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An Estimation of Springing Responses for Recent Ships

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ABSTRACT: *The estimation of springing responses for recent ships is carried out, and application to a ship design is 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, using the superposition method. The quadratic 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 addedasymptotic interpolation along the high frequency range. Several applications were carried out on the following existing ships: The Bishop and Price's container ship, S-175 container ship, large container, VLCC, and ore carrier. One of them is compared with the ship measurement result, while another with that of the model test. The comparison between the analytical solution and the numerical solution for a homogeneous beam-type artificial ship shows good agreement. It is found that most springing energy comesfrom 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 the springing calculation.*

1. INTRODUCTION

Modern ships are being built bigger and lighter with shallow draft and large breadth, which makes the ship flexible, as both her bending and torsional rigidities are 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, resulting in springing, whipping or slamming, etc. Ship springing is the resonant response of the ship to a hydrodynamic excitation, resulting from incident waves. It could occur in relatively moderate sea states, when the encountering wave frequency matches the natural frequency of the 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 the wave amplitude, comes from the sum frequency between the incident waves. Linear strip theory has been widely used for the past few decades, due to its simplicity and surprisingly accurate prediction of ship motions and wave induced loads. However, the effect of flared sections, bottom emergence, and reentry, due to 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 wet surface are satisfied at each time step. However, this calculation requires tremendous computer power and rather sophisticated numerical techniques. Thus, it is not a very practical approach for the early stages of ship design. Instead, we employed the quadratic strip method. We were also concerned that the combined effect of bending and torsion may accelerate the fatigue damage, due to the increased 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 partially rational and partially 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, as can be seen in studies done by 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

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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 it with estimated calculations. Vidic-Perunovic and Jensen (2004) studied the same ship, and derived that the nonlinear interaction between two directional waves can cause substantial energy. Park et. al. (2004) carried out springing responses for modern merchant ships.

Vertical modes of vibrations have been the focus of interest, as 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), despite the lack of substantial 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. Theoretical formulations and numerical computations were performed for predicting hydrodynamic forces on a ship advancing in waves. The quadratic 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, theoretical formulations and numerical computations were performed for the equations of girder, and were solved with the superposition method. The hull was modeled as a Timoshenko beamthat accounts for the rotary inertia, shear deformation, and cross-sectional warping stiffness, in order to analyze the response to the corresponding vibration mode. The developed analysis model is suitable for the evaluation of the springing damages of ships due to waves, including the torsion effect.

Several applications to the existing ships were carried out. The first two are Bishop and Price's container ship and S-175 container ship. The remaining three are commercial ships of container, VLCC, and ore carrier. One is compared with the ship measurement result, while another with that of the model test. Comparison between the 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.

2. SHIP RESPONSE

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

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

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

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

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

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

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, using 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. Next, 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, an analytic solution under the uniform beam assumption is also derived (Park, Jung and Korobkin, 2004)

The quadratic 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, a 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).

3. 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; their principal dimensions are tabulated in Table 1. For validation of the suggested numerical model, ship A and B are examined first. After that, for discussion and application, ship C, D and E are calculated.

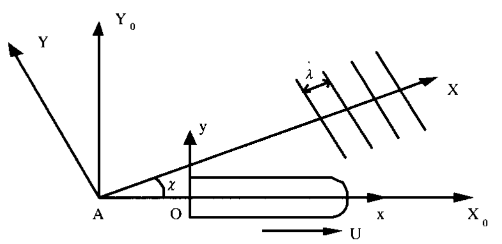

Fig. 1 Local and global coordinate systems

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

3.1 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 ship A are carried out. Wave heading is selected as 135 degrees, and advancing speed is 13.38m/sec. 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 the encounter frequency in rad/sec and the ordinate is normalized vertical bending moment by factor of gBL2. Solid line represents the present calculation, while the dotted line is that of Jensen. We can observe two distinct peaks around 0.8 and 5.2 rad/sec. The first peak comes from the wave, while the 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 for the breadth of the base of the second peak.

3.2 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 literature, there are lots of numerical and experimental data for comparison. 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 an encounter wave frequency of 12 rad/sec. We can see that the present calculations agree well with existing experimental data up to the available range of wave frequency.

3.3 Numerical results for Ship C

Having the usefulness of the suggested numerical model, we proceeded to examine the recent container ship; the results are shown in Fig. 4. Here, we can observe the third peak at around 6 rad/sec, which is considered as the third harmonic, due to elastic beam properties, although this third peak does not occur significantly in other examples.

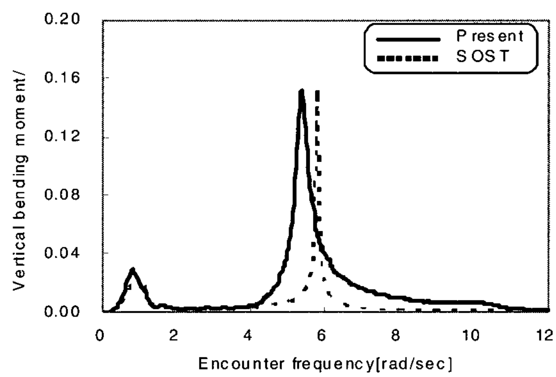

Fig. 2 Vertical bending moment of ship A (heading angle 135)

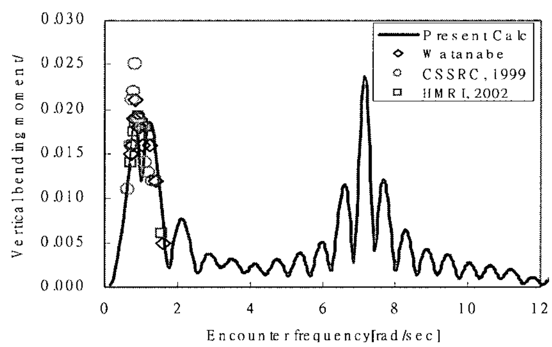

Fig. 3 Vertical bending moment of ship B

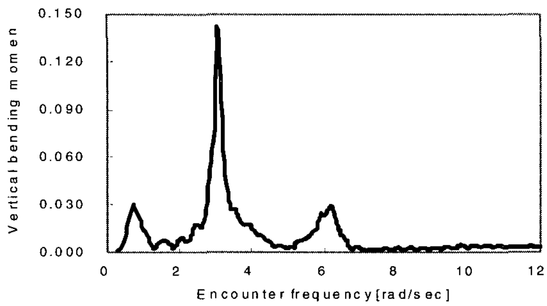

Fig. 4 Vertical bending moment of ship C (heading angle 135)

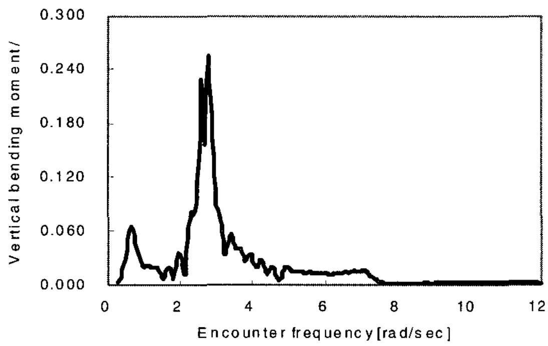

Fig. 5 Vertical bending moment of ship D (heading angle 135)

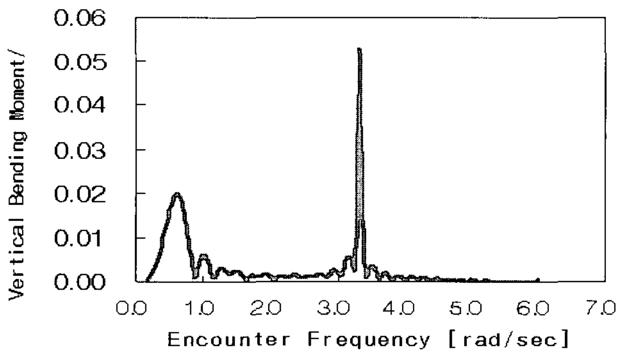

Fig. 6 Vertical bending moment of ship E (heading angle 135)

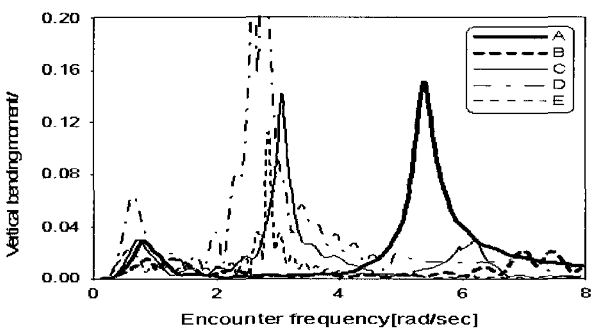

Fig. 7 Vertical bending moment of 5 ships **(heading angle 135)**

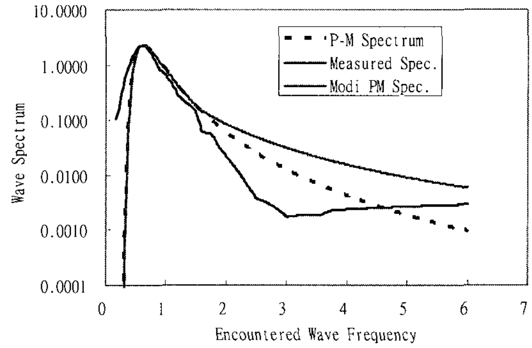

Fig. 8 Theoretical, measured and proposed wave spectrum

3.4 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

3.5 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

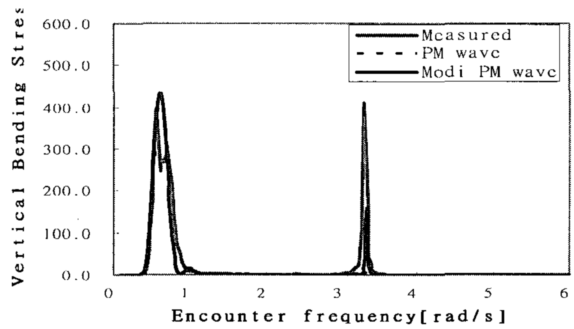

Fig. 9 Vertical bending stress response of ship E (heading angle 180)

4. DISCUSSION

We can see the general trend of the vertical bending moments of the suggested five ships. As shown in Fig. 7, ship A and B have rather high natural frequencies, around 6-7 rad/sec. Ship C, D and E have rather low natural frequencies, around 3 rad/sec. Based on this data, the recent large merchant ships become more flexible than the older ones. The most important contribution occurs at a 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 high-level tensile stresses. 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.

If the corresponding encounter frequency is close to an eigenfrequency, the ship response peaks, even if the amplitude of the hydrodynamic load is not very high. The second order effects are small for almost all the frequencies. 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 was 9.1 seconds, with ship speed of 5.5 m/sec. The solid line shows the calculation, and the 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 than the theoretical JONSWAP spectrum; 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..

5. 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.

REFERENCES

Bishop, R., Price, W.G., Tam, P.K.Y.(1977). "A Unified Dynamic Analysis of Ship Response to Waves", RINA trans Vol 119, pp 363-90.

Bishop, R., and Price, W.G. (1979). "Hydroelasticity of Ships", Cambridge University Press, Cambridge.

Chen, Y.N., and Chiou, J.W. (1981). "Evaluation of Analytical Methods for Predicting Hull Girder Dynamic Response in Waves",American Bureau of Shipping, Technical Report OE-81001.

Goodman, R.A. (1971). "Wave-Excited Main Hull Vibration Tankers and Bulk Carriers", Trans. RINA, Vol 113, pp 167-184.

Hoffman, D., and Van Hoof, R. (1976). "Experimental and Theoretical Evaluation of Springing on a Great Lakes Bulk Carrier", U.S. Coast Guard Report CG-D-74.

Holtsmark, G., Van Ruijven, R., Johansen, O.L. (2002). "Analysis of Springing Measurements of an Ore Carrier", DnV Technical Report No 2002-0765.

Hong, D.C. (1987). "On the Improved Green Integral Equation applied to the Water-wave Radiation-Diffration Problem",J SNAK, Vol 24, No 1, pp 1-8

Jensen, J.J., and Dogliani, M. (1996). "Wave-induced Ship Hull Vibrations in Stochastic Seaways", Marine Structures, Vol 9, pp 353-387.

Jensen, J.J.,and Pedersen, P.T. (1981). "Bending Moments and Shear Forces in Ships Sailing in Irregular Waves," J Ship Research, Vol 25, No 4, pp 543-251.

Jung, J.J., Park, Y.K., Shin, H.S., Park, I.K. and Korobkin, A. (2003). "An Estimation of Hull Girder Response Due to Wave Excitation", Proc ISOPE, Honolulu, USA.

Kim, P.Y., Park, J., Choi B.K. and Kim O.H. (2002). "Fatigue Life Calculation for a Ship Subjected to Hull Girder Vibration at Sea", Proc ISOPE, Kithakyushu, Japan.

Korobkin, A. (2003). "Development of the S/W for Springing Analysis including Torsion Effect", Final Report of LH-02 project, LIH, Russia.

Maeda, H. (1980). "On the Theory of Coupled Ship Motions and Vibrations", The University of Michigan, Department of Naval Architecture and Marine Engineering, Report No232, Ann Arbor, Mich.

Park, I.K., Jung J.J., Korobkin, A. (2004). "Exact Solution on the Vertical Hydro-elastic Responses of Ships having Uniform Sectional Properties", J SNAK, Vol 41, No 2, pp 47-54.

Park I.K., Lee SM, Jung J.J., Yoon M.C. (2004). "Springing Effects on the Design of the Modern Merchant Ships", Proceedings of the 9th International Symposium on Practical Design of Ships and Other Floating Structures, Lubeck-Travemunde

Park I.K., Lee S.M., Jung J.J., Yoon M.C. (2004). "An Estimation of Springing Responses for Recent Ships", 한국해양공학회 추계학술대회 논문집, 서울산업대학교

Phillips, O.M. (1985). "Spectral and Statistical Properties of the Equilibrium Range in Wind-generated Gravity Waves", J Fluid Mech, Vol 156.

Skjordal, S.O., ,and Faltinsen, O.M. (1980). "A Linear Theory of Springing," J Ship Research, Vol 24, No 2, pp 74-84.

Storhaug G., Vidic-Perunovic J., Rudinger F., Holts-mark G., Helmers, JB, Gu X, (2003). "Springing/Whipping Respsonse of a Large Ocean Going vessel - A Comparison Between Numerical Simulations and Full-scale Measurements", International Symposium on Hydro-elasticity for Marine Structures

Van Gunsteren, F.F. (1974). "Some Further Calculations of Wave-Induced Hull Vibrations", Proc Int. Symposium on Dynamics of Marine Vehicles and Structures in Waves, University College London.

Vidic-Perunovic J., Jensen J.J. (2004). "Springing Response due to Directional Wave Field Excitation", Proceedings of the 9th International Symposium on Practical Design of Ships and Other Floating Structures, Lubeck-Travemunde

Watanabe, I., and Soares, C.G. (1999). "Comparative Study on the Time-Domain Analysis of Non-linear Ship Motions and loads", Marine Structures Vol 12, pp153-170.

Wu, J.S. and Ho, C.S. (1987). "Analysis of Wave-induced Horizontal-and-Torsion-Coupled Vibrations of a Ship Hull", J. Ship Research, Vol 31, No, 4, Dec. 1987, pp 235-252.

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2005년 12월 8일 최종 수정본 채택