**Application of RANS to the hydrodynamics of bilge keels and baffles**

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**SUMMARY**

This investigation first focuses on modelling the flow around cantilever plates in normal oscillation, providing key guidance for attaining sufficient accuracy for most engineering applications, in terms of turbulence modelling, mesh refinement and time step selection. The modelling approach is then applied to the hydrodynamics of a ship-like section fitted with vertical bilge keels, rolling at free surface. The results obtained from RANS are compared to available experimental data and predictions obtained from potential flow analysis. Viscous effects arising from the presence of the bilge keels are shown to influence more hydrodynamic coefficients than just roll fluid damping. Finally, empirical formulae commonly used to allow for the influence of viscous roll damping are compared to RANS predictions and experimental data.

# NOMENCLATURE

|  |  |
| --- | --- |
| *A* | Amplitude of forced sway motion |
| *B* | Mean waterline beam |
|  | Hydrodynamic forces acting on the body sway and heave directions |
| *h* | Height of the cantilever plate |
| *H* | Height of the equivalent free plate |
| *KC* | Keulegan-Carpenter number |
| *Mx (t)* | Hydrodynamic roll moment acting on the body |
| ,, | Added mass/inertia in sway and heave and roll |
| ,, | Added mass/inertia coefficient in sway and heave and roll |
| , | Cross-coupling added mass of roll-into-sway and sway-into-roll |
| , | Cross-coupling added mass coefficient of roll-into-sway and sway-into-roll |
| ,, | Fluid damping in sway and heave and roll |
| ,, | Fluid damping coefficient in sway and heave and roll |
| , | Cross-coupling fluid damping of roll-into-sway and sway-into-roll |
| , | Cross-coupling fluid damping coefficient of roll-into-sway and sway-into-roll |
| *q* | Tangential fluid velocity in vortex |
| *r* | Radial distance to vortex centre |
| *Re* | Reynolds number for oscillatory flows |
| *t* | Time |
| *T* | Period of oscillation |
| ** | Stokes parameter |
|  | Kinematic viscosity |
|  | Amplitude of forced roll motion |
| ** | Frequency of oscillation |
|  | Non-dimensional frequency of oscillations |
|  | Sectional area corresponding to the underwater part of the section (defined with respect to the undisturbed free surface) |

# INTRODUCTION

Cantilever plates oscillating normally, or in normal oscillatory flow, are widely used in the marine and offshore industries when increased fluid damping is required. Bilge keels are, for example, often needed to minimize roll motions of ships and FPSOs, while baffles are used to reduce liquid sloshing in LNG tanks. Numerical prediction of the forces acting on such appendages usually involves empirical formulae. In particular, the roll response of ships and FPSOs is commonly obtained by combining potential flow analysis to determine added mass and radiation damping, with empirical formulae to estimate viscous roll damping. This follows the assumption by Himeno that the total roll damping coefficient can be calculated as the sum of five major components, namely: roll damping due to radiating wave (potential flow), skin friction, eddy making, bilge keel (if applicable) and lift (zero in absence of forward speed). Although the component of roll damping due to radiating waves is well represented by potential flow analysis, the other components are generally estimated from empirical models, as discussed by Chakrabarti . Such approaches are, however, only valid for the range of geometries and flow conditions the empirical formulae have been derived from. To overcome this problem, viscous effects need to be accounted for in the numerical model.

The continuous increase in computational power has enabled the development of Reynolds Averaged Navier Stokes (RANS) flow solvers and, especially, their application to marine and offshore industries. Gentaz et al used a fully coupled two-dimensional method, which included a free surface model, to study the viscous and vortical flows around rolling ship sections. The method was then applied by de Jouette et al to a three-dimensional ship. Noting that the application of a RANS solver to the estimation of all components of motions is still computationally intensive, Salui et al proposed an alternative to the three-dimensional problem in calculating added inertia and fluid damping of two-dimensional sections forced to roll harmonically on a free surface initially undisturbed. Quérard et al ([6], , ) extended the RANS approach to symmetric (heave) and anti-symmetric (sway and roll) motions of typical ship sections (rectangular, triangular, chine and bulbous). The hydrodynamic coefficients predicted, including the cross-coupling coefficients of sway-into-roll and roll-into-sway , were in overall good agreement with measurements and offered improved prediction, compared to potential flow analysis, especially for sway and roll fluid damping. The approach was subsequently applied to the hydroelastic response of a flexible barge stationary in regular oblique waves, using a two-dimensional approach . The application considered all rigid body motions, except surge, as well as symmetric and antisymmetric distortions. Overall, the RANS approach enabled improved predictions compared to numerical methods involving potential flow analysis, particularly in the case of responses related to antisymmetric motions and distortions.

The aim of the present paper is to further develop the RANS approach introduced by Quérard et al ([6], , ) to the roll motion of ship sections with bilge keels. The underlying objective is to investigate the influence of strong viscous effects on the roll hydrodynamics.

# THEORY

## Governing equations of fluid motion

The flow solver uses an element-based finite volume formulation, combining the geometric flexibility of the finite element method with the strict conservation properties of the finite volume. With this method, a finite volume is constructed around each mesh node, and the surface fluxes are evaluated at the centre of each surface commonly shared with the adjacent control volume. The coupled-implicit Volume Of Fluid (VOF) method proposed by Zwart et al. is used to capture free surface interface. This method has the advantage of maintaining a sharp interface, without the need for a very small time step.

Turbulence closure in the momentum equation is obtained by using the standard k- model in most of the study, with comparison to the laminar model (which does not apply any turbulence model) and the Shear Stress Transport (SST) model developed by Menter carried out in section . Finally, the scalable wall function developed by Menter and Esch is used to model the viscous sublayer in the boundary layer, when using the standard k- model. With SST, the hybrid wall-treatment developed by Esch and Menter is applied. This wall-treatment switches automatically from a low-Reynolds number formulation to a scalable wall function, based on grid density. Application of these models is discussed in section .

## Calculation of the hydrodynamic coefficients

For an imposed harmonic displacement, say in sway, *y(t)=A.sin(t)*, the instantaneous values of added mass and fluid damping are obtained by performing Fourier analysis at the frequency of oscillation over a moving window with the width of a single period T, as first presented by Yeung et al . Thus, for sway, the sway added mass and fluid damping are:

(1)

(2)

Similar expressions can be written for added mass/inertia and fluid damping in heave (mzz, Nzz) and roll (I, N). Furthermore, the calculation of the cross coupling added mass and fluid damping between sway and roll follows the same approach . Thus, using the sway force due to roll motion, the corresponding sway-into-roll added mass and fluid damping are:

(3)

(4)

The hydrodynamic coefficients are then non-dimensionalised as follows:

and (5)

where k=0 for the heave and sway, k=1 for the sway into roll and roll into sway cross coupling and k=2 for the roll hydrodynamic coefficients.

It has to be emphasized that in the above equations, the forces and moments to which Fourier analysis is applied are the dynamic forces and moments, i.e. without any hydrostatic component. The first step to obtain these dynamic forces and moments is to subtract the hydrostatic pressure (taken as -gz below the still waterline z=0) from the total pressure. The dynamic force F and moment M are then obtained by integrating the dynamic pressure and shear force over the ship section’s surface (Quérard et al ). Convergence of the instantaneous hydrodynamic coefficients is obtained after a few cycles (for example, ten cycles for the oscillatory cantilever plate presented in section , and five cycles for the rectangular section rolling on free surface presented in section ). Once convergence is attained, the instantaneous hydrodynamic coefficients are averaged over the last oscillation cycle to obtain the equivalent steady state coefficients.

# Numerical MOdelLing

The calculations were all performed using ANSYS-CFX11.0.

## Computational domain and boundary conditions

The computational domain and boundary conditions used to model the cantilever plate oscillating normally (see section ) are shown in figure 1. The domain size was chosen large enough (namely 20h x 10h, where h is the height of the cantilever plate) to avoid any blockage effect or other interference from the far field boundary conditions, which were defined as Neumann boundary conditions to allow flow in and out, thereby ensuring no influence on the flow near the cantilever plate.



Figure 1 Computational fluid domain and boundary conditions used for the oscillatory cantilever plate.

The domain size and boundary conditions used to model the rectangular cylinder with vertical bilges rolling on the free surface (section ) are shown in figure 2. A wall boundary condition was imposed in the far-field to conserve water mass. Using the dispersion equation, g, the far field walls were positioned 7 wave lengths away from the section to prevent wave reflection. This implies that the absolute position of the far field walls is dependent on the frequency of oscillation.



Figure 2 Computational fluid domain and boundary conditions used for the rectangular cylinder with vertical bilges

In both models, the forced body motions were modelled using an Arbitrary Lagrangian-Eulerian (ALE) kinematic description of the mesh deformation (see and ). Mesh deformation in the vicinity of the body (plate or cylinder) was avoided by creating an inner fluid domain with identical motion characteristics. Note that in figure 1, boundary conditions labelled as ‘unspecified motion’ are deforming according to the ALE scheme.

## Mesh

A fully structured mesh was used to model the flow around the oscillatory cantilever beam, as shown in figure 3. Two zones of local mesh refinement were created:

* An anisotropic inflation layer near the horizontal plate, to capture the Stokes boundary layer
* An isotropic (ie with identical grid spacing in y and z) mesh refinement region around the cantilever plate to capture vortex shedding.

Mesh refinement in the Stokes boundary layer of the horizontal plate was kept identical during this investigation, with a first grid point positioned at y0=10-4m. Mesh sensitivity analysis (see section ) was performed in the vortex shedding region using three grids, with spacing (equal in y and z) corresponding to: *A*/3.125, *A*/6.25 and *A*/12.5 (where *A* is the forced motion amplitude).



Figure 3 Detail view of the mesh used for the oscillatory cantilever plate

For the rectangular cylinder with vertical bilges rolling at free surface, a hybrid meshing technique, applied by Quérard et al. ([6], , ), was used to allow for mesh refinement using high quality hexahedral cells in the vicinity of the section (extending ¼ wavelength horizontally and ½ wavelength vertically) and in the free surface region, while the rest of the domain comprises coarser prismatic cells, as seen in figure 4. Using the hybrid mesh concept, high level of mesh refinement was achieved for a reasonably small total mesh size (66,780 cells for the mesh shown in figure 4). The mesh used here was adapted from the rectangular cylinder with slightly round bilge corners extensively validated by Quérard et al. ([6], , ). The grid spacing used are as follows:

* Grid spacing tangent to wall near the bilges:   
  /25 (where is the roll amplitude, and B the beam)
* Number of cells per wave amplitude: 8
* Maximum horizontal grid spacing in 1st wave length: 40



Figure Detail view of the mesh used for the rectangular cylinder with vertical bilges, rolling at free surface (\*=0.9)

## Timestep

Influence of timestep selection is assessed in section for the oscillatory cantilever plate using three refinements: N=333, 500 and 750 fixed timesteps per oscillation cycle. For the rectangular cylinder with vertical bilge keels rolling at free surface, N=750 timesteps per oscillation cycle have been used, following the recommendations by Quérard et al. [17] for the rectangular cylinder with slightly round bilge corners, rolling at free surface.

## Turbulence model

Influence of turbulence model is assessed in section for the cantilever plate, by comparing the laminar model to the standard k- and SST turbulence models. For the rectangular cylinder with vertical bilge keels rolling at free surface, only the standard k-turbulence model was used, as SST was found by Quérard et al. [17] to generally overpredict all the hydrodynamic coefficients of a rectangular cylinder with slightly round bilge corners, rolling at free surface.

## Numerical scheme

The advection scheme used in all the simulations presented is the so-called ‘High-Resolution’ advection scheme, which computes locally the blend factor to be as close to 1 (second order) as possible without introducing local numerical instability (see Barth and Jesperson for further details on the scheme). Discretization of the transient terms was achieved by using the second order backward Euler scheme.

# Validation: Cantilever plate oscillating normally

The case of a cantilever plate of height h=0.06m oscillating normally with an amplitude *A*=0.02m at Stokes parameter =1845 (where is the Reynolds number for oscillatory flows, and the Keulegan Carpenter number) was chosen for validation of the model away from the free surface, with comparison to the experiments by Sarpkaya and O’Keefe . Due to the large height to thickness ratio (102:6) of the cantilever plate studied by Sarpkaya and O’Keefe , the present plate was modelled as infinitely thin.

Sarpkaya and O’Keefe have shown that the flow regimes involving flat plates oscillating normally depend only on the Keulegan-Carpenter number *KC*. Here *KC*=2.07, and is therefore below 3, which corresponds to a flow regime where a newly created vortex (‘Vortex1’ in figure 5) forms a counter-rotating pair (‘Vortex2’ and ‘Vortex3’ in figure 5) with the vortex shed in the next half-cycle and convects away in a diagonal direction dictated by the starting direction. The topology of the vortices predicted using the mesh with grid spacing in the vortex shedding region of A/12.5, 750 timesteps per cycle and the SST turbulence model (figure 6) shows fair agreement with the experiments (figure 5). Discrepancies mainly arise from the convected vortices (‘Vortex 2’ and ‘Vortex 3’), which are predicted further apart to each other than observed experimentally. Furthermore, these vortices, which are mostly circular in the experiment show significant eccentricity in the CFD models used in this investigation.



Figure Photograph of the flow pattern about a cantilever plate oscillating at KC=2.5 and =1845, t =(n+ 7/8)T



Figure Vector plot of the velocity field and corresponding flow-lines about an oscillatory cantilever plate (with KC=2.07), at t =(12+7/8)T (using a grid spacing in vortex shedding region of A/12.5, N=750 timesteps per cycle, and the SST model)

Added inertia *Cm* and drag coefficients *Cd* have been measured by Sarpkaya and O’Keefe for a range of *KC* values. Note that in , *Cm* and *Cd* of the cantilever plate are reported for an equivalent free plate of height H=2h. In this paper, it was however preferred to report the hydrodynamic coefficients for the cantilever plate itself. Indeed, it can be shown that the hydrodynamic coefficients of a cantilever plate can be obtained from an equivalent free plate as follows:

(6)

(7)

Although the Morison equation is probably more appropriate for this problem due to the presence of the quadratic fluid damping term , which better models non-linearity arising from vortex shedding, it was preferred here to use the linear form of the equation of motion (namely using the hydrodynamic coefficients presented in section ), as the latter is more conventional in ship motion investigations. Linearization of the drag coefficient is obtained following the method described by Bishop and Price . Noting that added mass *Ca* and added inertia *Cm* coefficients are identical here, the coefficients presented in section can therefore be related to *Cm* and *Cd*, namely:

(8)

(9)

At *KC*=2.07*,* it is found from [19] that and . Thus, for the cantilever plate and .

## Mesh refinement analysis

Influence of grid refinement in the vortex shedding region on the horizontal and vertical position of the centre of the three vortices is respectively shown in tables 1 and 2. Note that here, the origin of the horizontal and vertical coordinates is taken as the tip of the cantilever plate, at t=T. Position of the vortex centres was defined as the local minima of pressure. Mesh refinement has very little influence on the position of the centre of ‘Vortex 1’. Positions of the centres of ‘Vortex2’, and to a larger extent ‘Vortex 3’, are more sensitive to mesh refinement. Sensitivity of vortex centre position to mesh refinement is therefore mainly dependent upon the distance convected away from the cantilever plate.

Influence of grid refinement on the tangential velocity profile is shown in figure 7. In the newly generated ‘Vortex 1’, mesh refinement mainly affects the magnitude of the peak of tangential velocity, while the core radius (defined as the radial distance at which the peak of tangential velocity is reached) is not affected. In the convected vortices (‘Vortex 2’ and ‘Vortex 3’), however, both the magnitude of the peak of tangential velocity and the core radius are sensitive to mesh refinement. Mesh coarsening generally tends to underpredict tangential velocity of all three vortices and overpredict the core radius in the convected vortices.

Influence of grid refinement in the vortex shedding region on added mass and linearised fluid damping coefficients is shown in table 3. Although not shown, convergence of the hydrodynamic coefficient was found to be reached faster with the coarsest mesh than with the finest mesh. Overall, 10 oscillation cycles were required to reach convergence. Particular good agreement (0.09% discrepancy) is obtained between the added mass coefficient predicted using the finest mesh and the experimental data by Sarpkaya and O’Keefe . For the equivalent linear fluid damping coefficient, the results obtained with the finest mesh are slightly underpredicted (10.87%). Mesh coarsening tends to increase underprediction of both hydrodynamic coefficient.

|  |  |  |  |
| --- | --- | --- | --- |
|  | *A*/3.125 | *A*/6.25 | *A*/12.5 |
| Vortex 1 | -1.210 | -1.175 | -1.230 |
| Vortex 2 | 1.540 | 1.588 | 1.610 |
| Vortex 3 | -5.165 | -2.285 | -1.575 |

Table Influence of grid spacing in the vortex shedding region on the horizontal position of the vortex centre at t =(12+7/8)T, N=750

|  |  |  |  |
| --- | --- | --- | --- |
|  | A/3.125 | A/6.25 | A/12.5 |
| Vortex 1 | 0.015 | -0.090 | -0.185 |
| Vortex 2 | -1.535 | -2.300 | -2.310 |
| Vortex 3 | -6.090 | -3.785 | -4.490 |

Table Influence of grid spacing in the vortex shedding region on the vertical position of the vortex centre at t =(12+7/8)T, N=750

|  |  |
| --- | --- |
| (a) | (b) |
| (c) |  |

Figure Influence of grid spacing in the vortex shedding region on the tangential velocity profile of (a)‘Vortex 1’, (b)‘Vortex 2’and (c)‘Vortex 3’ at t =(12+7/8)T, N=750

|  |  |  |  |
| --- | --- | --- | --- |
|  |  | **myy\* [-]** | **Nyy\* [-]** |
| Experiments | | 2.400 | 0.363 |
| Present  RANS  simulations | *A*/3.125 | 2.261 | 0.303 |
| *A*/6.25 | 2.374 | 0.320 |
| *A*/12.5 | 2.402 | 0.324 |

Table Influence of grid spacing in the vortex shedding region on the linear hydrodynamic coefficient, with comparison to the experiments by Sarpkaya and O’Keefe , N=750.

## Timestep refinement analysis

Although not shown here, ‘Vortex1’ is very little influenced by timestep selection, while sensitivity to timestep selection is experienced for the convected vortices, in terms of tangential velocity profiles and core radii.

Despite the influence of timestep selection on the position and tangential velocity profile of the convected vortices, the hydrodynamic coefficients are marginally influenced, as seen from table 4. It could, therefore, be supposed that the closest vortex to the cantilever plate (‘Vortex 1’, which is not significantly influenced by timestep selection) has the most influence on the hydrodynamics of the oscillatory vertical plate.

|  |  |  |  |
| --- | --- | --- | --- |
|  |  | **myy\* [-]** | **Nyy\* [-]** |
| Experiments | | 2.400 | 0.363 |
| Present  RANS  simulations | N=333 | 2.410 | 0.324 |
| N=500 | 2.405 | 0.324 |
| N=750 | 2.402 | 0.324 |

Table Influence of timestep selection on the linear hydrodynamic coefficient, with comparison to the experiments by Sarpkaya and O’Keefe , using a grid spacing in vortex shedding region of A/12.5

## Turbulence model analysis

The flow patterns obtained using SST (shown in figure 6) and k- are both in fair agreement with the experiments reported by Sarpkaya and O’Keefe (figure 5). With the Laminar model, however, the vortices generated from the previous oscillation cycles do not decay, thereby interacting with ‘Vortex1’, ‘Vortex2’ and ‘Vortex3’. Consequently, the sway force history obtained using the Laminar model shows larger amplitude variations than with k- and SST turbulence models. Likewise, convergence of the time history of the hydrodynamic coefficients obtained with the Laminar model is not reached, even after 13 oscillation cycles. The hydrodynamic coefficients obtained with SST converge after the 8th cycle, whereas only 5 oscillation cycles are required to reach convergence when using k- turbulence model. Once converged, both k- and SST turbulence models predict an almost identical linear fluid damping coefficient, slightly lower than the experimental value, as shown in table 5. The best predictions of added mass coefficient are obtained with SST, while it is slightly over-predicted (7.5%) when using k-.

|  |  |  |  |
| --- | --- | --- | --- |
|  |  | **myy\* [-]** | **Nyy\* [-]** |
| Experiments | | 2.400 | 0.363 |
| Present  RANS  simulations | k- | 2.580 | 0.327 |
| SST | 2.402 | 0.324 |
| Laminar | Not converged | Not converged |

Table Influence of turbulence model on the linear hydrodynamic coefficient, with comparison to the experiments by Sarpkaya and O’Keefe , using a grid spacing in vortex shedding region of A/12.5 and N=750

# APPLICATION: RECTANGULAR CYLINDER WITH VERTICAL Bilge keels, rolling at free surface

Following the validation study of an oscillatory cantilever plate presented in section and of a rectangular cylinder with slightly round bilge corners ([6], , ), application of the RANS approach is made to the case of a rectangular cylinder fitted with vertical bilge keels, rolling harmonically at free surface. The cylinder is 0.5m wide, with a draft of 0.25m (beam:draught ratio of 2), and fitted with vertical bilge keels of thickness 5mm and height 0.02m. The cylinder is forced to roll harmonically about its mean waterline at an amplitude =0.05rad (corresponding to ) and for a range of frequencies corresponding to \*=0.4 – 0.9.

## Influence of bilge keels on the roll hydrodynamics

The influence of bilge keels on the roll hydrodynamics is made by comparing the hydrodynamic coefficients (including the cross coupling coefficients between sway and roll) of the cylinder described above to the ones obtained by Quérard et al. for a rectangular cylinder of beam 0.4m and draft 0.2m (beam:draught ratio of 2), with 2.5mm round bilge corners, forced to roll harmonically about its mean waterline at =0.05rad. For both sections, the hydrodynamics predicted by the RANS approach are compared with results obtained from potential flow analysis – taken from an inviscid Boundary Element Method (BEM) for the section with bilge keels, and from a seven-parameter conformal transformation representations for the section without bilge keels. Additionally, the predictions of roll fluid damping of the section with bilge keels are compared to the experimental data presented by Na et al .

The presence of bilge keels tends to increase significantly inviscid roll added inertia, as seen from the comparison of the potential flow predictions of the rectangular section without and with bilge keels, shown in figure 8.a. This is also confirmed by the comparison of the RANS predictions of the sections with and without bilge keels. The predictions of roll added inertia of the rectangular section with bilge keels obtained from potential flow analysis (BEM) are higher than those obtained from RANS. This corroborates with the observations made by Seah and Yeung for the section with bilge keels, comparing the results obtained from potential flow analysis (BEM) to the ones obtained from FSRVM (which models viscous effects using a Random Vortex Method), and is therefore likely to be related to viscous effects.

The bilge keel does not affect significantly the inviscid roll fluid damping, as seen from figure 8.b. The RANS predictions, which account for viscous effects, are however able to predict a significant increase when using bilge keels. The differences between the magnitude of the roll damping coefficients obtained from RANS and potential flow analysis is much more significant in presence of bilge keels than without. This tends to show that the increase in roll fluid damping when using bilge keels is essentially due to viscous effects. Note that the values measured by Na et al. [23] are relatively low compared to the RANS predictions; this corroborates with the observations by Seah and Yeung [21] who noted that these experimental values were abnormally close to the potential flow predictions, which was therefore attributed to experimental errors.

The bilge keels have overall no significant influence on cross-coupling added mass coefficient of sway-into-roll, as seen from the comparison of the RANS predictions without and with bilge keel shown in figure 8.c. For the cross-coupling fluid damping coefficient of sway-into-roll, the presence of the bilge keels has however a significant influence, as seen from the comparison of the RANS predictions shown in figure 8.d.

|  |  |
| --- | --- |
| (a) | (b) |
| (c) | (d) |

Figure Influence of the vertical bilge keels on the variations with \* of:

1. added inertia coefficient
2. roll fluid damping coefficient
3. sway into roll added mass coefficient
4. sway into roll fluid damping coefficient

## Viscous roll damping

A number of radiation-diffraction codes use empirical models to include viscous effects in roll fluid damping. This follows the assumption by Himeno that the individual components of roll damping can be added. For a stationary floating structure, roll damping arises from radiation damping and viscous roll damping. Here, the RANS viscous roll damping coefficients for the rectangular section with bilge keels are obtained by subtracting the roll fluid damping coefficients obtained from potential flow to the ones obtained from RANS (including free surface effects, as per section ). In a similar manner, the experimental viscous roll damping coefficients are obtained by subtracting the roll fluid damping coefficients predicted by potential flow analysis to the ones obtained experimentally . Comparison is made to the empirical formulation presented in Chakrabarti . This formulation breaks down roll fluid damping into skin friction Nf(taken from Kato ), eddy making Ne(taken from Tanaka ) and bilge keel Nbk(taken from Ikeda et al ) damping coefficients. A comparison of the different components of viscous roll damping coefficient is shown in figure 9 for the rectangular section with vertical bilge keels. It can be noted that bilge keel roll damping coefficient dominates the contributions from skin friction and eddy making and is therefore governing the viscous roll damping coefficient.

The frequency variations of viscous roll damping coefficients obtained from RANS and experimentally are compared in figure 10 to the ones obtained from the empirical formulation described above. The viscous roll damping obtained from the measurements by Na et al. are abnormally low compared to the results obtained from the empirical formulation or from RANS. Fair agreement is found between the results obtained from RANS and from the empirical formulation in the lower frequency range (\*=0.4-0.6). At higher frequencies, however, the empirical formulation leads to significant over-prediction of viscous roll damping compared to the RANS predictions. This discrepancy could be related to the fact that the empirical formulation assumes that each component of roll damping adds up linearly, thereby neglecting any interaction between the components. In particular, the eddy-making and the bilge keel components of damping are likely to interact. Such interactions are however inherently accounted for in the results obtained from RANS.

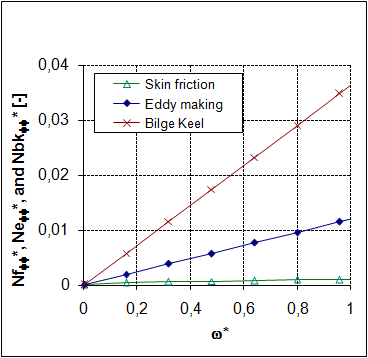


Figure 9 Comparison of skin friction (Nf), the eddy making (Ne), and bilge keel (Nbk) roll damping coefficients for the rectangular cylinder with vertical bilge keels.

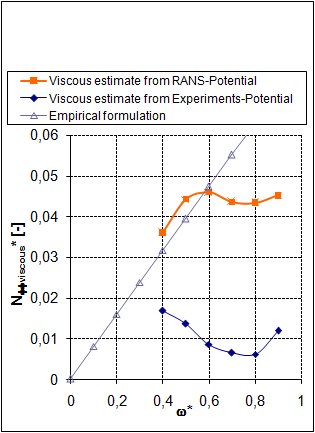


Figure 10 Comparison of frequency variations of viscous roll damping obtained from RANS and experimentally to the ones estimated from the empirical formulation, for the rectangular cylinder with vertical bilge keels.

# CONCLUSIONS

* 1. RANS modelling of oscillatory cantilever plates

The validation analysis of the oscillatory cantilever plate conducted to make general recommendations – in terms of meshing and timesteps selection – applicable to the modelling of low Keulegan-Carpenter oscillatory flows governed by vortical structures.

Position of the newly created tip vortex, its core radius and velocity profile were not found to be significantly influenced by mesh refinement or timestep selection. This is, however, not the case for the convected vortices, and their accurate capture requires high level of mesh refinement and a large number of timesteps per oscillation cycle. Nevertheless, the influence of the convected vortices on the hydrodynamic coefficients appears to be small, as seen for example in the analysis of timestep selection which influences significantly the convected vortices but marginally the hydrodynamic coefficients.

The Laminar model is not suited to model flow problems dominated by vortex shedding, because the model does not properly account for vortex decay, leading to non-physical flow features and large fluctuation in the time history of the instantaneous hydrodynamic coefficients. Convergence of the instantaneous hydrodynamic coefficients was found to be faster with k- turbulence model than with SST. Once converged, both k- and SST turbulence models predict a comparable linear fluid damping coefficient, slightly lower than the experimental value. The best predictions of added mass coefficient are obtained with SST, while k- leads to slight over-predictions.

* 1. Bilge keels

The presence of vertical bilge keels has been shown to increase roll added inertia and fluid damping as well as the magnitude of sway-into-roll fluid damping.

Viscous effects of such sections are not only increasing roll fluid damping and sway-into-roll fluid damping, but also have a significant influence (reduction) on roll added inertia. Incorrect modelling of these viscous effects could therefore not only lead to a wrong estimate of the roll response amplitude operator (RAO) at peak response, but also influence the prediction of the resonance frequency.

The empirical formulation used to calculate viscous roll damping has shown significant over-prediction compared to the results obtained from RANS in the higher frequency range of oscillation (\*>0.6). This over-estimation could be attributed to interactions between the various components of roll damping. Such interactions, which are indeed neglected by the empirical formulation, are inherently present in the RANS predictions. Furthermore, the empirical formulae only account for viscous effects in roll damping, neglecting the possible effects on other coefficients, or coupling influence into other motions.

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