Roll Damping Assessment of Intact and Damaged Ship by CFD and EFD Methods

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ABSTRACT

This paper presents an assessment of the roll damping of DTMB 5415 naval ship model in both intact and two compartments symmetric damaged scenarios. An experimental assessment of roll decay is performed at zero speed at different initial heel angles at the University of Strathclyde, Glasgow. Reported experimental results are decay curves, natural frequency and period of roll for intact and damaged ship. CFD calculations are performed by *CDAdapco StarCCM+* software investigating the accuracy and efficiency of the numerical approach. In the numerical procedure the sensitivity analysis on mesh refinement for damaged ship was performed. Furthermore, a sensitivity analysis on time step and turbulence models was performed for the intact ship. Numerical results are plotted against experimental to verify the precision of the numerical simulations. Obtained numerical results are shown to be reasonably accurate although the calculation time still precludes the use of CFD analysis as a standard design procedure.

Keywords: *DTMB 5415 navy ship, intact ship, damaged ship, CFD, EFD, roll decay*

1. INTRODUCTION

Although most vessel responses can be calculated with acceptable accuracy by potential theories in the frequency domain, this is more difficult for roll response due to the viscous damping effects which are not negligible in roll. Roll damping plays an important role in the vessel seakeeping, which is the basis for the precise prediction of vessel motions in waves. The most common approach adopted is based on the Ikeda (1976) empirical method in which the equivalent total damping coefficient is calculated as a sum of potential, friction, eddy-making, appendages and lift contributions. The roll damping coefficient can be also be obtained through a ship model roll decay tank test but there is evident lack of this approach in typical design procedures.

Very recently use of CFD methods in calculating roll damping has become possible due to developments in computing power. Numerical simulation based on CFD offers the advantage of considering viscous flow, although calculations are still very time consuming and experience of the modeling of this phenomenon is still very limited. A major problem in roll decay simulation, common to any problem of transient ship motion, is the necessity of special computational techniques such as deforming mesh, moving mesh and grid interface.

One of the first CFD assessments of roll decay is by Wilson (2006) who performed simulations for a bare hull and bilge-keel-appended surface combatant model (referred as DTMB 5512) using the software *CFDShip-IOWA*. Roll decay simulations are performed for three cases: the bare hull at Fr = 0.138 and 0.28 and the hull with bilge keels at Fr = 0.138. Comparisons of EFD and CFD damping coefficients for the low speed case with bilge keels showed very small differences, generally less than 0.4%, while comparisons for the bare hull cases at both speeds showed larger differences for damping coefficients (up to 20%) even though the difference in time histories for the roll motion showed reasonable agreement (<4.5%).

Yang *et al.* (2012) presented simulation performed using the commercial software package *Fluent* of roll decay for the same vessel, DTMB 5512, with initial heel angles: 5, 10 and 15 degrees at Fr = 0.28. The authors reported very good results in terms of damping coefficient and two examples of decay curve but no details on the method and calculation procedure are given. Yang *et al.* (2013) performed numerical simulations of free decay and forced rolling at various forward speeds and amplitudes for DTMB 5512 and S60 hulls to predict ship roll damping, using a RANS solver using a dynamic mesh technique. The influences of forward speed, roll amplitude and frequency on the ship roll damping are evaluated. The authors report the difference between numerical and experimental results as 1.3 to 2.5%.

Handschel *et al.* (2012) applied RANS simulations to calculate roll damping coefficients of a RoPax vessel in full scale. The influence of the roll amplitude up to 35 degrees, three ship speeds, the vertical position of the roll axis, and the interaction between the bilge keels and the ship hull are analysed. Detailed validation data for a RoPax ship was not available but authors compared the numerical results with Ikeda’s method. Avalos et al. (2014) investigated a roll decay test of the middle section of an FPSO with bilge keels by the numerical solution of the incompressible two-dimensional Navier–Stokes equations. The simulations indicated the strong influence of the bilge radius on the damping coefficient of the FPSO section. Very good results were generally obtained for cases with bilge keels, although sometimes the agreement for the oscillation period was not so good in the case with the larger bilge keel. The worst results in terms of damping and oscillation period were obtained for the section without bilge keels. The authors highlighted that the numerical simulation confirmed the occurrence of the so-called damping coefficient saturation: i.e. the phenomenon in which the damping coefficient does not increase with amplitude as predicted by conventional quadratic theory.

Gao & Vassalos (2011) presented results of numerical simulations of roll decay of DTMB 5415 with bilge keel in both intact and damage conditions by RANS. The comparison shows that the agreements between calculation and model test are acceptable with slightly larger period and smaller damping obtained from the calculation. Gao *et al.* (2013) presented an integrated numerical method that couples a seakeeping solver based on the potential flow theory and a Navier–Stokes (NS) solver with the volume of fluid (VOF), developed to study the behaviour of a damaged ship in beam seas. The integrated method was used to simulate the roll decay of a damaged Ro–Ro ferry and the ferry’s motion in regular beam seas. Validation against experimental data showed that the proposed method can yield satisfactory results with acceptable computational costs.

This work continues the stream of investigation on the applicability of CFD methods for roll damping determination. The commercial software *CD Adapco StarCCM+* is used for roll decay simulation of an intact and damaged bare hull DTMB 5415 model, tested by authors at the University of Strathclyde.

2. MODEL DTMB 5415 GEOMETRY AND DATA

2.1 DTMB 5415

Roll damping was studied for the well-known benchmark naval hull form DTMB 5415, constructed in fibreglass as 1/51 scale model used in experimental campaign in Begovic et al. (2013). The main particulars of the DTMB 5415 model are given in Table 1.

Table 1. Main Particulars DTMB 5415

Particulars	Ship	Model 51
L_{OA} (m)	153.300	3.0
L_{PP} (m)	142.200	2.788
B_{WL} (m)	19.082	0.374
B_{OA} (m)	20.540	0.403
D (m)	12.470	0.244
T (m)	6.150	0.120
V(m^3)	8424.4	0.0635
Δ (t, kg)	8635	63.5
C_B	0.505	0.505
C_P	0.616	0.616
C_M	0.815	0.815
KM (m)	9.493	0.186
KG (m)	7.555	0.148
GM (m)	1.938	0.038
LCG (m)	70.137	1.375
$k_{xx\text{-}WATER}$ (m)	6.932	0.136
$k_{yy\text{-}AIR}$ (m)	36.802	0.696
$k_{zz\text{-}AIR}$ (m)	36.802	0.696

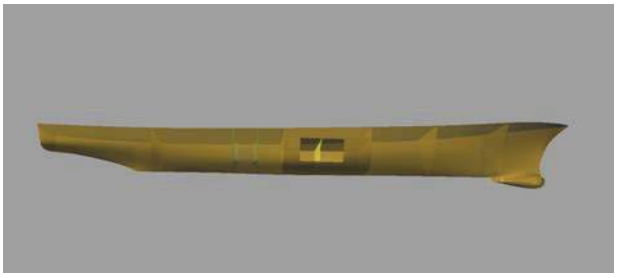

Figure 1 DTMB 5415

The internal geometry of the 1:51 model was identical to that presented by Lee *et al.* (2012). The model has been fitted with the 5 watertight bulkheads located as shown in Figure 1. The damage opening shown in Fig. 2 leads to two compartment (3 and 4) symmetric flooding. The flooded length extended from $x_1 = 65.66$ m (ship scale) to $x_2 = 90.02$ m, corresponding to 17% of the length between perpendiculars. This extension seemed reasonable for a destroyer type of ship, as it is expected that this type of ships have to preserve all functionality with two compartments damage. Both compartments were fitted with the small tube to assure the air-flow during tests, visible on the port side of model at Fig.2.

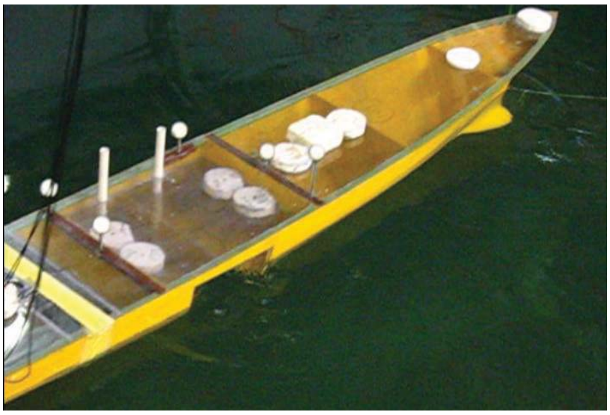

Figure 2 Damage opening of DTMB 5415

The exact amount of flooded water is determined from hydrostatic calculations, i.e. for the measured immersion and trim angle, the displaced volume was found. All characteristics of damaged ship are reported in Table 2.

Table 2. Damaged case principal characteristics

Particulars	SHIP	MODEL
$L_{\text{flooded compartments}}$ (m)	24.36	0.478
B_{WL} (m)	19.458	0.382
T_{mean} (m)	7.41	0.145
Trim [+ aft] (deg)	-0.656	-0.656
Δ (t)	11273.8	0.083
Mass of flooded water (t/kg)	2638.9	0.019
LCG (m)	71.622	1.404
KM (m)	9.427	0.185
KG (m)	6.654	0.130
GM (m)	2.773	0.054

2.2 Experimental results for intact ship

The tests have been performed at the Kelvin Hydrodynamics Lab, University of Strathclyde. The model motion has been tracked using a Qualisys optical system at frequency of 137.36 Hz. In Figure 3, four decay cases are reported for the bare hull intact ship, with different initial heel angles of: 4.00, 13.43, 19.38 and 28.00 deg.

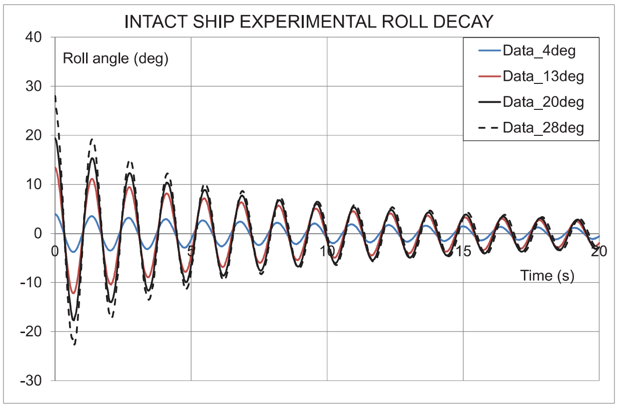

Figure 3 Roll decays of intact DTMB 5415

Results of simple analysis of roll damping coefficient for all tested decays according to ITTC (2011) nomenclature and standard logarithmic decay are natural period and damping coefficients: linear α and quadratic β, reported in Table 3. The trends of measured decays reported in Fig.3 indicate very small damping for small initial heel: in 15 roll cycles the roll amplitude decreased from initial 4.0 deg heel to 1.1 degree. It can be further noted that the 20 and 13 deg decay curves converge for amplitudes lower than 5 degrees indicating that the roll damping mechanism at large amplitude heel angles is different to that at small angles and that the damping formulations proposed by Fernandes & Oliveira (2009) and Bessler (2010) are suitable for both small and large angles.

2.3 Experimental results for damaged ship

For the damaged ship only cases with initial amplitudes higher than 10 deg have been considered due to the much higher damping of the damaged ship with respect to the intact case. Two cases with initial amplitudes of 13.5 and 19.1 deg are given in Figure 4. It can be noted that in 10 cycles the roll amplitude is reduced to 1deg. It can be further noted from Figs. 3 and 4 that the natural period of the damaged ship (1.518s) is significantly higher than that of the intact ship (1.368s).

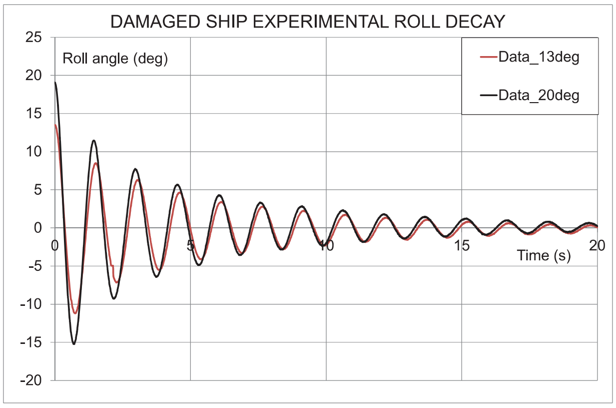

Figure 4 Roll decays of damaged DTMB 5415

Table 3. Roll decay analysis summary

	Intact	**Damaged**
ω_4 (1/rad)	4.593	4.135
T_4 (s)	1.368	1.518
α (1/s)	0.0604	0.1358
β (1/rad)	0.1237	0.2628

From Table 3 it can be seen that the damaged ship exhibits a higher natural roll period as well as much higher linear and quadratic damping coefficients α and β than those for the intact ship. This difference is mainly due to the flood water dynamics, inside and outside the compartment, generating some waves and some vortices. It can be noted that both linear and quadratic damping coefficients have increased by more than double.

3. NUMERICAL SET UP

In this work the commercial software *CD Adapco StarCCM+ V.8.04*. has been used for the calculations of roll decay curves. It is well known that the accuracy of CFD results and the calculation time strongly depends on the type of the mesh and number of cells used, and therefore meshing is optimized for the "most challenging" case, i.e. damaged ship with 19.1 deg initial heel. In present work, a moving mesh and grid interface have been used for modelling the roll decay phenomenon. For the interaction between the moving body and the free surface a Chimera grid or overset mesh technique is used. To solve the time-marching equations, an implicit solver has been used to find the field of all hydrodynamic unknown quantities, in conjunction with an iterative solver to solve each time step. The software uses a *Semi Implicit Method for Pressure Linked Equations* to conjugate pressure field and velocity field, and an *Algebraic Multi-Grid* solver to accelerate the convergence of the solution.
The free surface is modelled with the two phase volume of fluid technique (*VoF*). A segregated flow solver approach is used for all simulations. The Reynolds stress problem is solved by means of k-ε turbulence model.

3.1 Mesh generation and sensitivity analysis

A trimmed mesh of hexahedral type is used, shown in Fig. 5. In order to optimize the discretization of each region and to avoid large computational costs, the region around the hull is finer than the far field regions.

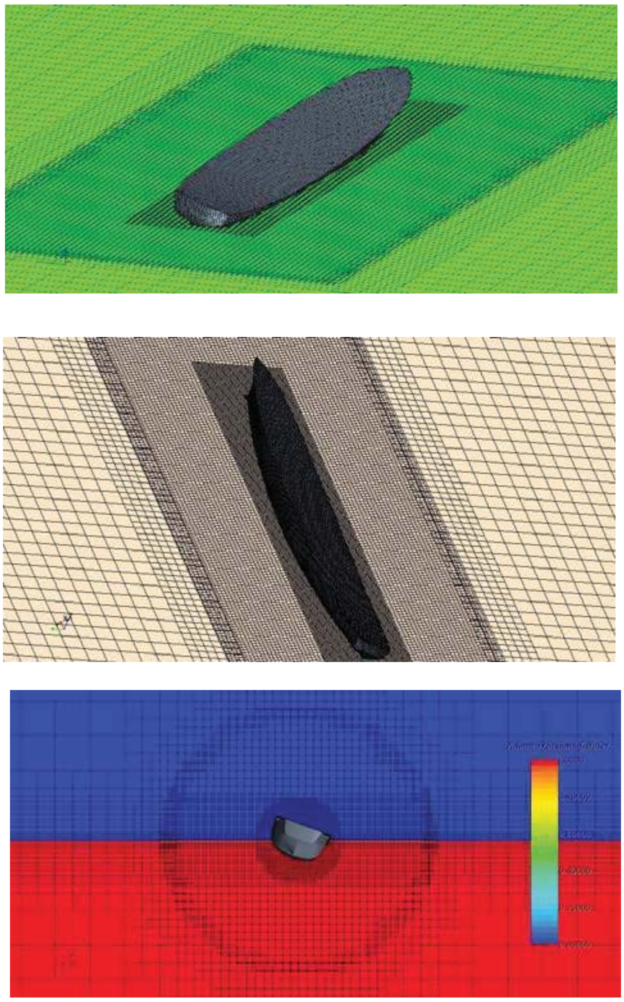

Figure 5 Hexahedral trimmed mesh

The mesh shown in Fig.5 is the result of the sensitivity analysis performed with two trimmed meshes and two hybrid meshes (polyhedral and trimmed) running 5 seconds of model roll decay simulation. A summary of cell numbers and CPU time for 32 processors is given in Table 4. The obtained roll decay histories are shown in Fig.6 indicating that the Hybrid_1 mesh gives completely incorrect results, and it was thus stopped after 3 seconds. It can be noted how the refinement of the free surface VoF (Hybrid_1 vs. all others) in the range of the complete hull model height (not only the "seakeeping" free surface) yields significant improvement in roll decay simulation. From Fig.6 very small difference can be noted between Trim_2 and Hybrid_2 meshes in quality of results while the computational time is extremely prohibitive for the Hybrid_2 case.

Table 4. Mesh sensitivity analysis summary

Grid Type	No. Cells	CPU Time
	$*10^6$	(h)
Hybrid_1	1.194	90
Trim_1	0.709	40
Trim_2	1.476	90
Hybrid_2	2.590	192

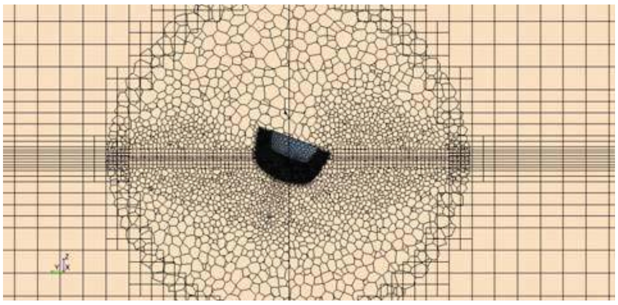

Figure 5a Mesh Hybrid_1

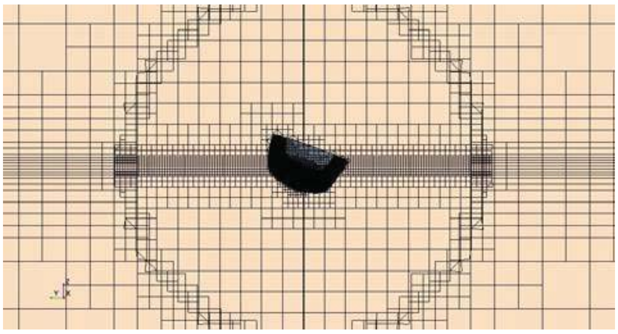

Figure 5b Mesh Trim_1

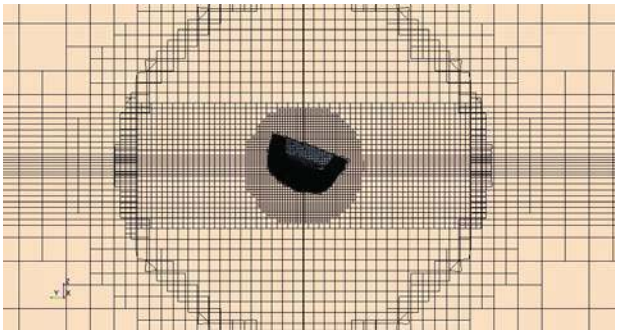

Figure 5c Mesh Trim_2

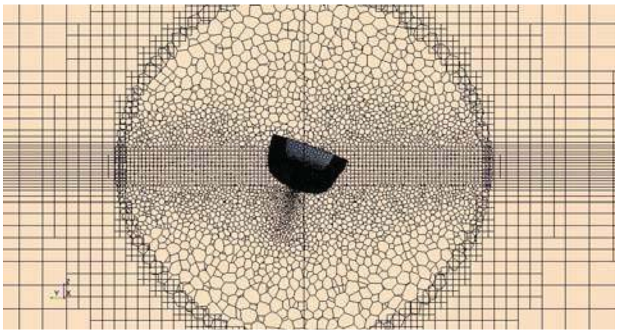

Figure 5d Mesh Hybrid_2

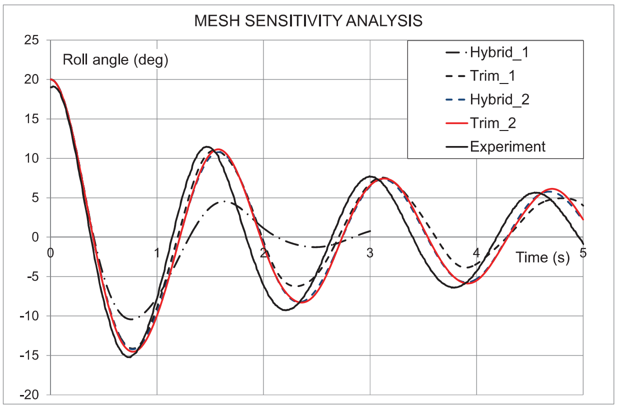

Figure 6. Mesh sensitivity results

3.2 Boundary Conditions and solver settings

All the boundaries, as defined in the numerical set up, are shown in Fig. 7.

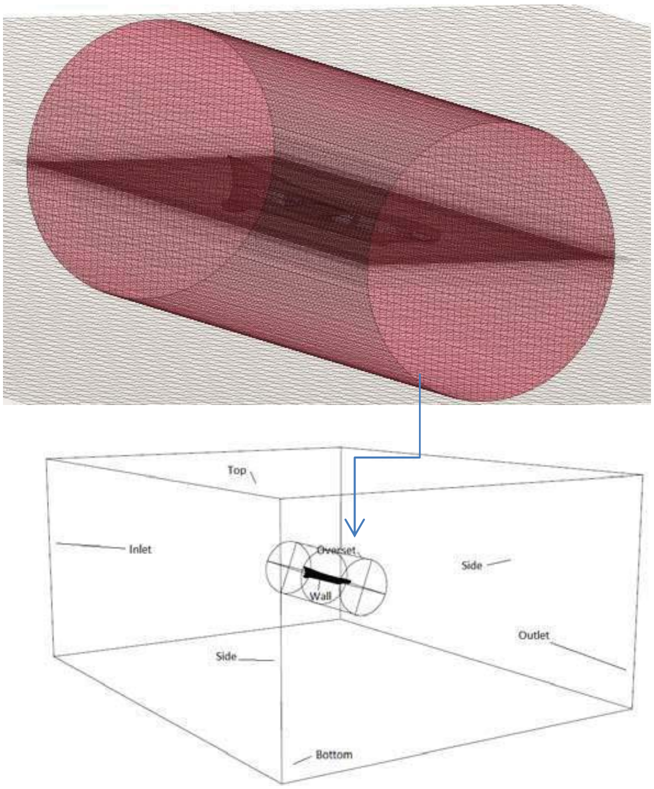

Figure 7 Domain and Boundary representation

The conditions applied to each of them are summarised in Table 5. For each simulation, the hull is heeled at the initial angle of the roll decay curve. The origin of the coordinate system is at the model CG. For the intact case the calculations have been performed with k-ε and k-ω turbulence models. All properties of the numerical solver are reported in Table 6.

Table 5 Boundary conditions summary

Inlet	Velocity inlet condition
Outlet	Velocity inlet condition
Bottom/Top	Velocity inlet condition
Sides	Pressure outlet
Hull	Wall with no-slip condition
Symmetry plane	Not existing
Overset	Boundary Interface

Once all the boundary conditions have been imposed, the last step is defining the numerical set up. The ITTC "Practical Guidelines for Ship CFD Applications" recommendation for time step choice for periodic phenomena such as roll decay and vortex shedding is at least 1/100 of phenomenon period. The measured roll period varies from 1.37 to 1.52 seconds resulting in recommended minimum of 0.015s. Sensitivity analysis has been performed for time steps equal to 0.002s and 0.001s. The simulations have been performed for intact ship at 19.43 deg initial heel and results are given in Fig. 8.

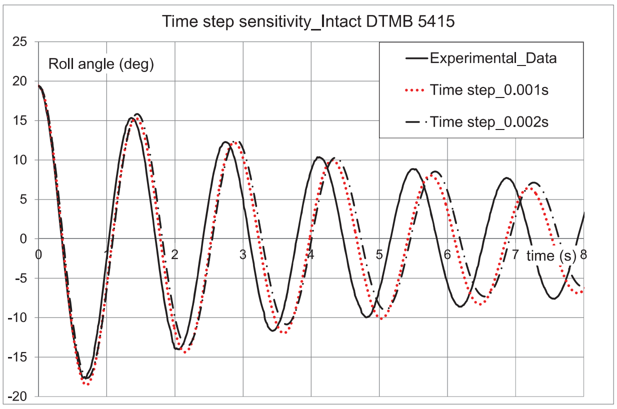

Figure 8 Time step sensitivity

Although the initial step of 0.002s is one order of magnitude lower that ITTC recommended time step, it can be seen that the simulation results is not stable with this time step. Both: decay curve and roll period are improved in simulation with 0.001s time step. Trying lower time step has been considered too expensive in terms of calculations costs.

Results of simulations with k-ε and k-ω turbulence models are given in Fig. 9. Numerical results are within 1% difference although it is not possible to appreciate the difference between two numerical curves.

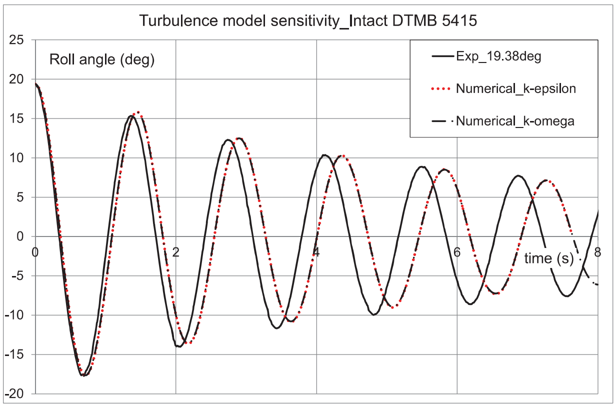

Figure 9 Turbulence models sensitivity

Final numerical set up used for the simulations is reported in Table 6.

Table 6 Solver settings summary

Convection Term	2nd order
Temporal Discretization	2nd order
Time-step (s)	0.001
Iteration per time step	12
Turbulence Model	k-ε / k-ω

4. NUMERICAL RESULTS

4.1 Intact ship

The final simulations for the intact ship have been performed for 4.00 and 28.00 degrees initial heel. The larger angle represents a limit for mesh functionality. The lower angle gives the part of extinction curve common to all experimental decays reported in Fig. 3, where none of the simulations arrived due to the necessary computing time. The mesh scene is given in Fig. 10 for both simulations. The total number of cells is 1.24M. The calculation time depends on the turbulence model and the initial heel angle; for the k-ε model using 32 processors, 1s of simulation takes about 13 hours for 4.00 deg and about 8 hours for 28.00 deg initial heel.

Results compared with the experimental data are given in Figs. 11 and 12.

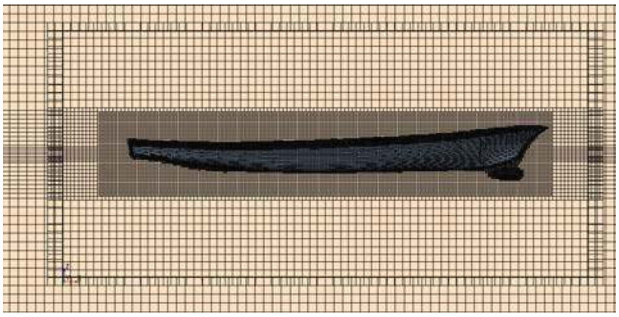

Figure 10 Mesh Scene for Intact model

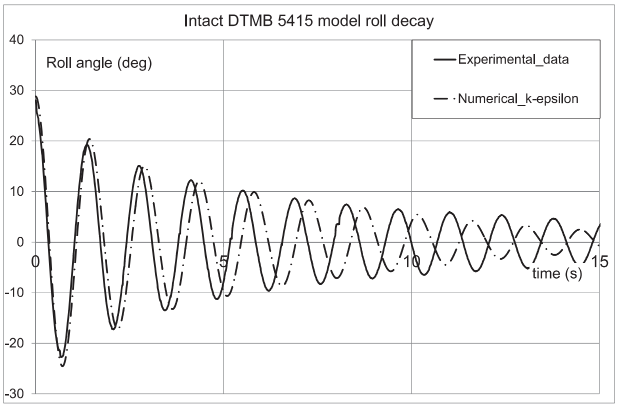

Figure 11 Comparison of experimental and numerical results

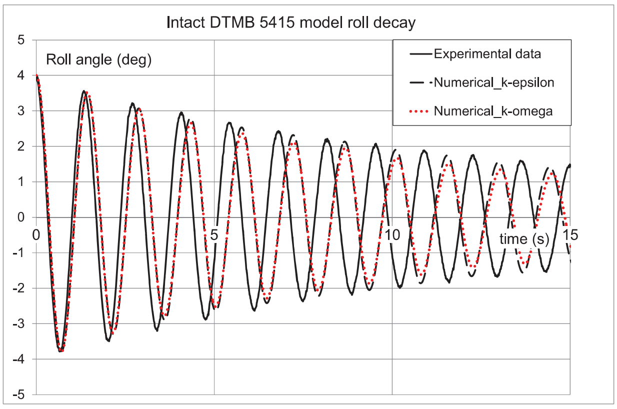

Figure 12 Comparison of numerical and experimental results

In both simulations a good trend of magnitude of decay curves with higher roll period can be observed. Roll oscillation period in all simulations is 1.443 seconds, and does not show dependence on roll angle. With respect to experimental result of 1.369s, this gives a difference of 5.4%.

4.2 Damaged ship

The final simulation for the damaged ship is performed for 15 seconds model time. Details of the mesh in the flooded compartments is shown in Fig. 12. The numerical roll decay curve compared with the experimental data for the damaged ship is given in Fig. 13.

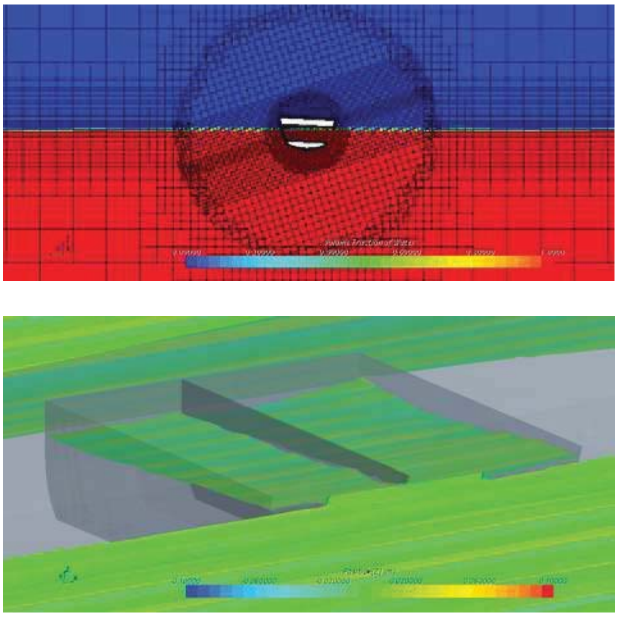

Figure 12 Damage detail

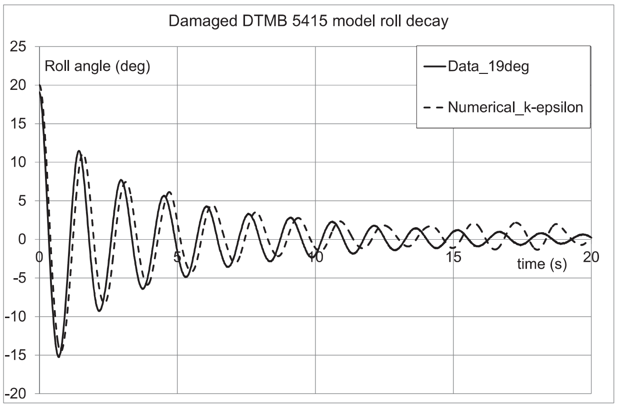

Figure 13 Numerical vs Experimental Roll Decay

It can be seen that oscillation period of numerical results 1.56 s is longer than of experimental, 1.518s, leading to the difference of 2.8%.

4.3 Damping coefficients comparison

Assessment of damping coefficients in experimental procedure generally is done analysing more than five decay curves. Results presented in Table 4 are calculated for 10 decays, including large and small initial angles. Due to required CPU time, it is not possible to use the same number of decays within numerical procedure; therefore the comparison of damping coefficients is done for two numerical cases vs. respective experimental results. Decay coefficient analysis is given in Fig. 14.

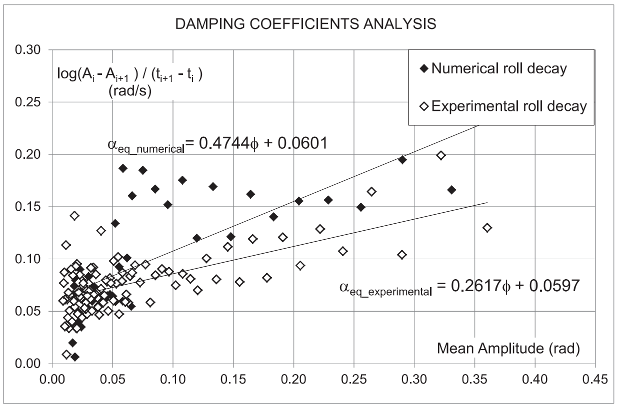

Figure 14 Damping coefficient determination

The linear coefficient α obtained for numerical and experimental results can be considered equal (0.0601 vs. 0.0597). The quadratic coefficient β is obtained by multiplying the angular coefficient of trend line by $0.75 \cdot \pi / \omega$. The values obtained are 0.243 and 0.124 for numerical and experimental results, respectively. Looking at the numerical data in Figs. 11 and 14, two problems for simulation at very high initial heel are evident. The first one is the higher predicted damping, which depends upon calculation settings (mesh, time step, solver, etc). The second problem, which presents a serious challenge, is that the time required for simulation to arrive at small angles is too long and without this part of the extinction curve the damping coefficient prediction will not be realistic.

5. CONCLUSIONS

This work focuses on the use of commercial software *CD Adapco StarCCM+* RANS solver for the analysis of roll damping properties of the bare hull naval ship DTMB 5415.

Roll damping is considered through the roll decay curve prediction, which is the beginning for any further analysis of roll damping coefficients and it is directly compared with the decay curves obtained from experiments performed by the authors. Experimental results concern intact and damaged ship behaviour in free roll decay starting from different angles ranging from 4 to 28 degrees and damaged ship data can be added to Gothenburg CFD workshop (2010).

Mesh sensitivity in numerical simulations is optimised for the damaged ship case considering hexahedral trimmed and hybrid meshes with different refinements, sizes and shapes. The trimmed mesh is chosen as it has the same accuracy of fine hybrid but significantly lower computational time. Obtained numerical results have reasonable damping coefficient prediction but the period of oscillations differ from experiments by up to 4%. These results are in line with those presented by Gao (2011, 2013) and Avalos (2014). It has to be commented that numerical predictions are highly determined by the quality rather than the quantity of the mesh

The serious challenge for the use of CFD method for damping prediction lies in the extremely high computational time required. Without considering the time necessary for the mesh generation, the calculation time of 5-6 days on 32 computers is impractical for common design practice. However there is great potential to use these simulations to generate damping coefficients numerically for flooded compartments of different geometry and to use these results to improve those semi-empirical formulae typically used in design practice, such as those based on experiments by Katayama (2009).

6. ACKNOWLEDMENTS

The authors gratefully acknowledge the availability of 32 processors at Calculation Centre SCoPE, University of Naples and thanks to SCoPE academic staff for the given support.

7. REFERENCES

Avalos, G. O.G., Wanderley J. B.V., Fernandes A.C., Oliveira, A.C, 2014, “Roll damping decay of a FPSO with bilge keel”, Ocean Engineering 87 , pp. 111– 120

Begovic, E., Mortola, G., Incecik, A., and Day, A.H., 2013, “Experimental assessment of intact and damaged ship motions in head, beam and quartering seas”, Ocean Engineering, 72, pp. 209 - 226.

Fernandes A.C., Oliveira A.C., 2009, The roll damping assessment via decay model testing (new ideas about an old subject), J.Marine Science Application (2009), 8:144-150,

Gao, Z., Gao, Q., Vassalos, D., 2013, “Numerical study of damaged ship flooding in beam seas”, Ocean Engineering 61 , pp. 77 – 87.

Gao, Q., Vassalos, D., 2011, “Numerical study of the roll decay of intact and damaged ships”, Proceedings of the 12th International Ship Stability Workshop, Washington D.C., pp. 277–282.

Katayama T., Kotaki M., Katsui T., Matsuda A., 2009, “A Study on Roll Motion Estimation of Fishing Vessels with Water on Deck”, Journal of the Japan Society of Naval Architects and Ocean Engineers, Vol.9, pp.115-125 (in Japanese)

Lee Y., Chan H.S, Pu Y., Incecik A., Dow R.S., 2012, “Global Wave Loads on a Damaged Ship”, Ships and Offshore Structures 7(3), pp. 237-268

Lewandowsky E.M., 2004, "The Dynamics of Marine Craft: Maneuvering and Seakeeping", World Scientific Publishing Co. Pte Ltd, Advanced Series on Ocean Engineering, Vol. 22

Sadat-Hosseini H., Kim D.H, Lee S.K, Rhee S.H., Carrica P., Stern F., Rhee K.P., 2012, "CFD and EFD Study of Damaged Stability in Calm Water and Regular Waves", Proceedings of the 11th International Conference on the Stability of the Ships and Ocean Vehicles, Greece, pp. 425 – 452

Wilson, R.,V., Carrica, P.M., Stern F., 2006, "Unsteady RANS method for ship motions with application to roll for a surface combatant", Computers & Fluids 35, pp. 501–524

Yang, B., Wang, Z.C., Wu, M., 2012, "Numerical Simulation of Naval Ship's Roll Damping Based on CFD", Procedia Engineering 37, pp. 14 – 18

Yang, C. L., Zhu, R.C., Miao G.P., Fan J., 2013, "Numerical simulation of rolling for 3-D ship with forward speed and nonlinear damping analysis", Journal of Hydrodynamics, 25(1), pp. 148-155

ITTC Recommended Procedures and Guidelines., 2011, "Practical Guidelines for Ship CFD Applications", 7.5-03-02-03