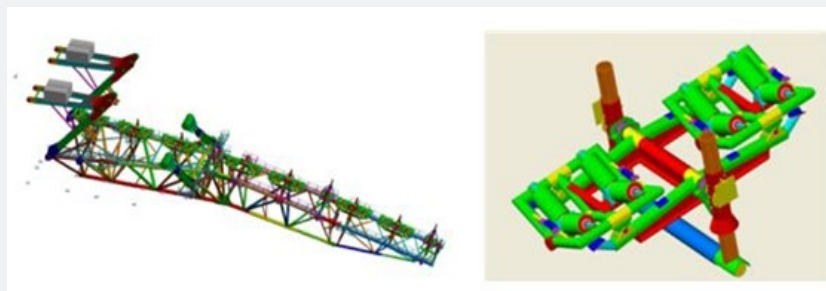


Figure 4: S-lay stinger and 'V' shape roller.

The S-lay stinger model

Stinger is hinged off the back of the pipelay vessel and used to support the pipeline. The 'OCEAN OIL 201's stinger is constructed in three separate sections and connected together by pup-pieces to adjusted the stinger radius. Ten discrete 'V' shape rollers are installed on the stinger to support the pipeline. The vertical height of the roller can be slightly adjusted to meet the local configuration requirement of the pipeline. The stinger model is shown in Figure 4. The rollers on the stinger are modeled as "V" shape structures,

which can model the contact interaction behavior between the pipeline and the stinger properly.

The pipeline model

The pipeline is modeled with a series of line segments in OrcaFlex. Each line segment is divided into two halves, and mass, weight, buoyancy, and drag properties are concentrated to the nodes at the end of each line segment. External forces and moments are also applied to nodes. Due to the large plastic

deformation of the pipeline during the deep-water operation, the non-linear pipeline material properties should be considered. The plasticity theory with isotropic strain hardening is adopted to describe the plastic behavior of the material, and the Ramberg-Osgood constitutive model is used by multi-linear approximations of the stress-strain relationship, as equation (1) illustrates; the parameters are listed in Table 2. The stress-strain relationship

remains linear elastic when the stress is below 300MPa or so. On the other hand, for stress levels higher than this value, the hardening behavior of the material will cause obvious nonlinear characters, as Figure 5 shows. Where E is the Young's Modulus; ϵ is the strain; σ_y is the stress; A is the yield offset; B is the hardening exponent for the plastic term; σ_y is the yield stress.

Table 2: Pipeline geometry and material parameters.

Water Depth (m)	Diameter (mm)	Wall Thickness (mm)	Material Grade	Young's Module (GPa)	A	B	Yield Stress (MPa)
2000	323.9	38.1	X65	207	1.29	25.58	448

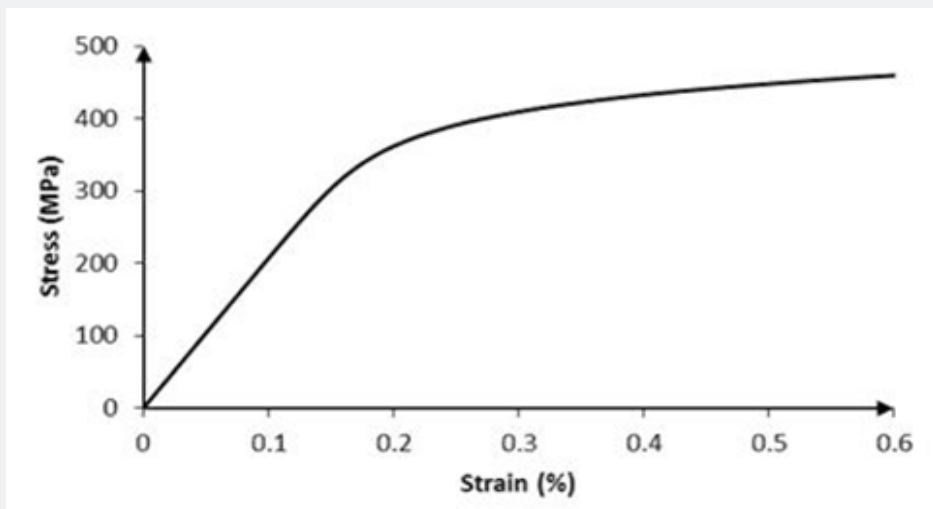


Figure 5: Nonlinear stress-strain relationship.

$$\sigma = E \epsilon + \sigma_y \left[\frac{E \epsilon - \sigma_y}{\sigma_y} \right]^B \quad (1)$$

Validation of the Finite Element Model by Model Test

To validate the accuracy of the finite element simulation model, a model test was carried out. Traditionally, offshore structure model tests would be carried out preferably in laboratory basins which can accommodate the full depth. However, in deep-water offshore model tests, the modeling of full-depth offshore floating system becomes difficult because no basin is large enough to perform such a test within reasonable limits of model scale. For example, if the structure model scale is set as 20 to obtain the available data, a 100m deep water basin is needed to model the whole length of the pipeline and pipelay system, which is impossible to find an available basin to execute the model test around the world. To solve this problem, an innovative hybrid substructure model test was carried out in the present study. Firstly, the pipelay vessel motion was measured from a traditional small scale (L 49) hydrodynamic basin model test. And then, the tested vessel motion is transferred into a controllable shaking table with stinger and pipeline installed on, which is used to simulate and reproduce the installation process of the S-lay operation. The roller force sensors are placed on each roller box to measure the real-time pipe-roller

reaction force, which would be used as the index to validate the calculated pipe-roller reaction force [3].

The small scale (149) hydrodynamic model test to obtain the vessel motion

The hydrodynamic model test (scale 1:49) was carried out to obtain the practical vessel motion corresponding to the environment conditions by Shanghai Jiao Tong University [13,14], another participant of this large JIP, see Figure 6. Wave and wind are both modeled in the basin model test. The vessel motion data was measured and recorded for every freedom. Although both the heave and pitch motion affect the pipelay operation significantly, the present paper mainly chooses the pipelay heave motion as the input vessel motion to drive the shaking table. The reason to use a single vessel heave motion is to begin this research with a simple external loading condition to derive clear conclusions and validate the OrcaFlex simulation model. Besides, the largest heave and pitch motions do not occur together: heave motion dominates in beam sea condition and pitch in head sea conditions. It is therefore reasonable to use a single large heave to represent a severe pipelay condition. The used pipelay heave motion time history is plotted as follows, which has a maximum 1.5m heave amplitudes and random periods, see Fig.6. It is reasonable to represent a relatively severe pipelay operation condition.

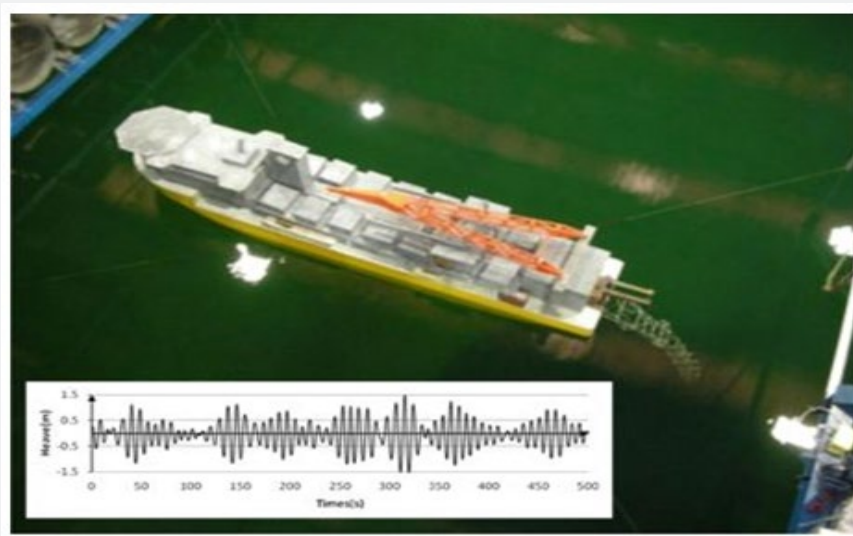


Figure 6: The small scale (1:49) hydrodynamic model test to obtain the vessel motion (scaled to the prototype).

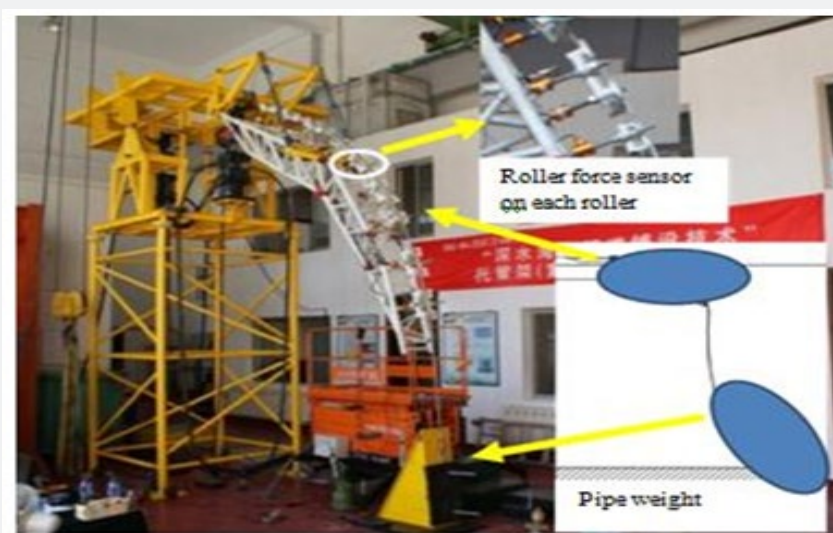


Figure 7: The large scale (1:20) dynamic pipe-stinger model test setup.

The large scale (1:20) dynamic pipe-stinger model test to measure the roller force

Because the size of the hydrodynamic model is too small, it is impossible to install force sensors on it to measure the roller reaction force directly. So the large scale stinger and pipeline model were constructed and tested to obtain the real-time roller reaction force. The stinger model was built following the geometry of the prototype of “Ocean Oil 201” with three separate sections. Each section is constructed as a space truss and connected to other sections, and has ‘V’ shape rollers at different locations. The rollers are constructed to model the same pipe-stinger contact behavior as the prototype. The force sensors are installed on the rollers to measure the roller reaction force directly, see Figure 7. The pipeline model includes two parts: an over bend section and a truncated sag bend section. A 5 m long stainless steel pipe is

used to model the over bend section. The length is long enough to cover the whole stinger surface. The diameter and wall thickness follows the geometric scaling. The pipeline was laid upon the stinger, with the top end connected to a shaking table and the other end to a lumped mass, which is used to model the weight of the submerged sag bend pipe. An inclined surface is constructed to maintain the pipeline departure angle. The whole model test includes two load steps. Firstly, the steel weight is loaded to apply the submerged pipe weight. The pipeline contacts with the stinger and deforms to the over bend configuration complying with the stinger’s geometry. The static roller forces are measured in this step. Secondly, after the shaking table has been turned on, the vessel motion time history is input to drive the shaking table. The pipe-stinger system is driven by the dynamic vessel motion. The real-time dynamic roller forces are measured.

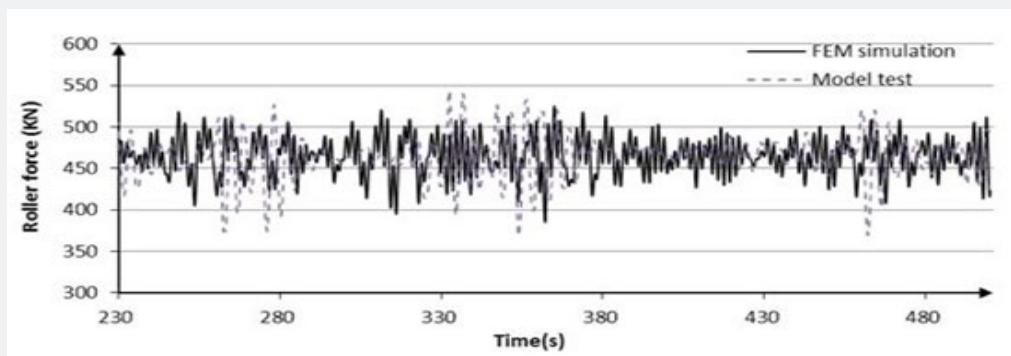


Figure 9: Dynamic pipe-roller reaction force history at the sixth roller support location.

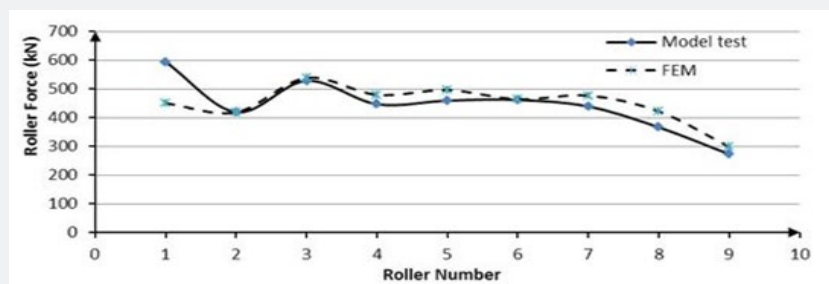


Figure 8: Static roller force distribution of each roller.

Validation of the finite element model based on the tested results

The measured roller force is used as an index to validate the simulation model [3]. Figure 8 shows the static roller force distribution obtained respectively from the model test and the finite element simulation. The roller force distribution is non-uniform on different rollers, which is common in the real installation operations. This is because the roller force is very sensitive to roller height and the geometry of the stinger. The curvature of the stinger could not be an ideal arc and induce very uniform roller force distributions. On the other hand, the model test forces are some smaller than the finite element results. This is thought to be because of the unavoidable construction errors between model and finite element model.

The roller force history of each roller can be measured in the model test when the shaking table heaves along with the stinger and pipeline. The vessel motion history is input to the shaking table to drive the shaking table. The dynamic roller force variation at the sixth roller support location is exported and shown here, as Figure 9 shows. The average roller force is about 464kN, and the force varies due to the pipelay vessel motion. It is observed that FEM simulation results and model test results have similar variation trend corresponding to the vessel heave motion in Figure

5, approximate dynamic amplifications. The same results are observed when roller forces at other locations are compared. The dynamic amplification factor is around 20%-30% compared to the static roller force. Comparing the dynamic roller force variation with the input vessel heave motion, it is clear that the roller force varies in a higher frequency. This is thought to be caused by the additional high frequency impact between the roller and the pipe, and the pipe repeatedly sliding over the roller during the model test.

Cyclic Loading History of the Pipeline During the S-Lay Installation Phase

In the pipe laying process, the pipeline departure angle fluctuates repeatedly when the vessel moves due to the environment loadings. The pipeline experiences cyclic bending moment and stress, especially at the stinger tip locations, as Figure 10 shows. If the installation work stopped for a period of time, the pipeline will be held on the stinger and exposed to many loading cycles, the fatigue damage accumulates in the process. The pipeline stress histories are extracted from the simulation model, Figure 11 is an example. It illustrates the pipeline loading history at the eighth roller support location (near stinger tip). It is clearly that stress varies during the installation phase. The stress range is around 60 to 90MPa.

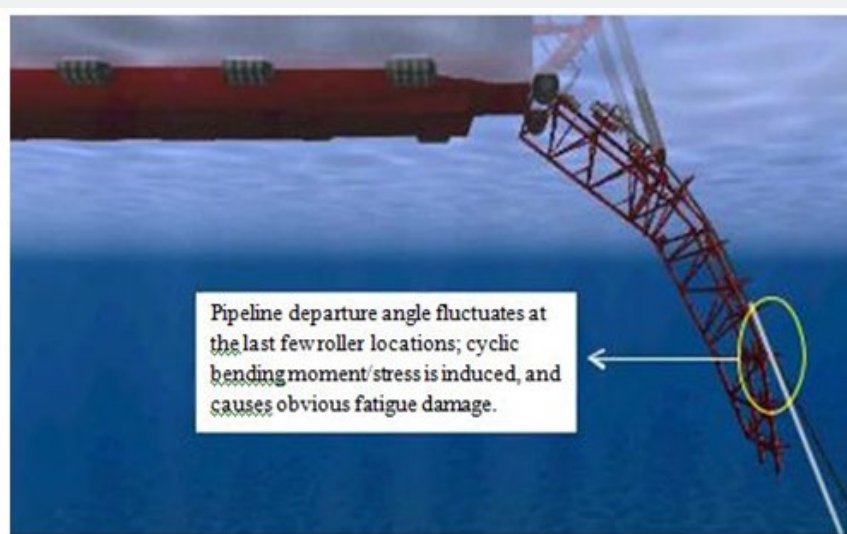


Figure 10: The fatigue mechanism of the pipeline at the last few roller supports.

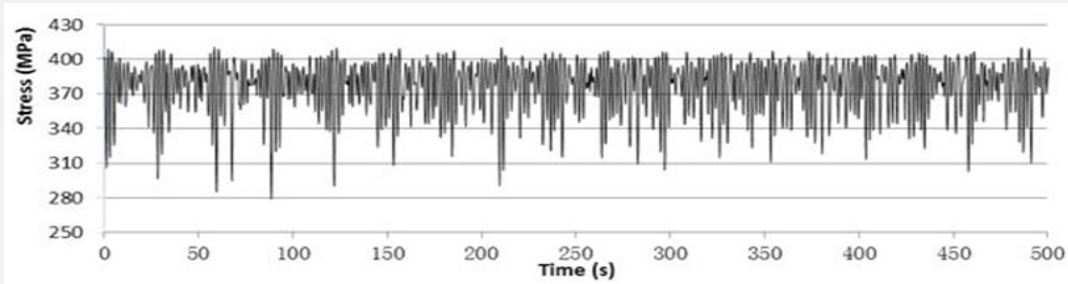


Figure 11: The stress history of the pipeline at the eighth roller support location.

Fatigue Damage of the Pipeline Caused by the Cyclic Loading

Confronting large dynamic cyclic loading history, fatigue damage accumulates under the fluctuating stress. The calculation of the fatigue damage is based on the S-N curve and Miner law's rule.

S-N curve

S-N curve defines the number of cycles to failure, which is widely used to calculate the fatigue damage of the pipeline in offshore industry. The basic description of S-N curve can be given as equation (2). In which, N , is predicted number of cycles to failure for stress range; s is the stress range of the pipeline's stress history; m is negative inverse slope of S-N curve; $\log a$, is intercept of $\log N$ -axis by S-N curve. The S-N data are often derived by fatigue testing of small specimens in test laboratories. For offshore pipeline engineering, the S-N curve in DNV-RP-C203 is often used [15]. In the present study, a bi-linear curve for the pipeline in seawater with cathodic protection is used here, as Figure 12 shows.

$$\log N = \log a + m \log s \quad (2)$$

Miner's rule and accumulated fatigue

The Miner's rule is one of the most widely used cumulative damage models for fatigue failure, which is used in the general case where stress fluctuations occur with varying amplitude. According to Miner's rule, failure occurs when the damage is accumulated to the Criteria. The Miner's rule is shown in equation (3). Where, k is the number of stress blocks; n_i is the number of stress cycles in stress block i ; N_i is the number of cycles to failure at constant stress range of magnitude; DFF is the design fatigue Factor, for installation phase, DFF 3 .

$$\frac{n_1}{N} + \frac{n_2}{N} + \dots + \frac{n_i}{N} + \dots + \frac{n_k}{N} = \frac{1}{DFF} \quad (3)$$

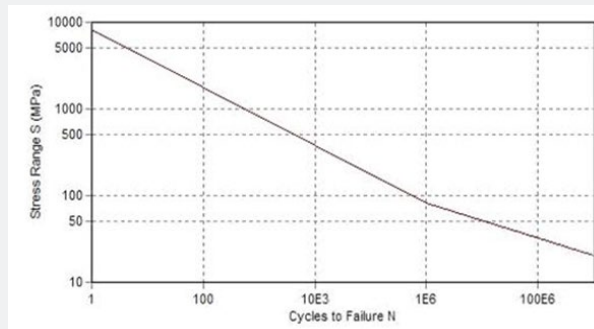


Figure 12: The S-N curve.

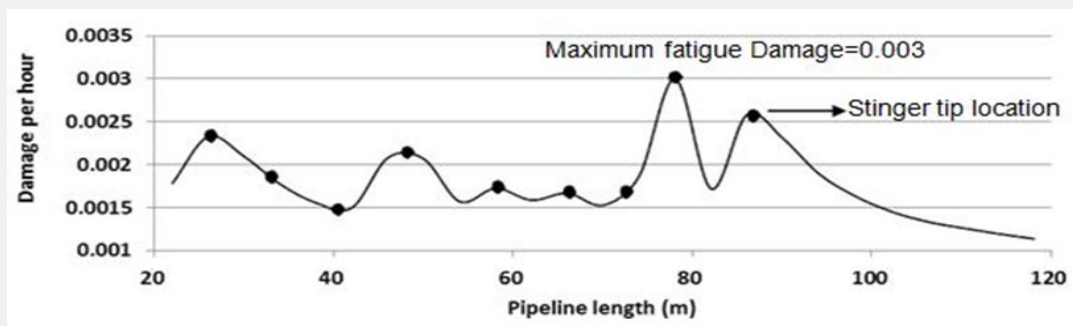


Figure 13: The accumulated fatigue damage of the overbend pipeline.

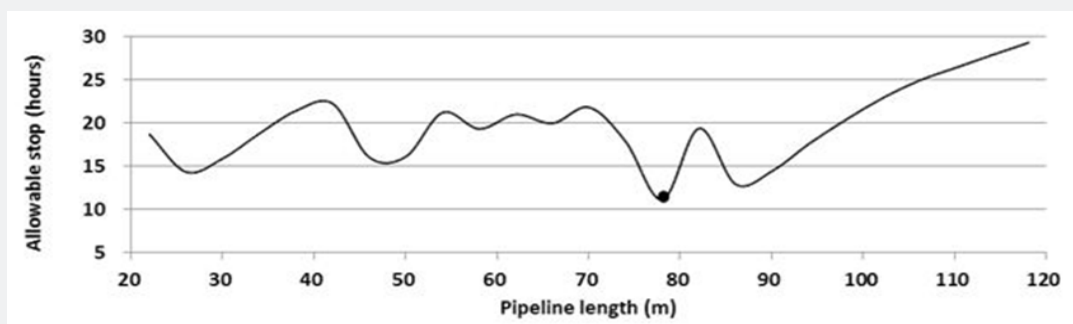


Figure 14: Allowable maximum pipeline temporary stop time during installation phase.

Criteria and allowable temporary stop duration

The stress history of the pipeline was processed by rain flow method to extract the stress range and count the cycle numbers. The stress range and cycles were then used to calculate the accumulated fatigue damage of the pipeline at each node location by Miner's rule. The results were shown in Figure 13. The roller locations are marked with dark spots in the figure. It is clear the largest fatigue damage happens at the last few roller locations. The largest fatigue damage for one hour at this sea state is 0.003.

The DNV-OS-F101 states that the fatigue damage during pipeline installation phase should be under 10% of the total life. The 10% of the total fatigue life of the pipeline represents the longest allowable time the pipeline can stay on the stinger during the operation process. If the temporary stop operation lasts longer than this time, abandon operation needs to be carried out. According to this criterion, the allowable pipeline stop time can

be calculated. The results are shown in Figure 14. The allowable stop time is controlled by the minimal value 11.1 hours. So, during the installation phase, if the pipeline lies on the stinger more than 11.1 hours, abandon operation need to be carried out to avoid fatigue damage.

Conclusion

In the present paper, two main topics are investigated. The first one aims to obtain the dynamic cyclic loading history of the pipeline when the installation process has a temporary short stop. In this part, a finite element model of "vessel-stinger-pipeline-seabed" has been constructed and then verified by the model test. The second topic is to evaluate the pipeline fatigue damage and the allowable stop time. The findings confirm that the installation process leads to cyclic loadings and causes obvious fatigue damage in the pipeline, which governs the temporary stop/standby time.

Highlights

- a. We built a finite element model including “vessel-stinger-pipeline-seabed” system to simulate the dynamic S-lay operation, and verified by a large scale model test.
- b. We obtained the dynamic cyclic loading history of the pipeline caused by the installation process, especially when the installation work was temporally stopped.
- c. We evaluated the fatigue damage of the pipeline and the allowable short stop time of the installation work.

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