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An Estimation of Springing Responses for Recent Ships

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Abstracts

The estimation of springing responses for recent ships are carried out and application to a ship design are described. To this aim, springing effects on hull girder were re-evaluated including non-linear wave excitations and torsional vibrations of the hull. The Timoshenko beam model was used to calculate stress distribution on the hull girder by the superposition method. The strip method was employed to calculate the hydrodynamic forces and moments on the hull. In order to remove the irregular frequencies, we adopted 'rigid lid' on the hull free surface level and added asymptotic interpolation along the high frequency range. Several applications to the existing ships were carried out. They are Bishop and Price's container ship, S-175 container ship, large container, VLCC and ore carrier. One of them is compared with ship measurement result while another with that of model test. Comparison between analytical solution and numerical one for homogeneous beam type artificial ship shows good agreement. It is found that most springing energy came from high frequency waves for the ships having low natural frequency and North Atlantic route etc. Therefore, the high frequency tail of the wave spectrum should be increased by ω^{-3} instead of ω^{-4} or ω^{-5} for springing calculation.

Key words: Springing; quadratic strip method; hydro-elasticity; wave spectrum, high frequency tail, fatigue life

Introduction

The modern ships are getting bigger and lighter with shallow draft and large breadth, which makes the ship flexible with both her bending and torsional rigidities being small. For such ships, precise estimation of wave-induced bending moments, shear forces and torsional moments are important. Otherwise, flexible hull girder may seriously suffer from the fatigue damage due to wave excitation such as springing, whipping or slamming, etc. Ship springing is the resonant response of the ship to a hydrodynamic excitation due to incident waves. It could occur in relatively moderate sea states, when the encountering wave frequency matches the natural frequency of hull girder. Springing is excited by both linear and non-linear excitation mechanisms. The linear exciting forces are associated with waves of small wavelengths relative to the ship length. The non-linear ship response, which is quadratic to wave amplitude, comes from the sum frequency between the incident waves. Linear strip theory has been widely used for the past few decades because of its simplicity and surprisingly accurate prediction of ship motions and wave induced loads. However, the effect of flared sections, bottom emergence and reentry, green water on deck and steep waves cannot be explained by any linear theories. In order to take into account those effects and other nonlinearities, a three-dimensional time domain simulation method may be needed so that the exact free surface condition and body boundary condition on the instantaneous wetted surface are satisfied at each time step. However, it requires tremendous computer power and rather sophisticated numerical techniques. So, it is not so practical approach for early ship design stage. Instead, we considered the combined effects between the bending and torsional modes. We are concerned that combined effect may accelerate the fatigue damage due to higher number of stress cycles on the hull girder.

Linear theories that calculate the springing excitation and motion coefficients are usually represented by strip or slender-body approximations. Strip theories are usually based upon a part rational and part intuitive approach. Some of the theories in this category are those published by Belgova (1962), Goodman (1971), van Gunsteren (1974) and Hoffman and van Hoof (1976). Slender-body theory makes use of the technique of matched asymptotic expansions. See, for example, Bishop, Price and Tam (1977), Maeda (1980), Beck and Troesch (1980), or Skjordal and Faltinsen (1980). Chen and Chiou (1981) make a systematic comparison of some of the more common strip and slender-body theories. The nonlinear problem is complicated by the complex free surface and hull boundary conditions. Jensen and Pedersen (1981) and Watanabe and Soares (1999) have investigated the long wave, nonlinear effects of ship springing. Storhaug et. al.(2003) analyzed the measured data thoroughly for large bulk carrier and compared with estimated ones. Vidic-Perunovic and Jensen (2004) studied same ship and derived that the nonlinear interaction between two directional waves can cause big energy. Park et. al. (2004) carried out springing responses for modern merchant ships.

Vertical modes of vibrations have been the focus of interest because the vertical wave loads are the largest loads. However, for ships with very low torsional rigidity, anti-symmetric springing could also be expected (Wu and Ho, 1987) even if there is no experimental evidence.

In the present paper, the problem of dynamic response of the ship has been formulated based on hydro-elastic theory. Both the vibration analysis and the sea-keeping fluid forces on the hull were taken into account. It was performed theoretical formulations and numerical computations for predicting hydrodynamic forces on a ship advancing in waves. The strip method was employed to calculate the hydrodynamic forces and moments on the hull. Since the springing is generally excited by high frequencies, about 2 to 6 rad/sec, accurate estimation of added mass and damping coefficients in this region free from irregular frequencies is important. In order to analyze the vibration of a ship it was performed theoretical formulations and numerical computations for the equations of girder and solved with superposition method. The hull was modeled as a Timoshenko beam that accounts for the rotary inertia, shear deformation and cross-sectional warping stiffness to analyze the response to corresponding vibration mode. The developed analysis model is suitable for the evaluation of the springing damages of ships due to waves including torsion effect.

Several applications to the existing ships were carried out. First two of them are Bishop and Price's container ship and S-175 container ship. Remaining three are commercial ships of container, VLCC and ore carrier. One of them is compared with ship measurement result while another with that of model test. Comparison between analytical solution and numerical one for homogeneous beam type artificial ship shows good agreement. Special considerations including detailed springing calculation are found to be necessary for the ships having low natural frequency and North Atlantic route etc.

Ship Response

The vertical response of the advancing ship is described within the Timoshenko beam theory by the equations.

$$\frac{\partial u_z}{\partial x}(x,t)=\theta_y(x,t)+\gamma_y(x,t) \qquad (1)$$

$$EI(x)\left[1+\eta_y(x)\frac{\partial}{\partial t}\right]\frac{\partial\theta_y}{\partial x}=M_y(x,t) \qquad (2)$$

$$\frac{\partial M_y}{\partial x}=I_y(x)\frac{\partial^2\theta_y}{\partial t^2}-V_z(x,t) \qquad (3)$$

$$\frac{\partial V_z}{\partial x}+F_z^T(x,t)=\mu(x)\frac{\partial^2 u_z}{\partial t^2} \qquad (4)$$

$$k_z(x)A(x)G\left[1+\alpha(x)\frac{\partial}{\partial t}\right]\gamma_y(x,t)=V_z(x,t) \qquad (5)$$

These equations are reduced to the system of eight ordinary differential equations with boundary conditions at the stern and at the bow. The response of the elastic ship in waves is considered to be periodical in time with the frequency of oscillations being equal to the wave encounter frequency, ω_e. The boundary-value problem for the system of ODE is solved numerically with the superposition technique. Within the superposition technique the system of ordinary differential equations is solved five times with different initial conditions at the stern using the fourth-order Runge-Kutta method. After that the five solutions are combined to satisfy the boundary conditions at the bow. It is shown that this approach provides accurate solution if the encounter frequency is not too large.

In order to validate the numerical results, analytic solution under the uniform beam assumption is also derived (Park, Jung and Korobkin, 2004)

The strip method was employed to calculate the hydrodynamic forces F_z^T on the hull in Eq. 4 (Korobkin, 2003). Since the springing is generally excited by high frequencies, about 2 to 6 rad/sec, accurate estimation of added mass and damping coefficients in this region is important. In order to prevent the irregular frequencies, rigid rid technique (Hong, 1987) with asymptotic approximation method is adopted (Jung, Park, Shin, Park and Korobkin, 2003).

Equations, which govern anti-symmetric response of a ship in regular waves, can be formulated similar to Eqs. 1~5 (Korobkin, 2003).

Numerical Results

The theory developed in this paper has been used to predict the linear and nonlinear wave-bending moments for 5 ships. They are designated as ship A, B, C, D and E, respectively. Ship A is Bishop and Price container ship and ship B is S-175 container ship whose input data are well known for most researchers in this field. While, C is a container ship, D is a VLCC and E is an ore carrier built for commercial services and their principal dimensions are tabulated in Table 1. For the validation of the suggested numerical model, ship A and B are examined first. After that, for discussions and application, ship C, D and E are calculated.

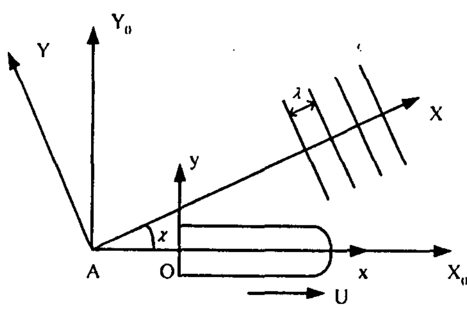

Fig. 1: **Local and global coordinate systems**

Table 1: **Principal particulars of the ships considered**

Ship	A	B	C	D	E
LBP[m]	281	175	319	320	294
B[m]	32.3	25.4	42.8	70.6	53
T[m]	12.2	9.5	13.0	8.35	12.21
Speed[m/s]	13.4	11.3	12.9	9.1	5.5

Numerical results for Ship A

Input data for ship A were taken from Bishop and Price (1979). The main dimensions of the ship are given in Table 1. Extensive numerical calculations for the ship A are carried out. Wave heading is selected as 135 degrees and advancing speed is 13.38m/sec for comparison purpose. Linear and nonlinear contribution on the vertical, horizontal and torsional moments was calculated and only linear contribution is plotted. In Fig. 2, vertical bending moments are compared with the SOST developed by Jensen and Dogliani (1996). Here, abscissa is encounter frequency in rad/sec and ordinate is normalized vertical bending moment by factor of ρgBL^2. Solid line represents the present calculation and dotted line that of Jensen. We can observe two distinct peaks around 0.8 and 5.2 rad/sec. The first peak comes from the wave and second peak comes from the resonance with the natural frequency of the ship and corresponds to the springing. We can see that good agreements are obtained except the breadth of the base of second peak.

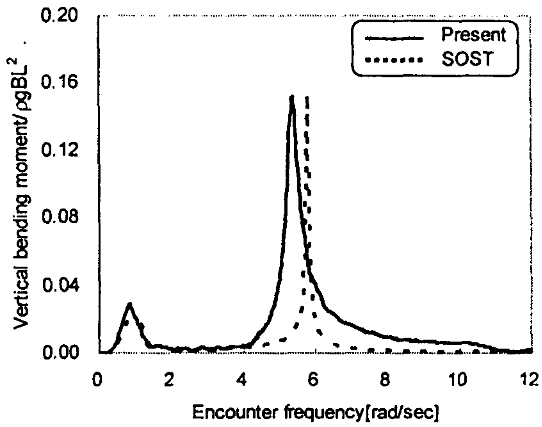

Fig. 2: **Vertical bending moment of ship A (heading angle 135)**

Numerical results for Ship B

Another numerical example for the validation was carried out for the S-175 container ship. Since, the hull form and elastic data are well known from various literatures, there are lots of the numerical and experimental data to compare with. However, existing calculation data and model test data shows only the wave frequency less then 2 rad/sec. Fig. 3 shows calculated vertical bending moments for head sea conditions up to encounter wave frequency 12 rad/sec. We can see that the present calculation agree well with existing experimental data up to available range of wave frequency.

Numerical results for Ship C

Having the usefulness of the suggested numerical model, we proceeded to examine the recent container ship and the results are shown on the Fig. 4.

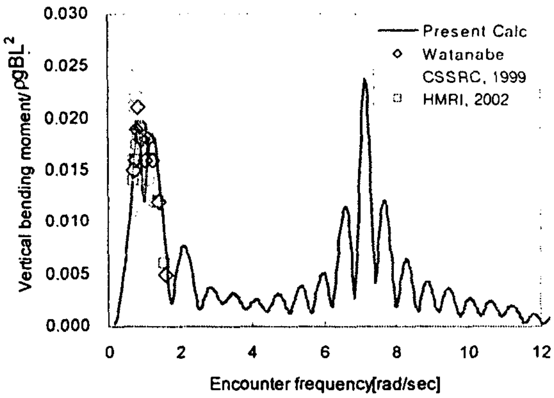

Fig. 3: **Vertical bending moment of ship B**

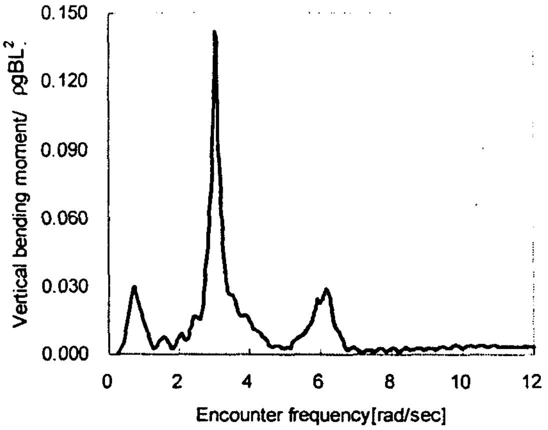

Fig. 4: **Vertical bending moment of ship C (heading angle 135)**

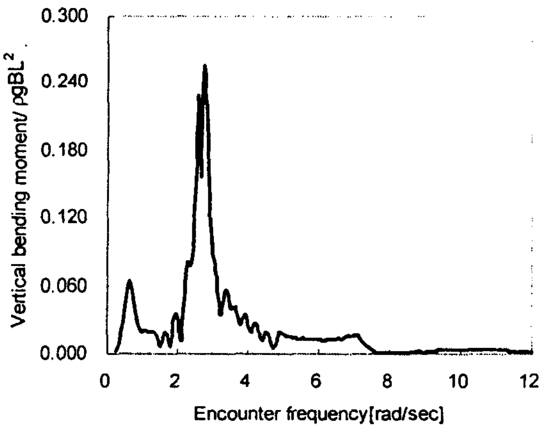

Fig. 5: **Vertical bending moment of ship D (heading angle 135)**

Numerical results for Ship D

Another example of the VLCC was examined. In this case, the hull form has broad beam with shallow draft that high wave load is concerned by the designer. The results are shown on Fig. 5

Numerical results for Ship E

Final example of ore carrier was shown on Fig. 6. Again, we can see two peaks. Springing peak location is around 3.2 rad/sec

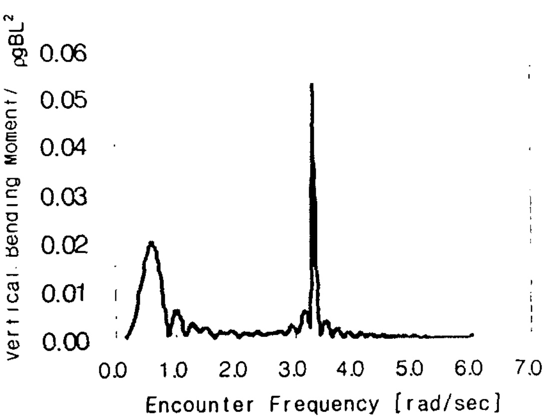

Fig. 6: Vertical bending moment of ship E (heading angle 135)

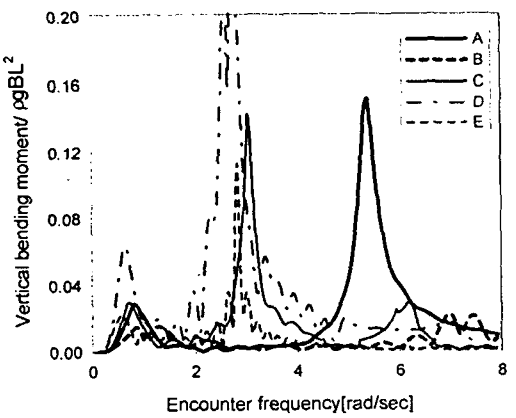

Fig. 7: Vertical bending moment of 5 ships (heading angle 135)

Discussions

We can see the general trend of the vertical bending moments of the suggested five ships as Fig. 7. Ship A and B have rather high natural frequencies around 6-7 rad/sec. Besides, Ship C, D and E have rather low natural frequencies around 3 rad/sec. It means that the modern large merchant ships become more flexible than the older ones. The most important contribution occurs at low frequency range, where the wave length fits the ship length. Within this range the generalized wave forces on the ship take their maximal values which produce tensile stresses of high level. Wave energy is concentrated usually from 0.5 rad/sec to 6 rad/sec with the maximum between 1 rad/sec to 2 rad/sec.

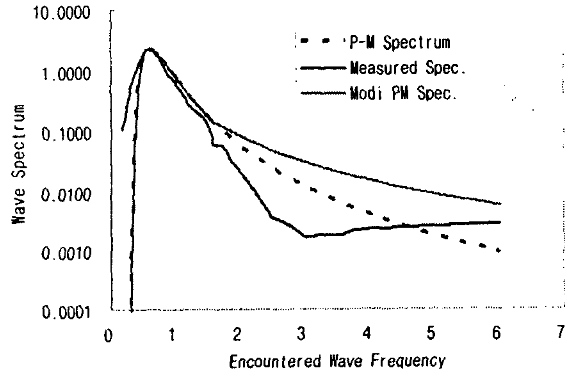

Fig. 8: Theoretical, measured and proposed wave

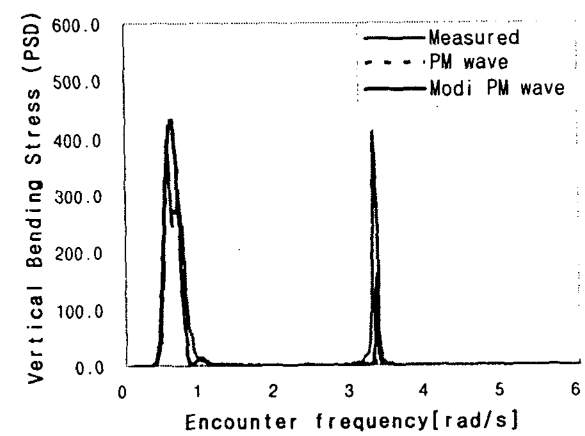

Fig. 9: Vertical bending stress response of ship E (heading angle 180)

If the corresponding encounter frequency is close to an eigenfrequency, the ship response peaks even if the amplitude of the hydrodynamic loads is not very high. The second order effects are small for almost all the frequencies except of that, double of which is close to an eigen-frequency. In this case, the second order ship response gives a considerable contribution to the total response. It was found that springing due to vertical bending and springing due to warping are separated. However, the total contribution of the springing effect to the total damage is small compared to that due to low frequency wave loads.

Fig. 8 shows wave spectrum of ship E. Here, the significant wave height was 5.2 meters and zero crossing period 9.1 seconds with ship speed as 5.5 m/sec. Solid line shows calculation and dotted line shows the measurement of the real ship. We can identify that they show good agreement for wave frequency range. However, there are big discrepancies for the springing maximum value. Storhaug et. al.(2003) dealt with similar topics of measurement results. It is claimed that the spectrum tail should be proportional to ω^{-4} instead of ω^{-5} as commented by Jensen and Dogliani (1996), Phillips (1985) and Holtsmark, Ruijven and Johansen (2002). From the elaborated analysis, we concluded that the wave condition of the North Sea has higher energy at high frequency tail then the theoretical JONSWAP spectrum that the spectrum tail should be proportional to ω^{-3} instead of ω^{-4} or ω^{-5} until reliable wave radar are available at the market.

. Fig 9 shows one of the typical PSD(Power Spectral Density) of the vertical bending stress at midship for ship E. The result of the fatigue life calculation reveals that some of the recent merchant ship has marginal value in fatigue life if serviced in North Atlantic route. For the fatigue life estimation, we followed Kim et al.'s (2002) methods.

Conclusions

From the previous study, we can draw following conclusions.

1) Practical tool to estimate the springing effect is developed.

2) Validation shows reasonable agreement with less computing time.

3) For the ships having low natural frequency and North Atlantic route, special consideration for springing is necessary.

4) High frequency tail of wave spectrum should be increased to ω^{-3} instead of ω^{-4} or ω^{-5} for springing calculation.

5) Further study on hull monitoring and whipping analysis are necessary.

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