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Coupled Dynamic Model of Riser-Tensioner System of Cylindrical Type FDPSO

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Keywords: Riser; tensioner; axial vibration; stiffness coefficient; damping coefficient

Abstract: In order to improve the safety of production riser of cylindrical type FDPSO, a riser-tensioner (RT) system is invented. A coupled dynamic model that considering the interaction between riser and tensioner for RT system is developed with micro-element method. The corresponding boundary conditions and difference scheme of the model are derived. ABAQUS software is used to verify the model. With the model, the dynamic behavior of riser system on a float platform in South China Sea is investigated, based on which the value range of riser length, tensioner stiffness coefficient and tensioner damping coefficient are proposed. The results of study can provide effective theoretical support for optimal design of riser and developing new type of tensioner.

1. Introduction

In the South China Sea, the existence of rich oil and gas resources has been confirmed and documented in detail [1-5]. Recently, the Shanghai Waigaoqiao Shippingbuilding Co., Ltd. and related units have been designing cylindrical floating, drilling, production, storage and offloading (FDPSO) vessel to realise highly efficient drilling and oil recovery operations. The development concept is shown in Fig. 1. Among them, riser vibration is the key to be solved is the key problem to be solved. In order to reduce the effect of tumbling motion and optimise the service life of the riser, three tensioners are installed at the bottom of the riser to reduce the axial vibration load. Therefore, it is very important to develop an effective dynamic model to investigate the dynamic behaviors of the production riser-tensioner (RT) system to provide markers to practice.

Commercial software, such as ABAQUS, ORCAFLEX and DEEPRISER are most frequently used to simulate the dynamic behavior of riser [6-8]. It is a good choice for professionals to use mature software based on finite element method (FEM) to make a qualitative analysis on the dynamic problem. However, it needs special computational mechanical knowledge to master such software, or else it is difficult to make right judgment to the calculation results. There are also some researchers trying to develop dynamic models of the problem based on the principle of vibration mechanics [9-10]. But in these models, neither the long-thin riser is regarded as infinitely long elastic pipe, nor is the deformation of the riser ignored with the riser considered as a rigid boundary of tensioner. According to these works, the mathematical models of the long-thin riser system may be simplified excessively and are great different with the practice structure.

Presently, the models which can be used to effectively study the dynamic behaviors of the riser system under perforating impact load are still very lacking. In view of this, the first purpose of this paper is to develop a coupled dynamics model of RT system in which the riser is viewed as an elastic pipe with finite length. The second purpose is to use the model to study the influence of riser length, tensioner stiffness coefficient, tensioner damping coefficient and quality of tensioner on the dynamic behaviors of the riser system.

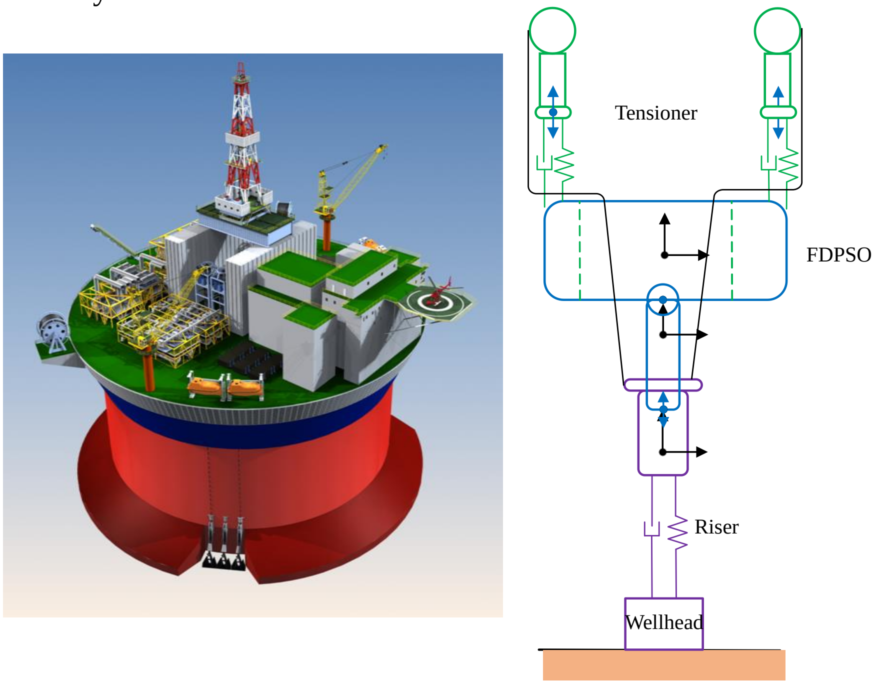


Figure 1: Principle of Cylindrical Type FDPSO with riser-tensioner system

2. Coupled dynamic model

2.1. Differential equation

According to the structure and force analysis, the coupled dynamic model of a RT system is shown in figure 2. The upper long-thin elastic pipe represents the riser, the lower quality-spring-damper system represents the three tensioners. The top end of riser has a heave motion together with a floating platform. The influence of flexible pipe and SPT is ignored.

In order to derive the coupled dynamic model, two coordinate axes, whose origins are respectively at the down bottom and quality point, are set up. Micro-element method and D'alembert principle are respectively used to establish the axial vibration equations of the RT system.

Using the micro-element method, an arbitrary micro section of the long-thin elastic pipe is cut off hypothetically and force analysis are made. In figure2, the following four physical quantities respectively represent the inertial force, the axial force, viscous damping force and gravity on the infinitesimal section.

According to the force equilibrium conditions of the infinitesimal section, the vibration partial differential equation of the riser is obtained.

$$\rho A\frac{\partial^2 u_1(x,t)}{\partial t^2}dx - EA\frac{\partial^2 u_1(x,t)}{\partial x^2}dx + \mu\frac{\partial u_1(x,t)}{\partial t}dx = \rho Agdx \tag{1}$$

where E, ρ and A respectively represent the elastic modulus, material density and cross sectional area of the riser. v is the damping coefficient of the internal and external liquid of the riser. g is the gravity acceleration. u1(x,t) is the displacement of cross section whose position is x from the coordinate origin of riser at time t.

With both sides divided by ρAdx, the above equation can be transformed into the following form.

$$\frac{\partial^2 u_1(x,t)}{\partial t^2}-a^2\frac{\partial^2 u_1(x,t)}{\partial x^2}+v\frac{\partial u_1(x,t)}{\partial t}=g_0 \tag{2}$$

where a=(E/ρ) ½ is the wave propagation speed in riser, g0 is a constant that represents the gravity of pipe section with unite length, ν=μ/ρA is the damping coefficient per length of riser.

The vibration differential equation of each tensioner can be written as follow according to D 'alembert principle:

$$m\frac{d^2 u_2(t)}{dt^2}+c\frac{du_2(t)}{dt}+ku_2(t)=0 \tag{3}$$

where m, k and c respectively represent the quality, stiffness and damping coefficient of each tensioner, u2(x,t) is the displacement of each tensioner at time t.

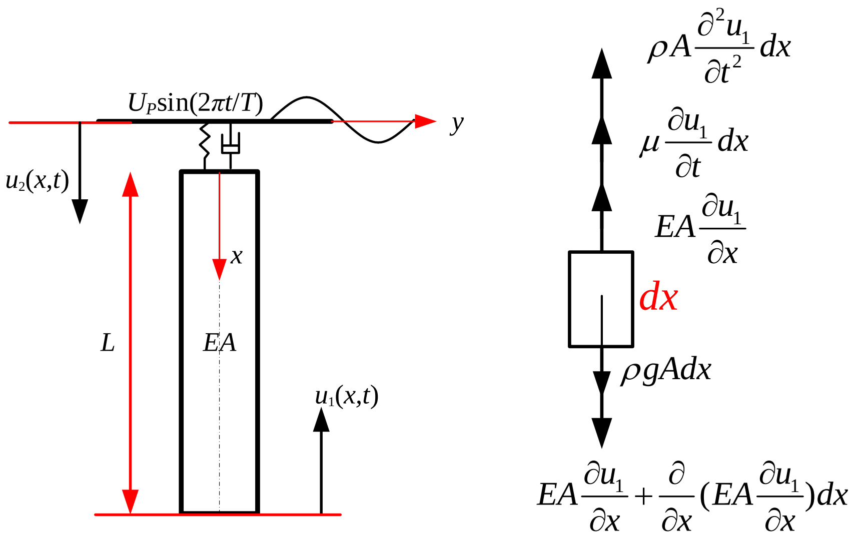


Figure 2: Axial vibration model of riser-tensioner system

2.2. Boundary condition

It is noted that the location of the dangling end, namely the down bottom of the riser changes with the deformation of the elastic string. Therefore the equation (3) can be adjusted as the following form in order to consider the location change of the dangling end.

$$m\frac{d^2 u_2(t)}{dt^2}+c\frac{d}{dt}\left[u_2(t)+u_1(0,t)\right]+k\left[u_2(t)+u_1(0,t)\right]=0 \tag{4}$$

The equation (4) can be also expressed by the following formula:

$$F_s(t)-F_d(t)=m\frac{d^2 u_2}{dt^2} \tag{5}$$

where $F_s(t)$ and $F_d(t)$ respectively represent the spring force and the damping force and

$F_s(t) = k\left[u_2(t) + u_1(0,t)\right]$, $F_d(t) = c\dfrac{d}{dt}\left[u_2(t) + u_1(0,t)\right]$

According to the force continuity condition on the joint between the riser and the tensioner.

$$EA\frac{\partial u_1(x,t)}{\partial x}\Big|_{x=L} = F_s(t) + F_d(t) \tag{6}$$

Formula (6) can be further written as

$$EA\frac{\partial u_1(x,t)}{\partial x}\Bigg|_{x=0} = c\frac{d}{dt}\left[u_2(t) + u_1(0,t)\right] + k\left[u_2(t) + u_1(0,t)\right] \tag{7}$$

Since the top end of riser has a heave motion together with float platform, the following displacement boundary can be given

$$u_1(1,t) = U_p \sin(\frac{2\pi}{T_s}t) \tag{8}$$

It can be found from the above analysis that equations (2), (4), (7), (8) describe together the dynamic model of the riser system under impact load namely.

$$\begin{cases} \dfrac{\partial^2 u_1(x,t)}{\partial t^2} - a^2\dfrac{\partial^2 u_1(x,t)}{\partial x^2} + v\dfrac{\partial u_1(x,t)}{\partial t} = g_0 & (9a) \\ m\dfrac{d^2 u_2(t)}{dt^2} + c\dfrac{d}{dt}\left[u_2(t) + u_1(0,t)\right] + k\left[u_2(t) + u_1(0,t)\right] = P(t) & (9b) \\ u_1(0,t) = U_p \sin(\dfrac{2\pi}{T_s}t) & (9c) \\ EA\dfrac{\partial u_1(x,t)}{\partial x}\Bigg|_{x=L} = 3c\dfrac{d}{dt}\left[u_2(t) + u_1(0,t)\right] + 3k\left[u_2(t) + u_1(0,t)\right] & (9d) \end{cases} \tag{9}$$

2.3. Difference scheme

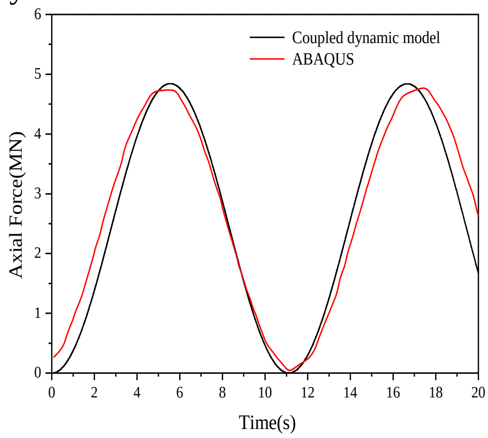
Newton difference technique is used to solve the equations (9). The riser is discretized uniformly with (N+1) nodes along its axis direction whose numbers are i=0,1, ⋯;N from top to bottom. The time of the vibration response is divided into K+1 spans whose sizes are △t and numbers are j=0,1, ⋯;K. Therefore, equations (9) can be expressed as equations (10) in difference scheme.

$$
\begin{cases}
u_{i,j+1} = \dfrac{\dfrac{a^2\Delta t^2}{\Delta x^2}\left(u_{i+1,j}-2u_{i,j}+u_{i-1,j}\right)+\left(2-v\Delta t\right)u_{i,j}-u_{i,j-1}+g\Delta t^2}{1+v\Delta t} \quad 1\le i\le N-1, 1\le j\le K & (10a) \\
\left(\dfrac{m}{\Delta t^2}+\dfrac{3c}{2\Delta t}\right)u_{N+1,j+1}+\dfrac{3c}{\Delta t}u_{N,j+1} & \\
=-\left(3k-\dfrac{2m}{\Delta t^2}\right)u_{N+1,j}-\left(\dfrac{m}{\Delta t^2}-\dfrac{3c}{2\Delta t}\right)u_{N+1,j-1}+\left(\dfrac{3c}{\Delta t}-3k\right)u_{N,j} \quad 1\le j\le K & (10b) \\
u_1(0,t)=U_p\sin\left(\dfrac{2\pi}{T_s}t\right) & (10c) \\
\left(-\dfrac{2\Delta xk}{EA}-\dfrac{\Delta xc}{EA\Delta t}\right)u_{N+1,j+1}+\left(1-\dfrac{2\Delta xk}{EA}-\dfrac{2\Delta xc}{EA\Delta t}\right)u_{N,j+1} & \\
=\dfrac{4}{3}u_{N-1,j+1}-\dfrac{1}{3}u_{N-2,j+1}-\dfrac{\Delta xc}{EA\Delta t}u_{N+1,j-1}-\dfrac{2\Delta xc}{EA\Delta t}u_{N,j} \quad 1\le j\le K & (10d)
\end{cases} \tag{10}
$$

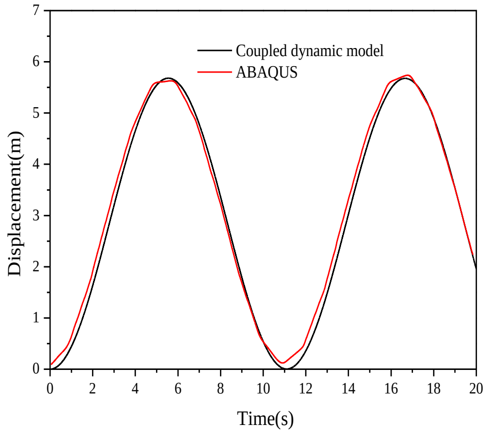
3. Case study and discussion

3.1. Case study

Taken an actual deep-water drilling operation for example, the Water Depth (WD) is 1600 m, the riser outer diameter is 346.2 mm, the riser wall thickness is 16.3 mm, the steel density is 7850 kg/m3, the elastic modulus of steel is 206GPa, the sea water density is 1025 kg/m3, the linear damping coefficient is 0.1, weight tensioner is 20T, the amplitude and period of heave action are 0.8m and 3 s respectively.



(a) Axial force on top end of riser



(b) Displacement of endpoint of riser

Figure 3: Results comparison between ABAQUS and coupled dynamic model

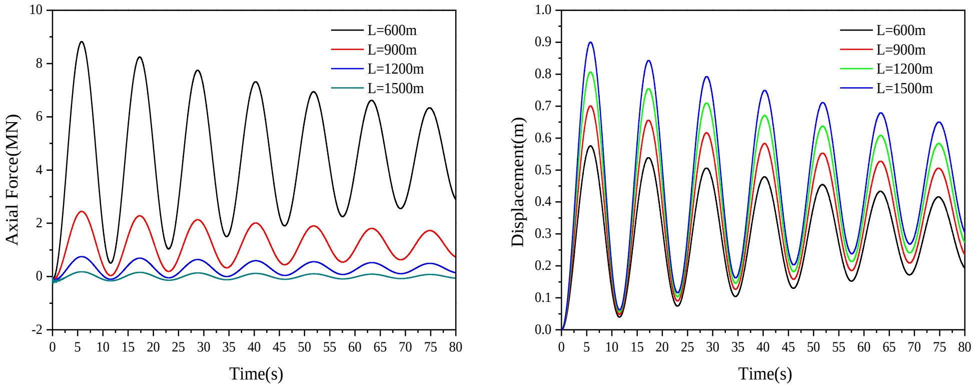
ABAQUS is used to verification the established mathematical model. As shown in figure 3, the results calculated by mathematical model are very close to the results obtained from ABAQUS. The maximum relative error of axial force on the top end of riser and the displacement of tensioner are 5.2% and 4.5% respectively, which verify the established coupled dynamic model in this paper.

3.2. Discussion

3.2.1. Riser length

The dynamic responses of the RT system with different riser length are shown in Figure 4. In figure 4(a), it can be found that the axial force on the top end of riser decreases gradually with the increase of riser length. The main reason is that the system energy is dissipated by tensioner damping. In figure 4(b), the displacement-time curve of the endpoint of riser shows that the longer the riser is, the larger of the amplitude. This is because each node of riser has certain displacement, the longer riser, the greater the accumulated value of displacement.

In figure 4, with the decreasing of riser length, there are no significant change of tensioner displacement and velocity. But the length decrease can lead to the increase of axial force on the top of riser. In this case, the instrument attached to the top end is more likely to be damaged. On the contrary, if the riser length increases, the axial force on top end of the RT would reduce, whereas the amplitude of endpoint of the RT would increase. Therefore the riser length should be selected in a reasonable range. Through comprehensive comparison in figure 4, it can be found that the dynamic response of the RT system can be controlled in a reasonable range when the range of riser length is selected between 900m~1500m.

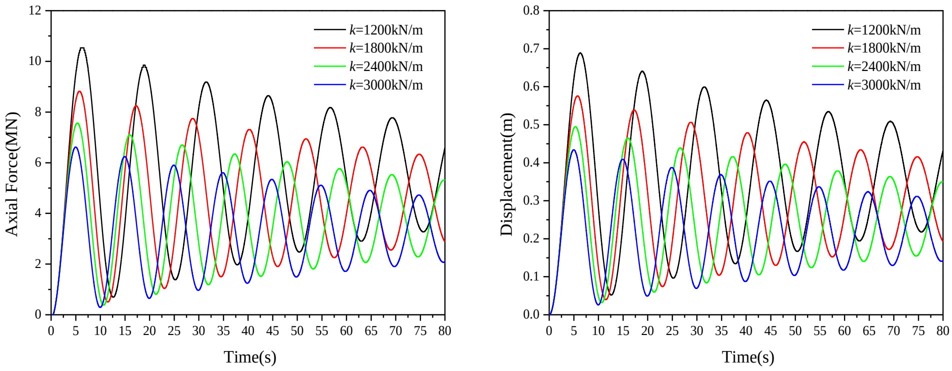


(a) Axial force on top end of riser (b) Displacement of endpoint of riser

Figure 4: Dynamic response time-history curve of RT system with different riser length

3.2.2. Stiffness coefficient of tensioner

Dynamic responses of the RT system with different stiffness coefficient of tensioner are shown in figure 5. With the increase of stiffness coefficient, the axial force on top end of riser, the displacement of endpoint and the velocity of tensioner decrease. Therefore increasing tensioner stiffness coefficient is beneficial to the safety of riser. But increasing the tensioner stiffness coefficient can bring difficulties to the tensioner design. The value of tensioner stiffness coefficient should be selected in a reasonable range. In this paper, the study found that the dynamic response of the RT system can be controlled in a reasonable range when the range of tensioner stiffness coefficient is selected as 1800~3000 kN/m.

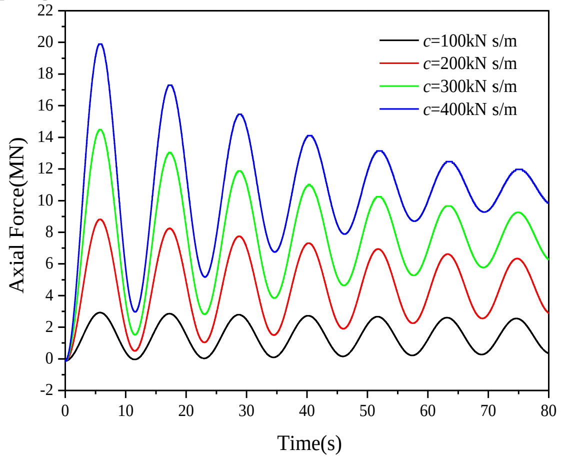


(a) Axial force on top end of riser (b) Displacement of endpoint of riser

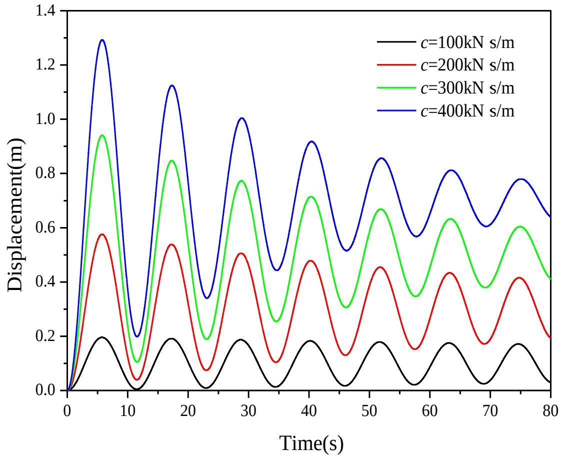
Figure 5: Dynamic response time-history curve of RT system with different tensioner stiffness coefficient

3.2.3. Damping coefficient of tensioner

Dynamic responses of the RT system with different tensioner damping coefficient are shown in figure 6. The axial force on top end and the displacement amplitude of the endpoint obviously increase with the increase of tensioner damping coefficient. Therefore the increase of damping coefficient may result in the failure of equipment on riser. The figure 6 shows that the displacement and velocity decrease with the increase of damping coefficient. Then, the helical buckling of riser at the bottom of tensioner is reduced. Therefore, the value of tensioner damping coefficient in a reasonable range is necessary. Through the analysis and comparison in figure 6, the reasonable range of tensioner damping coefficient is 300~400kN•s/m.



(a) Axial force on top end of riser



(b) Displacement of endpoint of riser

Figure 6: Dynamic response time-history curve of RT system with different tensioner damping coefficient

4. Conclusions

In this paper, a coupled dynamic model of the RT system and its differential scheme are presented. Based on the model, the dynamic behavior of RT system in a float platform in South China Sea is investigated. The following conclusions are obtained.

1) Compared with the existing models, the coupled dynamic model in this paper is more close to the practice RT system, since an actual elastic pipe with limited length is considered in the model and the vibration response of any cross section of the riser can be obtained.

2) The influence of riser length, tensioner stiffness coefficient, and tensioner damping coefficient to RT system are investigated. The study found that the dynamic response of the RT system can be controlled in a reasonable range when the range of riser length, tensioner stiffness coefficient, and tensioner damping coefficient respectively are selected as 900~1500 m, 1800~3000kN/m, 300~400kN•s/m.

Acknowledgements

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