Numerical Simulation of the Ship Roll Damping

Min Gu, *China Ship Scientific Research Center, Wuxi, China* gumin702@163.com

Jiang Lu, *China Ship Scientific Research Center, Wuxi, China* lujiang1980@aliyun.com

Shuxia Bu, *China Ship Scientific Research Center, Wuxi, China* bushuxia8@163.com

Chengsheng Wu, *China Ship Scientific Research Center, Wuxi, China* cswu@163.com

Gengyao Qiu, *China Ship Scientific Research Center, Wuxi, China* xiaogeng502@163.com

ABSTRACT

The roll damping is a critical hydrodynamic coefficient for predicting roll motion. In this paper, the forced roll motion of a 2-dimensional ship section and free roll motion of a 3-dimensioinal hull are simulated based on the RANS model in calm water. For the forced rolling, firstly, the influences of different calculation parameters are investigated through the methods of orthogonal design and variance analysis. Then the simulations about different roll amplitudes are carried out based on the selected parameters. For the free rolling, the free decay experiments and numerical simulations are performed. These calculated results are agreed well with experimental data, which validate the presented method can yield satisfactory results for roll damping coefficients.

Keywords: *roll damping; RANS; forced rolling; free rolling*

1. INTRODUCTION

The roll damping is a critical hydrodynamic coefficient for predicting roll motion, such as parametric rolling and stability under dead ship condition. The roll damping coefficient should be predicted with high accuracy. The vulnerability criteria are under development by the International Organization (IMO) of the second generation of intact stability criteria, in which the roll damping have been calculated by Ikeda`s (1977, 1978, 1979, 2000, 2004) simplified method. These formulas can be used quite well for the conventional ship, but the predicted results are sometimes conservative or underestimated for unconventional ships (Japan, 2011a, Japan, 2011b, Sweden, 2011). This is because the roll damping is strongly nonlinear, which has some direct relationships with fluid viscosity and flow characteristics, such as the flow separation and vortex shedding. So the experience or semi-experience formulas can`t take the full consideration of different characteristics for different objects. The calculated results of most traditional ships by Ikeda' method can fit experimental data well at the same order magnitude. However, if the size is outside the application range of Ikeda' method, or for the large amplitude roll motion in some phenomena, such as parametric rolling, the accuracy will be low in these conditions, which limit the scope of application of Ikeda' method.

The corresponding group of IMO proposed that the roll damping could be calculated by roll decay / forced roll test or CFD (United States & Japan, 2014). Although the model tests can predict the roll damping very well, but it is costly and time-consuming as well as most of experimental data are limited to a certain frequency range and particular geometry, which is impossible for the large-scale expansion of the application (Blok & Aalbers, 1991, Haddara & Bass, 1988).

The influence of viscosity should be considered during the calculation of roll damping. The CFD numerical simulation can consider different objects and its characteristic, which can also reduce the cost. With the development of CFD technology, the turbulent models have been improved, such as RANS model, discrete vortex method. In addition, the fine structure of the flow field can also be analyzed by CFD, so CFD could be widely used to predict roll damping. Forced roll method and free decay method are two main methods for calculation of the roll damping.

K.B.Salui et al. (2000), Ronald et al. (2002), Miller&Stern (2002), Salui & Vassalos (2003), Frederick Jaouen et al.(2011) simulated forced roll motions for different kinds of ship or two-dimensional ship sections using the RANS model. Wilson et al. (2006) predicted the roll decay of a DTMB Model 5512 hull based on the RANS technique. Miller et al. (2008) conducted roll decay and forced roll simulations using DTMB Model 5415 based on the RANS approach. Sun kyun Lee et al. (2011) performed CFD simulations for the roll damping of a damaged passenger ship by solving RANS equations. These results are in good agreement with the experimental data. The above analysis proved that the roll damping coefficients can be accurately solved using RANS approach.

In this paper, firstly the forced motions of two dimensional ship section of Series 60 based on the orthogonal design and variance analysis are carried out, in which different calculation parameters for roll damping are analyzed. Secondly, the free motions of a three dimensional 4250TEU containership have been simulated. The comparisons between the computed results and the experimental results proved that the roll damping can be predicted by RANS-based method. These can provide technical support for the development of second generation intact stability criteria.

2. FORCED ROLLING

For the forced roll motion, the section of Series 60 is chosen, as experimental tests on its forced roll have been conducted by Ikeda (Ikeda et al. 1977). The same principles are used in the simulations, as shown in table 1. During the calculation, the roll center is located in the intersection between waterline and mid-perpendicular. The formula (1) is used for the roll motion. Then formula (2) is used to get the dynamic moment. Finally, formulas (3) are used to get the roll damping coefficients and non-dimensional coefficients.

Table 1 Principal particulars of S.S.5.

Section	*B*	*T*	*KG*
S.S.5	0.237m	0.096m	0.096m

Where B is the width of model; T is the draught; KG is the vertical height of center of gravity.

$$\phi = \phi_o \sin(\omega t) \tag{1}$$

$$\begin{aligned} M_d(t) &= \iint_s (\tau_y z - \tau_z y) ds \\ &+ \iint_s p_d (n_z y - n_y z) ds \end{aligned} \tag{2}$$

$$B_{44} = \frac{MR}{\omega \phi_0} \Rightarrow \hat{B}_{44} = \frac{B_{44}}{\rho \nabla B^2} \sqrt{\frac{B}{2g}} \tag{3}$$

Where ϕ_0 is the initial roll amplitude, ε is the initial phase, τ is the shear stress on the surface of the hull, p_d is the dynamic pressure on the surface of the hull, M_d is the instant roll moment at the maximum rolling angular velocity, ∇ is the volume for the model.

Table 2 Calculation conditions

Cases	$\hat{\omega}$	ϕ_0 (rad)
1	0.58	0.1
2	0.58	0.15
3	0.58	0.175

The calculation conditions are shown in table 2. Where $\hat{\omega}=\omega\sqrt{B/2g}$, ω is the frequency of rolling. We can see that the non-dimensional frequency ($\hat{\omega}$) is equal to 0.58, and the initial roll amplitudes are 0.1rad, 0.15rad, 0.175rad, 0.22rad respectively.

2.1 Orthogonal design

The simulation results can be affected by different parameters, such as the mesh quantity, mesh quality(y+), turbulent mode, boundary condition and discretization method. In order to find out the best combination of these parameters, we choose the initial roll amplitude ϕ_0 =0.175rad to analyze these influencing factors based on the orthogonal design and variance analysis. According to the previous studies, the values of y+ are always very small during the forced roll motion, especially for ships with bilge keels, so the enhance wall function is used, in which y+ is approximately 1. The discretization method is SIMPLE which has a wide application. Finally, we focus our attentions on the following factors: mesh quantity, turbulent model and boundary condition.

The ship section is 16m, so we chose a circular section as the calculating domain, whose diameter is approximately 12.5 times of the model`s width (D ≈ 12.5B), and the boundary conditions including 3 parts: (1) the upper boundary of the circular domain; (2) the bottom boundary of circular domain; (3) the section surface, as shown in figure 1.

For the part of mesh quantity, we choose 10 thousands mesh as a benchmark. Three different kinds of mesh quantities are 10 thousands, 20 thousands and 40 thousands based on the geometric proportion increasing and decreasing design, as shown in figure 2. The selection of turbulent model should consider the practicality and efficiency. In this paper, we studied standard k-ω model (s k-ω), SST k-ω and RNG k-ε. The boundary conditions are all walls, all velocity-inlet, the bottom boundary of circular domains wall and the upper boundary of circular domains pressure-outlet, respectively.

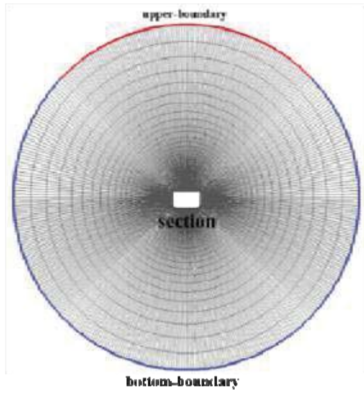

Fig.1 The boundary conditions

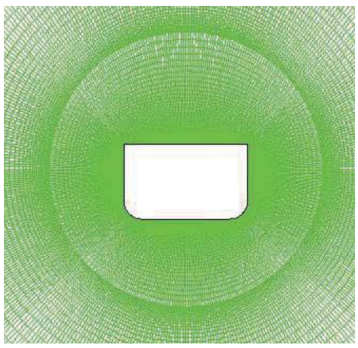

Fig.2 The part of the calculating domain (mesh quantity=40 thousands)

According to above analysis, we can get the table of factors and levels, as shown in table 3. The orthogonal layout and the two columns interaction layout $L_{27}(3^{13})$ are selected after considering the columns and degrees (Wei & Wu, 2013), and the layout is shown in table 4, in which the 9,10,12,13 are blank columns (error columns).

Table 3 Factors and levels

Level	Factor		
	A:Mesh quantity	B:Mesh quality	C:Boundary condition
1	A1: 10t	B1:sst k-ω	C1: 2 walls
2	A2: 20t	B2: s k-ω	C2: 2 vel
3	A3: 40t	B3:RNG k-ε	C3: 1wall+1pre

Table 4 Top design of the calculation program

Factor	A	B	$(A\times B)_1$	$(A\times B)_2$	C	$(A\times C)_1$	$(A\times C)_2$	$(B\times C)_1$			$(B\times C)_2$		
Num	1	2	3	4	5	6	7	8	9	10	11	12	13

2.2 Variance analysis

The numerical simulations for the combination of different parameters have been conducted based on RANS model. The VOF method is used for the free surface modeling. The pressure-correction algorithm of SIMPLE type is used for the pressure-velocity coupling. The modified HRIC is used for the discretization of VOF equation, and the dynamic mesh technique is used by UDF. The non-dimensional roll damping coefficients can be got by formula (3). We selected several y+ values from two calculating cases, and the results show that the enhanced wall function was appropriate, as shown in figure 3.

The non-dimensional coefficients were got for different computational schemes. Then the significance of the test was investigated through the table of variance analysis, as shown in table 5. During the variance analysis, the relative errors between simulation results and experimental results were adopted as the analyze benchmark.

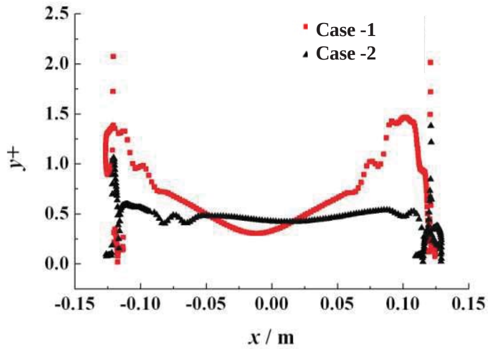

Fig. 3 The value of y+

Table 5 Variance analysis

Soruce of variation	Quadratic sum-S	Degree of freedom-f	Mean square-V	F	Significance	F_a
A	0.61	2	0.30	10.64	**	$F_{0.05}(2,12)=3.89$
B	0.47	2	0.23	8.18	**	
C	0.34	2	0.17	5.86	*	$F_{0.01}(2,12)=6.93$
A×B	0.52	4	0.13	4.52	*	
A×C	0.17	4	0.04	1.45		$F_{0.05}(4,12)=3.26$
B×C	2.12	4	0.53	18.50	**	
e	0.61	2	0.30			$F_{0.01}(4,12)=5.41$

The values of *F* showed that the factor B×C (the interaction between turbulent model and boundary condition), the factor A (mesh quantity) and the factor B (turbulent model) have large influence on the results. The factor A×B (the interaction between mesh quantity and turbulent model) and the factor C (boundary condition) also have effects on the results, but the effects are not obvious compared with the above three factors.

The collocation table of B and C was listed to seek the best combination, as shown in table 6. The results showed that the combination of B1and C1, B1 and C2 were both available. However, we find that the combination of B1 and C1 was easier to convergence and the computational process was more stable during the calculation, so we choose the combination of B1 and C1 as the best combination.

Table 6 The match of B and C

	B1	B2	B3
C1	0.33	1.76	1.23
C2	1.35	2.79	2.26
C3	0.25	1.69	1.16

Statistical hypothesis: the influence of controlled and control factors on results have no significant difference.

This hypothesis can be proved by formula (4). The results showed that the factors are significant differences, which meanings other factors which have not been taken into consideration have little effect on the results during our numerical simulation. Therefore the appropriate turbulent model and boundary condition as well as the mesh quantity can get good results on forced roll simulations. We should note that the enhanced wall function is adopted during the calculation. Otherwise, the results were not consistent with the actual situation. This means the mesh quality (y+) has the most important effect on the results. The current results can only be adopted on the premise of the guarantee of y+.

$$F = \frac{\sum S / \sum f}{S_e / f_e} = 8.18 > \begin{cases} F_{0.01}(18,8) = 5.41 \\ F_{0.05}(18,8) = 3.17 \end{cases} \quad (4)$$

From the above analysis we see that: on the guarantee of y+, the design of A2 (40 thousands mesh), B1 (SST k-ω), C1 (all boundary conditions are walls) is the best combination.

2.3 The calculation results and analysis

Based on the above combination, more research about other conditions were conducted, and the results are shown in figure 4. This figure shows a comparison between the numerical simulation results and experimental results, we can see that the results are in good accordance with the experimental results, so the combination is feasible.

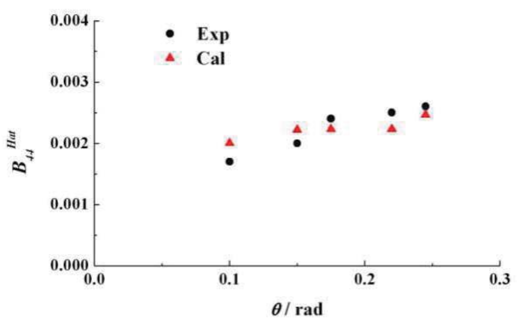

Fig.4 The non-dimensional damping coefficients for different roll amplitudes

3. THE FREE ROLLING

For the free roll decay motion, the object is a 4250TEU containership due to the availability of experimental data for validation. The free roll decay simulations were performed based on the unsteady RANS model and compared to experimental data.

3.1 Experiment

The principal particulars and body plan of this containership are shown in table 7 and figure 5, respectively. Roll decay experiments were performed with a 1/62.97 scaled model at the seakeeping basin (length: 69m, breadth: 46m, height: 4m) of CSSRC (China Ship Scientific Research Center), as shown in figure 6. The initial roll angle was 25 degrees in calm water.

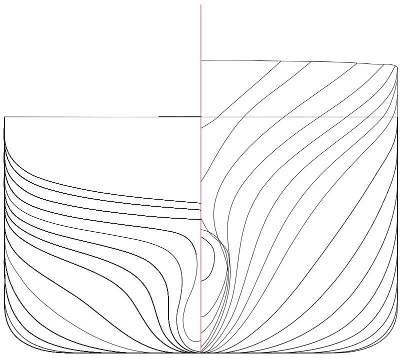

Fig. 5 Lines of 4250TEU containership

Table 7 Principal particulars of the 4250TEU Containership

Items	Ship	Model
Length: *L*	251.88m	4.0m
Draft: *T*	12.6m	0.2m
Breadth: *B*	32.2m	0.511m
Depth: *D*	19.3m	0.3065m
GM	1.62m	0.0257m
T_{φ}	21.19s	2.7s
K_{yy}	0.3*L*	0.3*L*

Fig.6 The ship model in free decay test

3.2 Simulation

In this paper, the simulations of roll decay at 25 degrees initial roll angle in calm water were performed. During the simulation, the VOF method is used for the free surface modeling. A pressure-correction algorithm of SIMPLE type is used for the pressure-velocity coupling. The SST k-ω model is incorporated for turbulence modeling. The solution domain is formed in two parts: the first part (S1) moves with the body, and the second part (S2) is fixed, as shown in figure 7. For the purpose of wave absorption, two artificial damping zones were located at the second part (S2), which is far away from the hull.

3.3 Comparison

The results of numerical simulations of roll decay histories were compared with the experimental results, as shown in figure 8. It shows that the period agrees well with the experimental data with the growth of the time. However, the amplitude of CFD becomes a little larger than the experiment. The future calculations are needed to verify these phenomena.

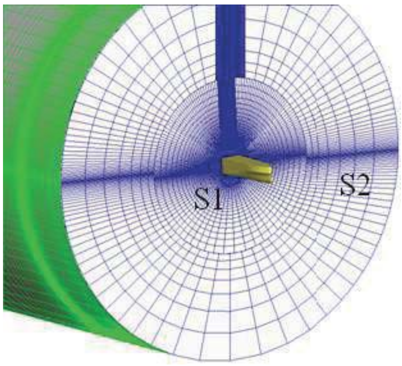

Fig. 7 The solution domain in free decay

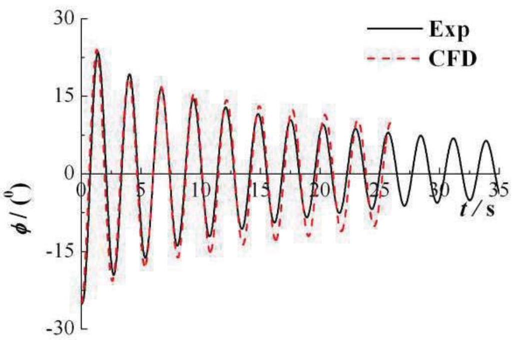

Fig. 8 The comparison of experimental results and numerical simulation of free decay

4. CONCLUSIONS

As a result of experimental and numerical study on roll damping by the forced rolling with two dimensional ship sections of Series 60 and by the free rolling with a 3-dimensioinal hull based on the RANS model, the following remarks are noted:

1) For the forced motion, an applicable results of roll damping can be got based on the combination of enhance wall function, SST k-ω model, the wall boundary conditions as well as the appropriate mesh quantity.

2) For the free roll motion, the roll motion of a 3-dimensioinal hull based on the RANS model in calm water was simulated, and the results were in reasonable agreement with the experimental results.

3) Both the forced rolling and free rolling based on RANS approach have the abilities to predict the roll damping.

4) More works should be made in future to improve calculating accuracy of roll damping, especially for free roll motion condition.

5. ACKNOWLEDGEMENTS

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