Some Considerations How to Reduce the Motions of Ships Moored at an Open Berth

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オープン・バースに係留された船舶の運動低減に関する考察

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船型の増大に伴い、ターミナルは水深の大きな沖合いに建設されている。その結果、船は大きな波を受け、この時の船体動揺が安全荷役や安全係留に影響を与えている。そこで船体動揺を抑え、係留施設の最も安全な利用のあり方を検討することが重要となる。係留船の運動や係留索力のシミュレーションを行う方法としては数値計算による方法と水理実験による方法とがあるが本研究では水理実験により船体運動低減化の方法について議論を行っている。得られた結果を要約すれば次のようになる。

- (1) サブハーモニック波による長周期運動はサージとスウェーで卓越し、一方ヒーブ、ロールそしてピッチは一次の波成分により引き起こされている。ヨーは長周期と短周期の両方で応答する。
- (2) サージとヨーの長い固有周期は堅い係留索（テイルロープ付きワイヤー）を用いることによりかなり容易に変えることが出来る。もしサージとヨーの固有周期を一次の波とサブハーモニック波の卓越周波数の谷間に来るように設定すれば、長周期船体運動はかなり低減することができる。
- (3) 長周期船体運動を低減するための第1の方法として固有周期の制御が試みられるべきである。第2の方法としてダッシュポット・システムが有効である。
- (4) 係留索とフェンダーの係留力の非対称性のためにスウェーはサブハーモニック運動を起こす。係留索のパネ定数を堅くすることは非対称性の程度を減少させ、その結果としてスウェーを減少させる。
- (5) 係留索のパネ定数を変えることによって、ロール、ピッチそしてヨーの短い固有周期を変えることは難しい。この場合にはダッシュポット係留索が船体運動の低減に有効である。

1. Introduction

Due to the continuous increase in vessel sizes more and more docks and terminals are constructed offshore in deeper water. As a consequence, ships are exposed to a more severe wave climate, and the resulting ships motions adversely affect cargo handling and safe mooring. Therefore, the application of modern techniques to reduce ship motions and to guarantee the most effective utilization of berthing facilities is very important. Physical as well as numerical methods are commonly used to simulate the motions and line forces of moored ships. This paper will discuss methods to decrease ship motions in a physical simulation.

2. Experimental Methods

It is well known that the floating vessel is very sensitive to minute changes in the environmental parameters as well as in the actual properties of the mooring system. As shown in Table 1, we list up many factors which should be considered in the experiment. Among others the correct

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Table 1 Items of Ship Mooring Simulation

Bathymetry	Plan View, Variable Depth
Natural Condition	Wave: Irregular Waves, Wave Grouping, Subharmonic Waves Wind: Steady Wind, Gustiness Current
Ship Condition	Arbitrary Ship Shape GM, Natural Period of Roll
Mooring Condition	Mooring Line: Nonlinearity, Hysteresis, Eternal Extension, Initial tension Fender: Buckling-Type, Pneumatic-Type, Hysteresis Mooring System: Asymmetrical Parameter (α) $= (\text{Initial Spring Constant of Fender} / \text{Initial Spring Constant of Mooring Line for Sway})$ Dash-Pot System
Threshold	Operational Condition (Cargo Handling): Ship Motion Survival Condition (Mooring): Mooring Line, Fender

Fig. 1 Ship Model Basin

reproduction of nonlinear and hysteretic characteristics of mooring lines (Wilson, 1967)⁽²⁾ and fenders (Ueda, 1987)⁽³⁾ and of the irregular wave conditions are important. Since sub-harmonic waves (Barthel et al, 1983)⁽⁴⁾, created by wave grouping, can easily excite long period ship motions, the correct generation of the irregular sea state is of particular importance. Fig. 1 shows an arrangement of a ship model basin. At a scale of 1:70, the model of a bulk carrier of l. o. a. 195m, breadth 28.6m and draft 11.34m at full load condition was moored at the offshore terminal which was fully open and pile-supported. The ship was subjected to bow waves at 30 degrees. The average water depth is 15m. Selected irregular wave trains of duration 20 minutes were generated in deep water offshore.

The detail mooring arrangement with a total number of 14 lines is shown in Fig. 2. As shown for line 1/2, sometimes one simulator was used to reproduce the characteristics of two mooring lines if they were attached to the same bollard.

Fig. 3 shows a mooring line simulator (Barthel et al, 1990)⁽⁴⁾. The non-linearity of the mooring

Fig. 2 Mooring Arrangement

Fig. 3 Mooring Line Simulator

Fig. 4 Characteristics of Soft Mooring Line
(64mmφ Polypropylene Braid Rope)

line is obtained by an arrangement of springs attached at a certain angle. Hysteresis characteristics are added by squeezing the line with a spring to generate Coulomb friction. Mooring forces were measured using load cells built into the line simulators. Prior to the experiment, an initial pretension of 10t was added to each line.

Fig. 4 and 5 display the characteristics of a soft (polypropylene) and a hard (steel) mooring line for the case of LINE1/2, respectively. It was attempted to reduce ship motions by using hydraulic dash-pots. Line 3, 7, 8 and 12 can be used for this purpose. Dash-pots are only effective in the pull process for the sake of feasibility in the field. A counter weight (10t) and a nonreturn valve, as shown in Fig. 6, support the reverse motion. A clearance between piston and cylinder induces a damping force by viscous resistance. From the maintenance point of view, this clearance precludes excessive wear of the piston and its housing. To prevent fluttering of the piston, a dummy piston with three big holes is arranged in the cylinder. Fig. 6 shows also the relation between piston velocity and load. Under the assumption of a general Couette flow (Schlichting, 1968)⁽⁵⁾ through the gap, the damping coefficient k can be obtained by the following equation:

$$k = \frac{A_P \cdot \Delta P}{U} = \frac{12\mu l (A_c - \pi D_c h_0) \left(A_c - \frac{\pi D_c h_0}{2} \right)}{\pi D_c h_0^3}$$

Fig. 5 Characteristics of Hard Mooring Line
(44mm ϕ Steel Wire Rope with 10m
Polypropylene Tail Rope)

Fig. 6 Characteristics of Dashpot
(Silicone Oil with 20cst was used in
the experiment)

Fig. 7 Simulator of Buckling-Type Fender

where, A_p : an area of piston, Δp : difference of pressure at the both ends of piston, U : moving velocity of piston, μ : coefficient of molecular viscosity, l : thickness of piston, A_c : area of cylinder, D_c : diameter of cylinder, h_o : clearance between piston and cylinder. In the experiment, 20cst (cent-stokes) silicone oil is used. The theoretical and experimental results in Fig. 6 show a good agreement and seem to confirm the assumption made above.

Fig. 7 shows a simulator of a buckling-type fender (Barthel, 1990). The simulator has two springs. A compensator spring in a lower carriage gives an initial stiffness of the fender and another spring in the upper carriage gives a constant force of the fender. The bumper moves between two prongs which induces a coulomb friction reproducing the hysteresis of the fender.

Fig. 8 shows the characteristics of target fender and the experimental fender. The motions were monitored at a high degree of accuracy with an optical tracking system (SELSHOT). All measured data are stored through a central data acquisition system and are processed immediately after the test.

Photo 1 shows the experimental set-up.

3. Experimental Results

3.1 Effects of Mooring Lines on Ship Motions

Some experiments with different mooring line configurations as listed on Table 2 were carried out to investigate the effect of lines strength, length and orientation. For the sake of an easy com-

Fig. 8 Characteristics of Target Fender and Experimental Fender

Table 2 Mooring Line Configurations

Mooring system	Mooring Line Number									
	1/2	3	4/5	6	7	8	9	10/11	12	13/14
SOFT	B.R	B.R	B.R	B.R	/	/	B.R	B.R	B.R	B.R
SOFTDP2	B.R	S.W,D.P	B.R	B.R	S.W,D.P	S.W,D.P	B.R	B.R	S.W,D.P	B.R
SOFTDP3	B.R	S.W,D.P	B.R	B.R	S.W,D.P	S.W,D.P	B.R	B.R	S.W,D.P	B.R
SOFTDP4	B.R	S.W,D.P	B.R	B.R	S.W,D.P	S.W,D.P	B.R	B.R	S.W,D.P	B.R
HARD	S.W	S.W	S.W	S.W	/	/	S.W	S.W	S.W	S.W
HARDDP	S.W	D*S.W	D.S.W	/	S.W,D.P	S.W,D.P	/	D*S.W	D*S.W	D*S.W
HARDDP2	D*S.W	/	D*S.W	/	S.W,D.P	S.W,D.P	/	D*S.W	/	D*S.W
HARDDP3	D*S.W	/	D*S.W	/	S.W,D.P	S.W,D.P	/	D*S.W	/	D*S.W

Remark: B. R means a braid rope of 64 ϕ , D. P means a dash-pot mooring line, S. W means a steel wire rope of 44 ϕ with 10m double polypropylene tail rope and D means double lines.

Table 3 Mooring Line Strength (tons)

Mooring system	Mooring Line Number									
	1/2	3	4/5	6	7	8	9	10/11	12	13/14
SOFT	180	90	180	90	/	/	90	180	90	180
SOFTDP2	180	124	180	90	124	124	90	180	124	180
SOFTDP3	180	124	180	90	124	124	90	180	124	180
SOFTDP4	180	124	180	90	124	124	90	180	124	180
HARD	248	124	248	124	/	/	124	248	124	248
HARDDP	248	248	248	/	124	124	/	248	248	248
HARDDP2	496	/	248	/	124	124	/	248	/	496
HARDDP3	496	/	248	/	124	124	/	248	/	496

parison, mooring line forces were normalized by mooring line strength as shown in Table 3. Ship motions and fender deflections were normalized by limiting values for cargo handling in Table 4 and the allowable maximum deflection (55%) of the fender length, respectively.

Fig. 9 illustrates experimental results for a soft mooring system showing offshore significant

wave height versus normalized motions and mooring line forces. The limiting value (100%) are indicated: Heave, roll and pitch, which are excited by the first wave components, are directly proportional to the wave height. On the other hand, surge is apparently proportional to a higher power of the wave height. While pretension of the mooring lines prevents the direct surge response to first-order waves, the ship is being moved by the sub-harmonic waves which can be a consequence of wave grouping offshore.

Fig. 10 shows wave spectra at offshore and berth locations. HMO is the significant wave height based on the zeroth moment of the spectrum m_0 . With a sufficiently large wave height one can see the low-frequency components (sub-harmonic waves) in the spectrum. The peak period of the low frequency spectrum can be read approximately 110s from the enlarged figure. Fig. 9 indicates four cases of exceedance of the threshold with four limiting wave heights.

In the same manner, experimental results are summarized for a hard mooring system in Fig. 11, which shows exceedance of the threshold in three cases. Table 5 summarizes the limiting wave heights for various mooring systems. The combination of this table with Figures 9 and 11 allows for easy evaluation of the critical values of each mooring system.

Comparing Fig. 9 and 11, surge is decreased predominantly. Table 6 shows the natural periods of ship motions. From the table, one can see that surge has decreased substantially using the hard mooring system. From Table 6, which shows the natural periods of the six-degrees-of-freedom-motions of the vessel, we can see that the natural periods of surge and sway can easily be influenced by the stiffness of the mooring lines. Hence, by increasing the stiffness of the mooring system, resonance effects of the mooring system to the sub-harmonic waves at approximately 110 s can be avoided. The natural periods of a short period motion such as roll cannot be changed by

Table 4 Limiting Value of Ship Motions for Safety Cargo Handling

Mode	By Per Bruun	By Authors
Surge	$\pm 1.5\text{m}$	$\pm 1.5\text{m}$
Sway	$\pm 0.5\text{m}$	$\pm 1.5\text{m}$
Heave	$\pm 0.5\text{m}$	$\pm 0.5\text{m}$
Roll	$\pm 4^\circ$	$\pm 4^\circ$
Pitch	/	$\pm 1^\circ$
Yaw	$\pm 2^\circ$	$\pm 2^\circ$

Table 5 Limiting Wave Height for Various Mooring Systems

Mooring system	Offshore Wave Height Limit				Remark
	First	Second	Third	Fourth	
SOFT	1.85m fender1	1.95m pitch	2.50m fender2	2.80m fender1	soft line with hysteresis
SOFTDP2	1.7m fender1	2.1m fender2	2.2m pitch		soft line with hysteresis four dash-pots with 20cst
SOFTDP3	1.7m fender1	2.1m fender2	2.2m pitch		soft line with hysteresis four dash-pots with 5cst
SOFTDP4	1.7m fender1	2.05m fender2	2.1m pitch		soft line with hysteresis four dash-pots with 1cst
HARD	1.45m fender1	1.65m fender2	2.0m pitch		hard line with hysteresis
HARDDP	1.4m fender1	1.85m pitch	2.35m ML4/5		hard line with hysteresis roll dash-pots with 5cst
HARDDP2	1.4m fender1	1.65m fender2	2.05m pitch	2.09m ML4/5	hard line with hysteresis roll dash-pots with 5cst
HARDDP3	1.3m fender1	1.55m fender2	2.0m pitch	2.15m ML4/5	hard line without hysteresis roll dash-pots with 5cst

Table 6 Natural Period of Ships Motions

Mooring system	SURGE		SWAY		ROLL	
	SEC	Hz	SEC	Hz	SEC	Hz
SOFT	87.8	0.011	52	0.019	19.1	0.052
SOFTDP2	87.8	0.011	58	0.017	17.6	0.057
SOFTDP3	87.8	0.011	55.6	0.018	18.3	0.055
HARD	48	0.021	40	0.025	17.1	0.059
HARDDP2	48	0.021	40	0.025	17.1	0.059
HARDDP3	45.9	0.022	49.7	0.02	17.8	0.056

easy to reduce. However, it is difficult to influence the short period ship motions such as heave, pitch and roll by changing line stiffness. Sub-harmonic motions appear also in sway motions due to a restoring force asymmetry of mooring line and fender arrangement⁽¹⁾. For this, an increased stiffness will also effectively reduce motions.

Fig. 13 shows spectra of ship motions for various kinds of dash-pot systems. In the comparison

the stiffness.

3.2 Moored Ship Motions in the Frequency Domain

Fig. 12 shows spectra of ship motions for soft and hard mooring systems. Long-period motions are predominant for surge and sway, while heave, roll and pitch were excited by the first-order wave components. High and low frequency components were responsible for yaw. We see again that the long-period ship motions, surge, sway and yaw, are

Fig 9 Experimental Results for Soft Mooring System

Fig. 10 Wave Spectra at Offshore and Berth Locations

with Fig. 12, surge reduction by dash-pot arrangement is not very distinct. On the other hand, sway, yaw and roll are definitely decreasing due to the dash-pot system.

4. Conclusion

The experimental results can be summarized as follows:

- (1) Long-period motions due to sub-harmonic waves are predominant in surge and sway, while heave, roll and pitch are excited by the first-order wave components. High and low frequency components were responsible for yaw.
- (2) The long natural periods of surge and yaw can relatively easily be modified through the use of stiff (steel) mooring lines. If we adjust the natural periods of surge and yaw to fall between the frequency range of first-order and sub-harmonic waves, long-period ship motions can be substantially reduced.
- (3) Subharmonic motions appear in sway modes due to a restoring force asymmetry of mooring lines and fenders. Hardening the stiffness of mooring lines is effective to decrease the degree

Fig. 11 Experimental Results for Hard Mooring System

of the asymmetry and results in reducing the sway motions.

- (4) In connection with the long natural period control, secondary dash-pot systems could be very useful.
- (5) It is difficult to change the short natural periods of roll, pitch and yaw by varying the stiffness of mooring lines. In this case, the dash-pot mooring lines are effective to reduce the ship motions.

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Fig. 12 Spectra of Ship Motions for Soft and Hard Mooring Systems

Fig. 13 Spectra of Ship Motions for Various Kinds of Dash-Pot Systems

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質疑応答

鞠谷宏士：大変興味ある話有難うございました。長周期運動はダッシュポットで吸収することは難しいということでしたが、その理由について御伺いしたい。

久保雅義：長周期運動は係留索のバネ定数により固有周期をかなり容易に変えることができるので、これにより外乱の長周期より遠ざける方が運動を小さくできます。一方短周期運動は係留索で固有周期を変えることがほとんど不可能なので、この場合には dashpot により減衰力を付加することが有効です。勿論長周期運動でも dashpot で減衰させることは可能です。つまり dashpot で減衰力を加えると図に示すように応答関数は小さくなりますが、固有周期をスペクトルの谷間に移動させる方が、応答関数を下げる上でより有効であると考えています。