

Viscous Damping Effect Investigation on Global Performance of SPM Buoy in Shallow Water

Ma Shan¹, Shi Shan², Miao Wenju³

¹ Deepwater Engineering Research Center, Harbin Engineering University, Room703, Ship&Ocean Building, No.145
Nantong Street, Harbin, China, 150001
E-mail: mashan0451@126.com

² Deepwater Engineering Research Center, Harbin Engineering University, Room703, Ship&Ocean Building, No.145
Nantong Street, Harbin, China, 150001
E-mail: shan.shi@deepoil.com

³ Deepwater Engineering Research Center, Harbin Engineering University, Room706, Ship&Ocean Building, No.145
Nantong Street, Harbin, China, 150001
E-mail: miaowenju@yahoo.com.cn

Abstract

In this paper, firstly the wave induced motions of a single point mooring (SPM) buoy in shallow water is solved based on linear potential theory in frequency domain. In order to investigate the viscous damping effect from the skirt plate and the buoy cylinder on the wave induced motions, Morison's equation is used to calculate the viscous drag force on the buoy in waves. Via numerical comparison of the motion response it is found that the viscous damping effect is significant in reducing the unreasonable resonant peak of the surge and pitch motions. Without including these effects, the resonant peak of wave induced motions will be overestimated. Furthermore a hull/mooring dynamic coupled program HARP is used to check the influence of viscous damping on the global performance of a buoy with catenary anchor leg mooring (CALM) system in survival conditions.

Keywords: Single point mooring (SPM) system, wave induced motions, coupled dynamic analysis, viscous damping effect, global performance.

Introduction

The single point mooring system typically consists of a floating buoy and subsea pipelines leading to the onshore storage facility. The offloading tanker will transfer the oil to the buoy through the floating hoses. The under buoy hoses, which connect the buoy with the subsea pipeline will convey the oil to the subsea pipelines. In order to prevent undesired drift motion of the buoy, several mooring line is moored to it and anchored at seafloor. Nowadays the SPM system with catenary anchor leg mooring (CALM) has been installed at hundreds of locations worldwide. This well proved technique is still in great demand for loading and offloading terminals in shallow water region. In survival environmental condition, the accurate prediction of the global performance of the moored buoy is important for proper design of the mooring system and under buoy hoses. [1] performed the model test of the hydrodynamic property

for the single point mooring system in deepwater. Based on Morison's equation the viscous damping from the skirt plate is investigated on the hydrodynamic damping contribution of the whole buoy system. [2] studied the pitch motions of a deepwater SPM system in waves. The theoretical results are compared with the model tests, it is concluded that the viscous damping from the skirt plate influences the accurate prediction of the heave and pitch motions of the moored buoy. In this paper, the wave induced motions and mooring line tensions of a CLAM system in shallow water is studied. Firstly the motion performance of the moored buoy is analysed in the frequency domain. The viscous damping effect of the skirt plate and the cylinder on buoy motions is investigated. Secondly the vessel mooring-riser coupled dynamic analysis program HARP is used to check the influence of the viscous damping on the motions of the buoy and the mooring tensions.

1 Description of the SPM system

In the present study, the single point mooring system consists of a buoy with catenary spread moorings. In the operation environmental condition, the tanker will be connected to the moored buoy via two hawsers and transfer crude oil from loading tanker to the buoy through a cargo floating hose. The under buoy risers which connect the pipeline end manifold and the buoy are used to export the oil from the buoy to the pipeline in the seafloor. At last, the crude oil will be sent to the onshore refinery via the pipeline. In the design storm environmental conditions, the moored buoy will stand alone without the tanker. Presently the moored buoy motion in the survival storm condition is mainly concerned. Table 1 shows the primary technical data of the buoy. The single point mooring system has six mooring lines with equally separation angle of 60 degrees. The mooring lines are selected as steel chains. Table 2 shows the material and dynamic property of the steel chains in the numerical model. The whole system is intended to be deployed in water depth of 26 meters.

Table 1: the primary technical data of the buoy

Item	Data
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Buoy mass	176 tons
Outer diameter	10 m
Inner diameter	3.57 m
Skirt diameter	13.87 m
Buoy Height	4 m
Draft at installed condition	2.54 m
Vertical height of gravity	0.23 m
Buoy roll radius of gyration	3.64 m
Buoy pitch radius of gyration	3.64 m
Buoy Yaw radius of gyration	4.57 m

Table 2: The material and dynamic properties for catenary mooring line in the numerical model

Material	Steel Chain
Diameter (cm)	6.35
Length (m)	350
Axial Stiffness (KN)	1273000
Dry Weight (kg/m)	88.42
Wet Weight (kg/m)	76.92
Inertial Coefficient	2
Drag Coefficient	2.4

2 The numerical modeling of the moored buoy motion response in frequency domain

Strictly speaking, the accurate analysis of responses of moored buoy in survival conditions needs to consider some important nonlinear factors including the mooring line restoring forces, Froude-Krylov forces and hydrodynamic loads due to large amplitude motions of the buoy. The whole numerical simulation has to be implemented in the time domain. The present paper will focus on the viscous damping effect of the buoy on the wave induced motions. Firstly the motion equations of the buoy is linearized and approximately solved in the frequency domain.

Assuming the moored buoy is performing oscillatory motions in small amplitude regular waves in water depth H . In order to express the incident wave, the buoy motions and the fluid flow, two Cartesian coordinate systems are defined. $o-xyz$ is defined as the earth fixed coordinate system. The origin O of the coordinate system is situated in the center of the waterline of buoy at the calm water surface. The z axis is vertically upward, the $o-xy$ plane coincides with the calm water surface. The incident wave potential and the fluid flow are expressed in this coordinate system. $o_b-x_b y_b z_b$ is defined as the body fixed coordinate system. At undisturbed position of the buoy, this coordinate system coincides with the earth fixed coordinate system. The oscillatory displacement of the buoy can be

measured using the relative position of $o_b-x_b y_b z_b$ to $o-xyz$. Assuming the interactions between the oscillatory buoy and the incident wave has reached a periodical state, the oscillatory displacement of the buoy can be represented as follows:

$$\{\eta(t)\} = \{\eta_a\} e^{i\omega t} = (\eta_{1a} \eta_{2a} \eta_{3a} \eta_{4a} \eta_{5a} \eta_{6a})^T e^{i\omega t} \quad (1)$$

Where $\{\eta_a\}$ is the six displacement component amplitude complex matrix of the buoy.

The fluid potential around the buoy can be expressed as:

$$\Phi(x, y, z, t) = \text{Re} \left\{ \left[\zeta_a (\phi_0 + \phi_j) + \sum_{j=1}^6 \eta_{ja} \phi_j \right] e^{i\omega t} \right\} \quad (2)$$

Where ζ_a is the amplitude of incident wave, ϕ_0 is the incident wave potential with unit amplitude, ϕ_j is the diffraction potential with unit amplitude due to disturbance of buoy with the incident wave. ϕ_j is radiation potential due to oscillatory displacement of buoy in j th direction with unit amplitude.

The incident wave potential ϕ_0 can be expressed as follows:

$$\phi_0 = \frac{ig}{\omega_0} \cdot \frac{ch[k_0(z+h)]}{ch(k_0 h)} \cdot e^{-ik_0(x \cos \beta + y \sin \beta)} \quad (3)$$

The fluid potential ϕ_j , $j = 1, 2, \dots, 6$ and ϕ_j can be obtained using the three dimensional boundary element method based on free surface pulsating source Green function. Once the numerical solution has been solved, the Bernoulli's equation can be used to get the hydrodynamic pressure acting on the buoy in waves. Integration of the fluid pressure on the mean wetted surface of the buoy, we can get the linear fluid forces on the buoy. Finally the motion equations of the moored buoy can be given as follows:

$$([M] + [A])\{\ddot{\eta}(t)\} + [B]\{\dot{\eta}(t)\} + ([C] + [D])\{\eta(t)\} = \{f(t)\} = \{f_a\} e^{i\omega t} \quad (4)$$

Where $[M]$ is the mass matrix of the buoy, $[A]$ and $[B]$ are the fluid added mass and wave damping matrix due to the harmonic oscillations of the buoy in calm water, $[C]$ is the hydrostatic restoring force matrix due to the offset of the buoy from its balanced positions. $[D]$ is the restoring force matrix due to the tensions of the mooring system. $\{f_a\}$ is the wave exciting force complex amplitude matrix from the incident wave force and diffraction wave force together.

Here we discuss the steady response of the moored buoy in waves. Replacing expression (1) into equation (4), we can get the following complex systems of equations with the unknown $\{\eta_a\}$:

$$\{-\omega^2[M + A] + i\omega[B] + [C + D]\}\{\eta_a\} = \{f_a\} \quad (5)$$

Solving equation (5), we can get the unknown six component displacement amplitude $\{\eta_a\}$.

In equation (4) the damping matrix [B] includes only the contributions of damping from wave radiation. In order to consider the influence of viscous forces of buoy structures on the wave induced buoy motions, the drag force acting on the skirt plate and the cylinder of the buoy needs to be considered and empirically expressed in the following terms:

$$F_D = -\frac{1}{2} \rho C_D S |V_R| V_R \quad (6)$$

Where C_D is the drag coefficient which is dependant on many parameters such as the shape of the body, Reynolds number, Keulegan-Carpenter number etc. V_R is the relative velocity of the body to the surrounding wave particle normal to the structure surface S.

In the present case, the skirt plate is placed horizontally and it will produce the vertical drag force and corresponding rolling and pitching moment on the buoy when the buoy is moving vertically in the waves. In order to calculate the vertical drag force and corresponding moment, the skirt plate is discretized into about 16 parts. The drag force and moment on each part can be deduced from formula (6). Make a summation of the drag force and moment contribution from these small parts of the plates, the total drag force and moment from the whole skirt can be obtained.

The vertical cylindrical hull of the buoy will produce the horizontal drag force and corresponding rolling and pitching moment when the buoy is moving horizontally in the waves. In order to calculate the drag forces and moments, the cylindrical hull is discretized into 20 smaller cylinders. The drag force and moment from each smaller cylinder can be reduced from the formula (6). Make a summation of the drag force contribution from these smaller cylinders, the total drag force and moment from the cylindrical hull can be obtained.

In principle once the total drag forces and moments from the skirt plate and the cylinders have been calculated, their influence on the wave induce motion can be investigated by including the viscous force in equation (5). One thing to be mentioned is that the drag force based on formula (6) is proportional to the square of the relative velocity V_R and needs to be linearized for the convenient of solving the motion equations in a linear manner. Assuming formula (6) can be linearized in the following equation:

$$F_D = -\frac{1}{2} \rho C_D S A_0 V_R \quad (7)$$

Where A_0 is the unknown coefficient, which can be determined by equalizing the work done by the drag force expressions (6) and (7) in a regular wave period namely:

$$W = \int_0^T V_R |V_R| V_R dt = \int_0^T A_0 V_R V_R dt \quad (8)$$

Where the relative velocity takes the harmonic form:

$$V_R = V_{R0} \cos \omega t \quad (9)$$

Where V_{R0} is the amplitude of the relative velocity between the body and the wave particle.

Solving equation (8), the unknown coefficient A_0 can be obtained:

$$A_0 = \frac{8}{3\pi} V_{R0} \quad (10)$$

Using equation (7), the drag force and moment from the vertical cylindrical hull and the skirt plate on the buoy can be expressed into two parts: one part is the multiplication of the viscous damping coefficient matrix $[B^v]$ with the velocity matrix $\{\dot{\eta}(t)\}$. Another part is the viscous wave exciting force $\{f_v\}e^{i\omega t}$ which is related to the velocity of the wave particle. Replacing these two parts into equation (4), we can get the following equations of buoy motions in waves:

$$([M] + [A])\{\ddot{\eta}(t)\} + [B + B^v]\{\dot{\eta}(t)\} + ([C] + [D])\{\eta(t)\} = \{f(t)\} = \{f_a + f_v\}e^{i\omega t} \quad (11)$$

When solving equation (11), it can be found that there is unknown displacement amplitude $\{\eta_a\}$ in $[B^v]$ and $\{f_v\}$. The iteration algorithm needs to be used to solve the equation (11). The following flow chart Fig. 1 shows how the motion equation is solved including the viscous drag forces.

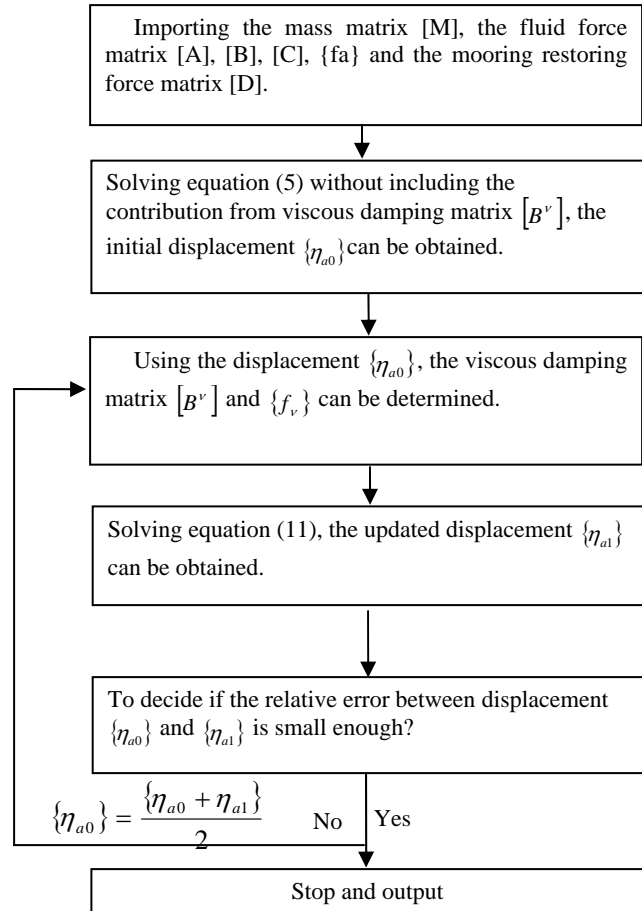


Figure 1: The flow chart of computation procedure for wave induced buoy motions

3 Numerical results and discussion

According to the numerical algorithm introduced above, the wave induced motions of SPM buoy in regular wave is solved. Firstly the fluid force matrix $[A]$, $[B]$, $[C]$, $\{f_a\}$ have been calculated by using three dimensional panel method program WAMIT [3]. The mooring stiffness restoring matrix $[D]$ is post processed using the numerical results based on HARP (Hull-mooring And Riser Program) whose basic theory and numerical method are given for example in Ref. [4],[5],[6]. With regarding to the drag coefficient used in the cylinder and skirt plate of the buoy, the value is approximated determined referring to [7] at steady flow condition. C_d is taken as 1.0 for the cylinder and 1.9 for the skirt plate.

Figs. 2~4 show the surge, heave and pitch response amplitude operator (RAO) at heading waves according to the numerical algorithm introduced in this paper. Three curves are compared together in each figure. Among them “Viscous damping from cylinder and skirt plate” means that the motion equation (11) is solved including the viscous damping from the cylinder and skirt plate together. “Viscous damping from skirt plate” means that the motion equation (11) is solved including the viscous damping from the skirt plate only. “Viscous damping from cylinder” means that the motion equation (11) is solved including the viscous damping from the cylinder only. By comparison the results among them, the viscous damping effect on the wave induced motion of moored buoy can be investigated.

Fig. 2 shows that all the three curves have a peak value at about 7.9 second. This peak should be induced by resonant response of buoy due to mooring restoring stiffness in the horizontal direction. This resonant peak also means that the mooring restoring force can affect the wave induced motion of the buoy in shallow water location. By observing these results it can be seen that the surge RAO by “viscous damping from skirt plate” has an unreasonable resonant peak, whereas the results by “viscous damping from cylinder” reduces the resonant peak a lot. It means that the viscous damping from the cylinder can not be neglected and can provide a significant amount of damping effect to decrease the unreasonable resonant peak of the surge motions.

Fig. 3 shows that all the three curves agree with each other very closely. It means that the vertical viscous damping from the skirt plate has a relative small contribution to reduce the resonant heave peak value.

Fig. 4 shows that the pitch RAO curve has two peaks. One peak at about 7.9 second has the same peak period with the surge peak period shown in Fig. 2. It means that this peak value is induced by the resonant surge motions. The pitch motion by “viscous damping from skirt plate” has a relatively large peak value at 7.9 second because the unreasonable large surge motion couples with the pitch motion due to lack of enough viscous damping in surge direction. Another peak happens at about 4.5 second. This peak should be related to resonant pitch motion. When considering the viscous damping from the skirt plate, it is seen that resonant peak can reduce further. It can be

concluded that the viscous drag force from the skirt plate provides a significant damping to depress the pitch motions.

In order to further confirm the effect of viscous drag force on global performance behavior of moored buoy in survival condition. The dynamic coupled analysis program HARP is used to model the SPM buoy response in design storm condition shown as in Table 3. In the hydrodynamic model of the buoy, Morison’s elements are used to approximate the viscous forces on the cylinder and skirt plate of the buoy system. The cylinder of the buoy is modeled using truss elements. The skirt plate of the buoy is modeled using plate elements. The drag coefficients of the cylinder and the skirt plate of the buoy have taken the same value as used in the previous wave induced motion analysis.

In order to investigate the viscous damping impact from these members on the global performance of SPM system. The numerical simulation is performed in three cases separately. Namely the viscous damping from the skirt plate and cylinder together, from the skirt plate only and from the cylinder only is included separately in the numerical simulation. The dynamic response of the SPM system from these three cases will be compared with each other.

In the coupled dynamic analysis of the buoy and mooring lines the mooring line is modeled and solved using the elastic rod theory [8]. 30 elements are used to discretize each mooring line. The whole simulation time lasts for 1000 seconds. The time step used in the numerical simulation is 0.01 second. Fig. 5 shows the numerical model of the SPM buoy used in the coupled dynamic analysis program HARP.

Figs. 6-9 show the time series of the surge, heave and pitch motions of the buoy and maximum loaded 1# mooring line tensions at fairlead position. It is observed that the results from three different way of inclusion the viscous damping agree to each other closely. Table 4 compares the statistical value of the motion response of the buoy and maximum loaded 1 # mooring line tensions. It is further confirmed that results of the global performance of the SPM system agree closely to each other for the given storm conditions. The reason can be explained in the following.

As it can be seen from Fig. 2-4, the difference of surge and pitch RAO mainly locates at resonant peaks from three different way of including the drag damping. The resonant periods for the surge and pitch motions of the moored buoy are less than 10 seconds. The wave induced first order motion will dominate the buoy motions. In Table 3 for the given storm conditions the wave peak period is 16 seconds and the period of main wave energy is far from the periods when the resonant surge and pitch happens. It means that the difference of surge and pitch RAO from three different way of including the drag damping have little contribution to the final dynamic response of the coupled buoy and mooring system in the given environmental conditions.

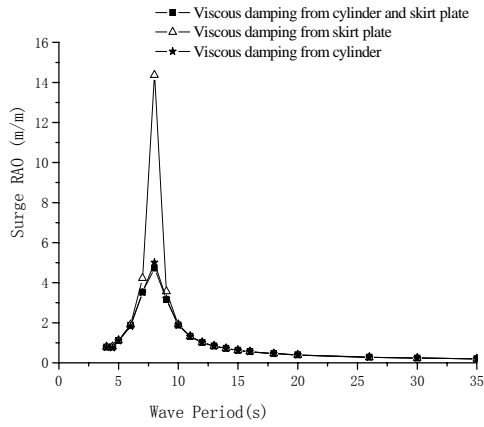


Figure 2: Surge RAO comparison of moored buoy considering different contribution of viscous damping

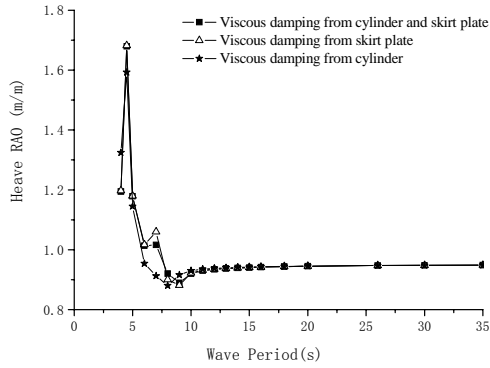


Figure 3: Heave RAO comparison of moored buoy considering different contribution of viscous damping

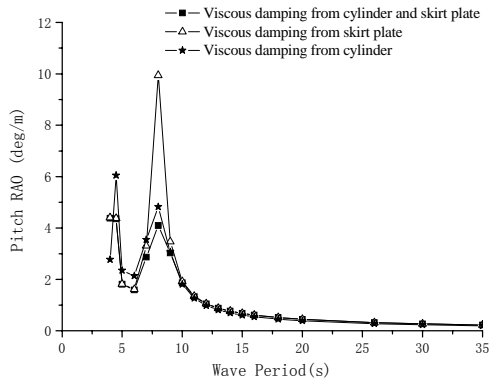


Figure 4: Pitch RAO comparison of moored buoy considering different contribution of viscous damping

Table 3: The survival environmental condition used in the SPM buoy global performance analysis

Environmental conditions	Item	Data
Wave	Hs (Significant wave height)	9.45 m
	Tp (Peak period)	16.0 s
Wind	V ₁₀ (mean wind speed per hour at 10 m elevation)	21.7 m
Current	Surface velocity	1.54 m/s
	Bottom velocity	0.2 m/s

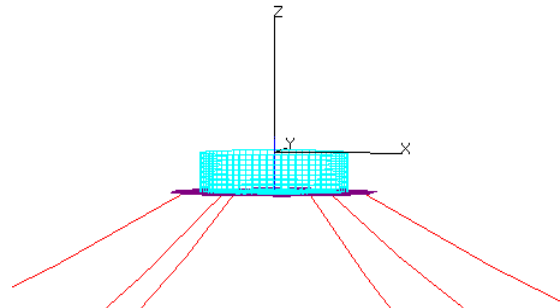


Figure 5: The Numerical model of the SPM buoy in HARP program

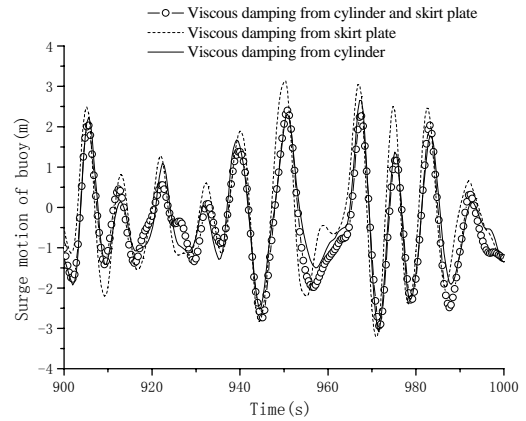


Figure 6: Surge motion comparison of moored buoy considering different contribution of viscous damping

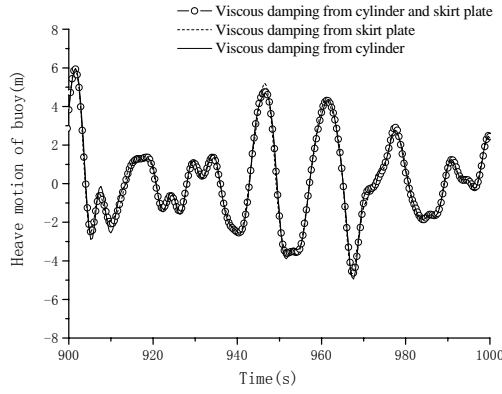


Figure 7: Heave motion comparison of moored buoy considering different contribution of viscous damping

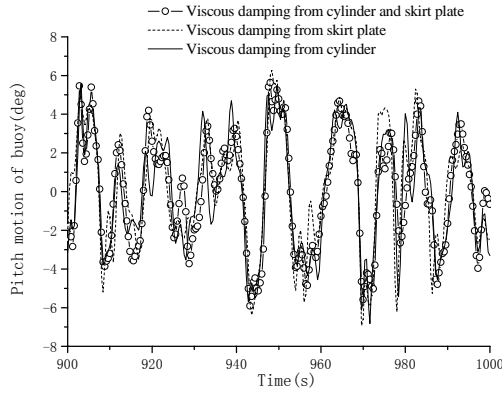


Figure 8: Pitch motion comparison of moored buoy considering different contribution of viscous damping

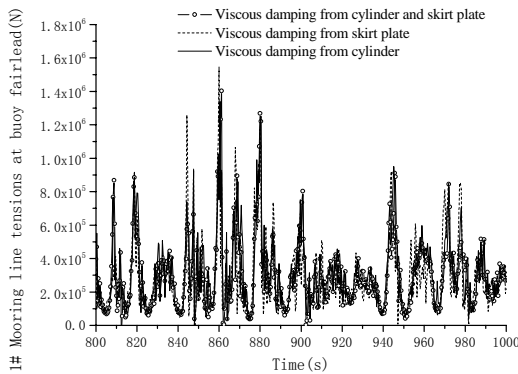


Figure 9: Mooring line tension comparison of moored buoy considering different contribution of viscous damping

Table 4: The statistical data comparison of SPM buoy motions and mooring line tension response considering different contribution of viscous damping

Response of the moored buoy	Statistical data	Viscous damping from cylinder and skirt plate	Viscous damping from skirt plate	Viscous damping from cylinder
Surge motion of buoy (m)	Mean value	-0.25	-0.06	-0.21
	Standard Deviation	1.57	1.57	1.54
	Maximum value	3.81	3.90	3.83
	Minimum Value	-3.26	-3.34	-3.35
Heave motion of buoy (m)	Mean value	0.1	0.07	0.09
	Standard Deviation	2.37	2.37	2.39
	Maximum value	7.69	7.77	7.72
	Minimum value	-6.59	-6.48	-7.05
Pitch motion of buoy (deg)	Mean value	0.1	0.21	0.08
	Standard Deviation	3.22	3.34	3.24
	Maximum value	6.90	7.10	7.75
	Minimum value	-8.64	-8.45	-8.86
1# mooring line tension at fairlead(K N)	Mean value	265.5	252.9	265.4
	Standard Deviation	205.5	193.8	204.5
	Maximum value	1589.0	1551.2	1563.1
	Minimum value	-64.10	-125.4	-87.9

4 Conclusion

This paper discusses the viscous drag damping effect on the global performance of SPM buoy in survival condition. Firstly the wave induced equation of the moored buoy is approximated established in frequency domain. The drag damping forces from the cylinder and skirt plate are linearized and included in the motion equations. It can be concluded that the vertical cylindrical hull can provide a significant damping to reduce the unreasonable resonant peak of surge motion. Whereas the skirt plate can provide significant damping to reduce the resonant peak of pitch motions. Finally the coupled dynamic program HARP is used to simulate the coupled dynamic response of buoy and mooring system in the given survival environmental conditions. Due to the peak period of wave where the main wave energy locates is far away from the resonant period of surge and pitch motion, the viscous damping effect from

the cylinder and the skirt plate does not affect significantly on the global performance of the SPM buoy.

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References

- [1] H. Cozijn, R. Uittenbogaard and E. Brake. *Heave, roll and pitch damping of a deepwater CALM buoy with a skirt*. Proceedings of the 15th Inter. Offshore and Polar Engineering Conference, Seoul, Korea, June 19-24, pp. 388-395, 2005.
- [2] S. Ryu, A.S. Duggal, C.N. Heyl and Y.H. Liu. *Prediction of deepwater oil offloading buoy response and experimental validation*. Intern. Journal of Offshore and Polar Engineering, Vol. 16, No. 4, pp. 290-296, December 2006.
- [3] *WAMIT user manual*, Version 5.4, WAMIT, Inc.
- [4] M.H. Kim, Z. Ran and W. Zheng. *Hull/Mooring/Riser coupled dynamic analysis of a truss spar in time domain*. Proc. ISOPE'99, Brest, Vol. 1 301-308
- [5] M.H. Kim, A. Tahar and Y.B. Kim, *Variability of TLP motion analysis against various design/methodology parameters*. 11th Proc. ISOPE'01, Stavanger, Norway
- [6] Z.H. Ran, *Coupled dynamic analysis of floating structures in waves and currents*, Doctoral dissertation, Texas A&M University, 2000
- [7] *Environmental conditions and environmental loads*, Recommended practice DNV-RP-C205, April 2007
- [8] D.L. Garrett. *Dynamic analysis of slender rods*. Journal of Energy resources technology, Transactions of ASME, Vol. 104, 302-307, 1982