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Study on the design of a model experiment for deep-sea S-laying

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ABSTRACT

As a primary device of the S-lay technique, the stinger is used to protect the overbend section in a pipeline. It is difficult to determine the design loads acting on the rollers during the dynamic pipelaying condition using numerical methods alone. Instead, a test model is used to analyze this complex problem. This paper introduces some similarity criteria based on the dynamic similarity of a stinger, a pipeline, and a vessel. Then, the stinger of the first pipelaying crane vessel in China, “HYSY201”, is modeled with these similarity criteria. The loads for the stinger's structural design and movement mechanism in the off-design conditions are simulated, producing some useful conclusions and suggesting that local reinforcement of this stinger be performed. The vessel recently began its first deep-sea pipelaying project in Liwan in the South China Sea. Thus, the model method proposed in this paper is effective and could be used to design and test other articulated stinger structures rather than relying on engineering heuristics.

1. Introduction

The S-lay technique is an economic and an efficient pipelaying method that can be used in both shallow (water depth < 500 m) and deep water (water depth between 500 m and 1500 m) depending on the stinger. In the S-lay method, the pipeline being laid can be subdivided into two main sections: the overbend, which is the pipeline section over the lay ramp and the stinger from the tensioner to the stinger tip, and the sagbend, which is the pipeline section in between the stinger tip and the touch-down point, as shown in Fig. 1. The stinger is a key piece of equipment used to protect the pipeline in the overbend section (Veritas, 2012). The geometric parameters of the stinger, such as its length, curvature, and roller spacing, are the core technology of stinger design (Wang et al., 2009; Zhang et al., 2011a, 2011b). Currently, many successful studies have been reported including the configuration of the pipeline during static laying (Palmer et al., 1974; Palmer and King, 2008, Zhu and Cheung, 1997, Guarracino and Mallardo, 1999, Gong et al., 2011a,b), analysis of the contact between the pipeline and stinger (Dai et al., 1999, Gong et al., 2011a,b, Feng et al., 2012), and coupled dynamic analyses of the stinger and the pipeline (Lee et al., 2001, Eduardo et al., 2009, Jensen and Fossen, 2009; Suschitz and Taylor/McDermott, 2011). However, analyzing and predicting the contact forces based on numerical methods is still difficult because of the material's nonlinearity, the geometric nonlinearity of the pipeline, the moving boundary of the vessel, and the rigid body motion in the stinger structure during pipelaying. Few research studies have been published that considered all of these difficulties. Thus, a model would be a useful tool to simulate the complicated cases (Lü et al. 2007, Roitman et al., 1992, Gibbings, 2011).

Unfortunately, studies on S-lays by test model have rarely been reported, which may be the result of laboratory space limitations in large scale considerations of the pipeline in ultra-deep-sea (water depth > 1500 m) pipelaying. Meanwhile, the overbend section and the stinger structure model will be small and difficult to analyze at a small scale. Thus, it is a challenging but meaningful task to devise an overbend section experiment for stinger design in which the force acting on the roller and the strain distribution along the overbend section are both considered. In this paper, we discuss the stinger and the pipe in an overbend model test of ultra-deep-sea pipelaying on a large scale of 1:20.

2. Model and similarity

2.1. Sagbend section

In deep- and ultra-deep-sea pipelaying, the flexibility of the sagbend pipeline will be close to a catenary (Wang et al., 2009) and several orders of magnitude larger than that of the stinger. Nevertheless, some assumptions should be stated for the dynamic case. First, compared to the water depth, the range of motions of the pipelaying vessel could be negligible, and thus, the departure angle at the tail section of the stinger is fixed. The inflection point moves in such a small area that it can be neglected, and the added mass produced by fluid can also be neglected. Moreover, the length of pipeline in the water is unaltered. Based on these assumptions, the

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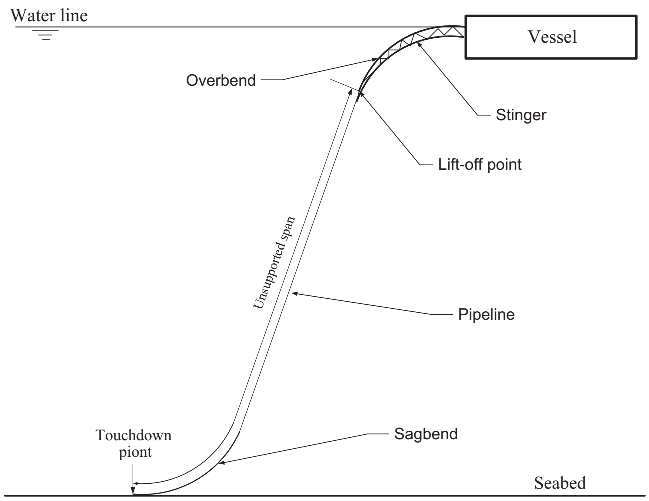

Fig. 1. S-lay configuration.

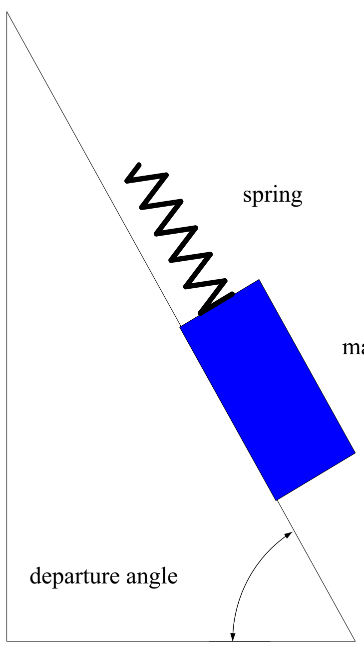

Fig. 2. Pipeline model in the sagbend section.

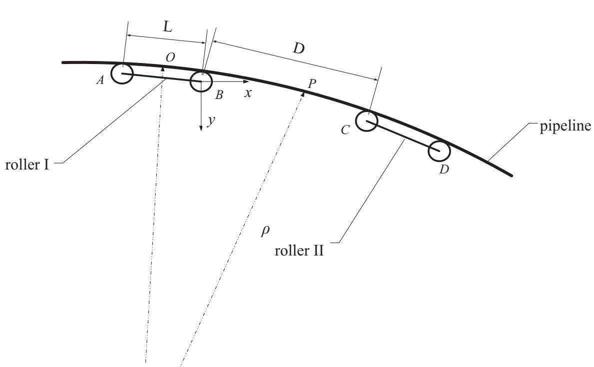

Fig. 3. Rollers and pipeline in the overbend.

pipeline in the sagbend section can be simplified as a spring-mass system residing in an angular surface, as shown in Fig. 2. And the tensile stiffness of the spring and the mass can be determined by

$$K_{SB} = EA/L, \; m = \rho_L L \tag{1}$$

where K_{SB} is the tensile rigidity of the sagbend, E is the elastic modulus of the pipe material, A and ρ_L are the cross-sectional area and linear density of the pipeline in water, respectively, and L is the length of the sagbend section.

2.2. Overbend section

In the overbend section, the pipeline is designed based on the displacement-controlled condition (Veritas, 2012), which can be expressed as

$$\left(\frac{\varepsilon_{Sd}}{(\varepsilon_c(t_2,0)/\gamma_\varepsilon)}\right)^{0.8} + \frac{P_e - P_{\min}}{(P_c(t_2)/\gamma_m \gamma_{SC})} \le 1 \tag{2}$$

where ε_{Sd} is the design compressive strain, $\varepsilon_c(t_2, 0)$ is the characteristic collapse strain based on pipe thickness t_2 and pressure 0, P_e is the external pressure, P_{min} is the minimum internal pressure that can be sustained, $P_c(t_2)$ is the characteristic collapse pressure based on pipe thickness $t2$, and γ_ε is the resistance strain factors, γ_m is the material resistance factor, and γ_{SC} is the safety class resistance factors. All of the parameters in this equation are shown in the standard.

While the stinger is simplified as some points located in the same rigid arc, the pipeline in the overbend is shown in Fig. 3, and the forces acting on the arc are shown in Fig. 4.

The maximum curvature value at B can be determined by (Yun et al., 2004)

$$\kappa_B = \frac{\alpha(\theta_L + \theta_R)}{\tanh \alpha x_L + \tanh \alpha x_R} \tag{3}$$

where $\alpha = \sqrt{T/EI}$, T is the tension applied at the overbend, EI is the bending stiffness of the pipeline, and θ_L and θ_R are the included angles of half of the length and spacing of a roller, respectively. By geometric transformation, an intuitive solution is easily obtained:

$$\kappa_B = \frac{\alpha(L+D)}{2\rho(\tanh \alpha L/2 + \tanh \alpha D/2)} \tag{4}$$

where L and D are the length and spacing of the roller boxes, respectively, and ρ is the radius of the stinger. The strain at B is

$$\varepsilon_B = r\kappa_B = \frac{\alpha(L+D)}{2(\tanh \alpha L/2 + \tanh \alpha D/2)} \frac{r}{\rho} \tag{5}$$

where r is the radius of the pipeline.

Meanwhile, the load acting on the roller is shown as

$$R_f = 2(R_L + R_R) = \delta T \tag{6}$$

where R_f is the roller load, R_L and R_R are pressures given by pipe on both sides of the roller, as shown in Fig. 4. $\delta = (L+D)/\rho$ is a dimensionless parameter.

Clearly, according to Eq. (3)–(6), ε_B is a function of the geometry of the stinger (L, D, ρ), the modulus of elasticity of the material (E), the cross sectional properties (I) of the pipeline, and the tension force (T). Furthermore, the roller load is only related to the tension force of a given stringer. Given Eq. (2), the geometric

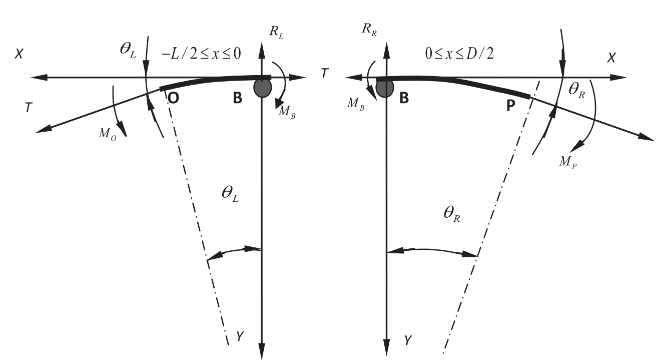

Fig. 4. Pipeline diagram in the overbend section.